Multidisciplinary Design Project Databook

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MDP Databook

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For more information about the project please visit the MDP website at http://www-mdp.eng.cam.ac.uk/ or contact :

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The Cambridge-MIT Institute (CMI) is a pioneering partnership between Cambridge University and the Massachusetts Institute of Technology (MIT). It was set up in 2000 to enhance the competitiveness, productivity and entrepreneurship of the UK economy by improving the effectiveness of knowledge exchange between university and industry, educating leaders and developing programmes for change in universities, industry and government.

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General Information

I

PERIODIC TABLE

Atomic Properties of the Elements

National Institute of Standards and Technology
Technology Administration, U.S. Department of Commerce

Physics Laboratory
physics.nist.gov

Standard Reference Data Group
www.nist.gov/srd

Frequently used fundamental physical constants

For the most accurate values of these and other constants, visit physics.nist.gov/constants

1 second = 9 192 631 770 periods of radiation corresponding to the transition between the two hyperfine levels of the ground state of ^{133}Cs

speed of light in vacuum	c	299 792 458 m s^{-1}	(exact)
Planck constant	h	6.6261 × 10^{-34} J s	($\hbar = h/2\pi$)
elementary charge	e	1.6022 × 10^{-19} C	
electron mass	m_e	9.1094 × 10^{-31} kg	
	m_ec^2	0.5110 MeV	
proton mass	m_p	1.6726 × 10^{-27} kg	
fine-structure constant	α	1/137.036	
Rydberg constant	R_∞	10 973 732 m^{-1}	
	$R_\infty c$	3.289 842 × 10^{15} Hz	
	$R_\infty hc$	13.6057 eV	
Boltzmann constant	k	1.3807 × 10^{-23} J K^{-1}	

Solids
Liquids
Gases
Artificially Prepared

Period	Group 1 IA	2 IIA	3 IIIB	4 IVB	5 VB	6 VIB	7 VIIB	8 VIII	9 VIII	10 VIII	11 IB	12 IIB	13 IIIA	14 IVA	15 VA	16 VIA	17 VIIA	18 VIIIA
1	1 $^2S_{1/2}$ **H** Hydrogen 1.00794 $1s$ 13.5984																	2 $^1S_0$ **He** Helium 4.002602 $1s^2$ 24.5874
2	3 $^2S_{1/2}$ **Li** Lithium 6.941 $1s^22s$ 5.3917	4 $^1S_0$ **Be** Beryllium 9.012182 $1s^22s^2$ 9.3227											5 $^2P^\circ_{1/2}$ **B** Boron 10.811 $1s^22s^22p$ 8.2980	6 $^3P_0$ **C** Carbon 12.0107 $1s^22s^22p^2$ 11.2603	7 $^4S^\circ_{3/2}$ **N** Nitrogen 14.0067 $1s^22s^22p^3$ 14.5341	8 $^3P_2$ **O** Oxygen 15.9994 $1s^22s^22p^4$ 13.6181	9 $^2P^\circ_{3/2}$ **F** Fluorine 18.9984032 $1s^22s^22p^5$ 17.4228	10 $^1S_0$ **Ne** Neon 20.1797 $1s^22s^22p^6$ 21.5645
3	11 $^2S_{1/2}$ **Na** Sodium 22.989770 [Ne]3s 5.1391	12 $^1S_0$ **Mg** Magnesium 24.3050 [Ne]$3s^2$ 7.6462											13 $^2P^\circ_{1/2}$ **Al** Aluminum 26.981538 [Ne]$3s^23p$ 5.9858	14 $^3P_0$ **Si** Silicon 28.0855 [Ne]$3s^23p^2$ 8.1517	15 $^4S^\circ_{3/2}$ **P** Phosphorus 30.973761 [Ne]$3s^23p^3$ 10.4867	16 $^3P_2$ **S** Sulfur 32.065 [Ne]$3s^23p^4$ 10.3600	17 $^2P^\circ_{3/2}$ **Cl** Chlorine 35.453 [Ne]$3s^23p^5$ 12.9676	18 $^1S_0$ **Ar** Argon 39.948 [Ne]$3s^23p^6$ 15.7596
4	19 $^2S_{1/2}$ **K** Potassium 39.0983 [Ar]4s 4.3407	20 $^1S_0$ **Ca** Calcium 40.078 [Ar]$4s^2$ 6.1132	21 $^2D_{3/2}$ **Sc** Scandium 44.955910 [Ar]$3d4s^2$ 6.5615	22 $^3F_2$ **Ti** Titanium 47.867 [Ar]$3d^24s^2$ 6.8281	23 $^4F_{3/2}$ **V** Vanadium 50.9415 [Ar]$3d^34s^2$ 6.7462	24 $^7S_3$ **Cr** Chromium 51.9961 [Ar]$3d^54s$ 6.7665	25 $^6S_{5/2}$ **Mn** Manganese 54.938049 [Ar]$3d^54s^2$ 7.4340	26 $^5D_4$ **Fe** Iron 55.845 [Ar]$3d^64s^2$ 7.9024	27 $^4F_{9/2}$ **Co** Cobalt 58.933200 [Ar]$3d^74s^2$ 7.8810	28 $^3F_4$ **Ni** Nickel 58.6934 [Ar]$3d^84s^2$ 7.6398	29 $^2S_{1/2}$ **Cu** Copper 63.546 [Ar]$3d^{10}4s$ 7.7264	30 $^1S_0$ **Zn** Zinc 65.409 [Ar]$3d^{10}4s^2$ 9.3942	31 $^2P^\circ_{1/2}$ **Ga** Gallium 69.723 [Ar]$3d^{10}4s^24p$ 5.9993	32 $^3P_0$ **Ge** Germanium 72.64 [Ar]$3d^{10}4s^24p^2$ 7.8994	33 $^4S^\circ_{3/2}$ **As** Arsenic 74.92160 [Ar]$3d^{10}4s^24p^3$ 9.7886	34 $^3P_2$ **Se** Selenium 78.96 [Ar]$3d^{10}4s^24p^4$ 9.7524	35 $^2P^\circ_{3/2}$ **Br** Bromine 79.904 [Ar]$3d^{10}4s^24p^5$ 11.8138	36 $^1S_0$ **Kr** Krypton 83.798 [Ar]$3d^{10}4s^24p^6$ 13.9996
5	37 $^2S_{1/2}$ **Rb** Rubidium 85.4678 [Kr]5s 4.1771	38 $^1S_0$ **Sr** Strontium 87.62 [Kr]$5s^2$ 5.6949	39 $^2D_{3/2}$ **Y** Yttrium 88.90585 [Kr]$4d5s^2$ 6.2173	40 $^3F_2$ **Zr** Zirconium 91.224 [Kr]$4d^25s^2$ 6.6339	41 $^6D_{1/2}$ **Nb** Niobium 92.90638 [Kr]$4d^45s$ 6.7589	42 $^7S_3$ **Mo** Molybdenum 95.94 [Kr]$4d^55s$ 7.0924	43 $^6S_{5/2}$ **Tc** Technetium (98) [Kr]$4d^55s^2$ 7.28	44 $^5F_5$ **Ru** Ruthenium 101.07 [Kr]$4d^75s$ 7.3605	45 $^4F_{9/2}$ **Rh** Rhodium 102.90550 [Kr]$4d^85s$ 7.4589	46 $^1S_0$ **Pd** Palladium 106.42 [Kr]$4d^{10}$ 8.3369	47 $^2S_{1/2}$ **Ag** Silver 107.8682 [Kr]$4d^{10}5s$ 7.5762	48 $^1S_0$ **Cd** Cadmium 112.411 [Kr]$4d^{10}5s^2$ 8.9938	49 $^2P^\circ_{1/2}$ **In** Indium 114.818 [Kr]$4d^{10}5s^25p$ 5.7864	50 $^3P_0$ **Sn** Tin 118.710 [Kr]$4d^{10}5s^25p^2$ 7.3439	51 $^4S^\circ_{3/2}$ **Sb** Antimony 121.760 [Kr]$4d^{10}5s^25p^3$ 8.6084	52 $^3P_2$ **Te** Tellurium 127.60 [Kr]$4d^{10}5s^25p^4$ 9.0096	53 $^2P^\circ_{3/2}$ **I** Iodine 126.90447 [Kr]$4d^{10}5s^25p^5$ 10.4513	54 $^1S_0$ **Xe** Xenon 131.293 [Kr]$4d^{10}5s^25p^6$ 12.1298
6	55 $^2S_{1/2}$ **Cs** Cesium 132.90545 [Xe]6s 3.8939	56 $^1S_0$ **Ba** Barium 137.327 [Xe]$6s^2$ 5.2117		72 $^3F_2$ **Hf** Hafnium 178.49 [Xe]$4f^{14}5d^26s^2$ 6.8251	73 $^4F_{3/2}$ **Ta** Tantalum 180.9479 [Xe]$4f^{14}5d^36s^2$ 7.5496	74 $^5D_0$ **W** Tungsten 183.84 [Xe]$4f^{14}5d^46s^2$ 7.8640	75 $^6S_{5/2}$ **Re** Rhenium 186.207 [Xe]$4f^{14}5d^56s^2$ 7.8335	76 $^5D_4$ **Os** Osmium 190.23 [Xe]$4f^{14}5d^66s^2$ 8.4382	77 $^4F_{9/2}$ **Ir** Iridium 192.217 [Xe]$4f^{14}5d^76s^2$ 8.9670	78 $^3D_3$ **Pt** Platinum 195.078 [Xe]$4f^{14}5d^96s$ 8.9588	79 $^2S_{1/2}$ **Au** Gold 196.96655 [Xe]$4f^{14}5d^{10}6s$ 9.2255	80 $^1S_0$ **Hg** Mercury 200.59 [Xe]$4f^{14}5d^{10}6s^2$ 10.4375	81 $^2P^\circ_{1/2}$ **Tl** Thallium 204.3833 [Hg]6p 6.1082	82 $^3P_0$ **Pb** Lead 207.2 [Hg]$6p^2$ 7.4167	83 $^4S^\circ_{3/2}$ **Bi** Bismuth 208.98038 [Hg]$6p^3$ 7.2855	84 $^3P_2$ **Po** Polonium (209) [Hg]$6p^4$ 8.414	85 $^2P^\circ_{3/2}$ **At** Astatine (210) [Hg]$6p^5$	86 $^1S_0$ **Rn** Radon (222) [Hg]$6p^6$ 10.7485
7	87 $^2S_{1/2}$ **Fr** Francium (223) [Rn]7s 4.0727	88 $^1S_0$ **Ra** Radium (226) [Rn]$7s^2$ 5.2784		104 $^3F_2$? **Rf** Rutherfordium (261) [Rn]$5f^{14}6d^27s^2$? 6.0 ?	105 **Db** Dubnium (262)	106 **Sg** Seaborgium (266)	107 **Bh** Bohrium (264)	108 **Hs** Hassium (277)	109 **Mt** Meitnerium (268)	110 **Uun** Ununnilium (281)	111 **Uuu** Unununium (272)	112 **Uub** Ununbium (285)		114 **Uuq** Ununquadium (289)		116 **Uuh** Ununhexium (292)		

Lanthanides	57 $^2D_{3/2}$ **La** Lanthanum 138.9055 [Xe]$5d6s^2$ 5.5769	58 $^1G^\circ_4$ **Ce** Cerium 140.116 [Xe]$4f5d6s^2$ 5.5387	59 $^4I^\circ_{9/2}$ **Pr** Praseodymium 140.90765 [Xe]$4f^36s^2$ 5.473	60 $^5I_4$ **Nd** Neodymium 144.24 [Xe]$4f^46s^2$ 5.5250	61 $^6H^\circ_{5/2}$ **Pm** Promethium (145) [Xe]$4f^56s^2$ 5.582	62 $^7F_0$ **Sm** Samarium 150.36 [Xe]$4f^66s^2$ 5.6437	63 $^8S^\circ_{7/2}$ **Eu** Europium 151.964 [Xe]$4f^76s^2$ 5.6704	64 $^9D^\circ_2$ **Gd** Gadolinium 157.25 [Xe]$4f^75d6s^2$ 6.1498	65 $^6H^\circ_{15/2}$ **Tb** Terbium 158.92534 [Xe]$4f^96s^2$ 5.8638	66 $^5I_8$ **Dy** Dysprosium 162.500 [Xe]$4f^{10}6s^2$ 5.9389	67 $^4I^\circ_{15/2}$ **Ho** Holmium 164.93032 [Xe]$4f^{11}6s^2$ 6.0215	68 $^3H_6$ **Er** Erbium 167.259 [Xe]$4f^{12}6s^2$ 6.1077	69 $^2F^\circ_{7/2}$ **Tm** Thulium 168.93421 [Xe]$4f^{13}6s^2$ 6.1843	70 $^1S_0$ **Yb** Ytterbium 173.04 [Xe]$4f^{14}6s^2$ 6.2542	71 $^2D_{3/2}$ **Lu** Lutetium 174.967 [Xe]$4f^{14}5d6s^2$ 5.4259
Actinides	89 $^2D_{3/2}$ **Ac** Actinium (227) [Rn]$6d7s^2$ 5.17	90 $^3F_2$ **Th** Thorium 232.0381 [Rn]$6d^27s^2$ 6.3067	91 $^4K_{11/2}$ **Pa** Protactinium 231.03588 [Rn]$5f^26d7s^2$ 5.89	92 $^5L^\circ_6$ **U** Uranium 238.02891 [Rn]$5f^36d7s^2$ 6.1941	93 $^6L_{11/2}$ **Np** Neptunium (237) [Rn]$5f^46d7s^2$ 6.2657	94 $^7F_0$ **Pu** Plutonium (244) [Rn]$5f^67s^2$ 6.0260	95 $^8S^\circ_{7/2}$ **Am** Americium (243) [Rn]$5f^77s^2$ 5.9738	96 $^9D^\circ_2$ **Cm** Curium (247) [Rn]$5f^76d7s^2$ 5.9914	97 $^6H^\circ_{15/2}$ **Bk** Berkelium (247) [Rn]$5f^97s^2$ 6.1979	98 $^5I_8$ **Cf** Californium (251) [Rn]$5f^{10}7s^2$ 6.2817	99 $^4I^\circ_{15/2}$ **Es** Einsteinium (252) [Rn]$5f^{11}7s^2$ 6.42	100 $^3H_6$ **Fm** Fermium (257) [Rn]$5f^{12}7s^2$ 6.50	101 $^2F^\circ_{7/2}$ **Md** Mendelevium (258) [Rn]$5f^{13}7s^2$ 6.58	102 $^1S_0$ **No** Nobelium (259) [Rn]$5f^{14}7s^2$ 6.65	103 $^2P^\circ_{1/2}$? **Lr** Lawrencium (262) [Rn]$5f^{14}7s^27p$? 4.9 ?

Key:

Atomic Number	58 $^1G^\circ_4$	Ground-state Level
Symbol	**Ce**	
Name	Cerium	
Atomic Weight†	140.116	
Ground-state Configuration	[Xe]$4f5d6s^2$	
	5.5387	Ionization Energy (eV)

†Based upon ^{12}C. () indicates the mass number of the most stable isotope.

For a description of the data, visit physics.nist.gov/data

NIST SP 966 (September 2003)

The charts/tables are taken from *Transactions* published by OMEGA. (Copies are included in the library section of the MDP resources) The Omega website, http://www.omega.com, contains extensive additional resources and data.

I

Pressure & Load Measurement

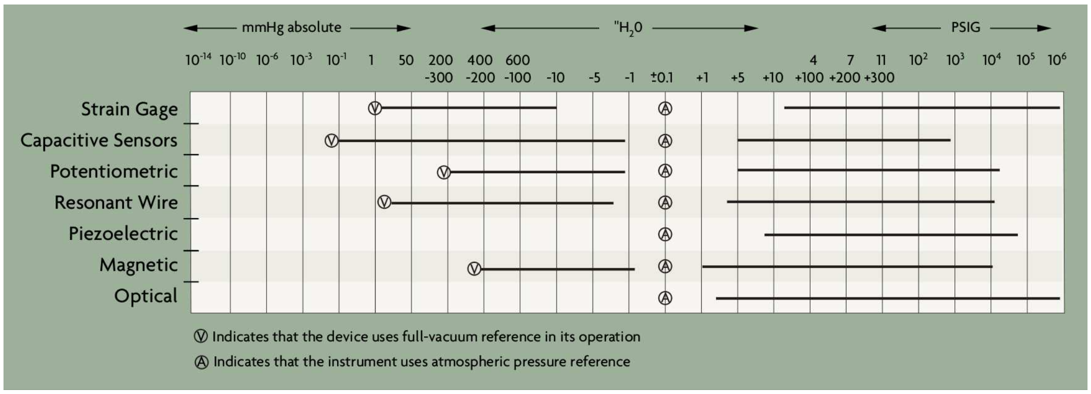

Figure 1: Electronic Pressure Sensor Ranges

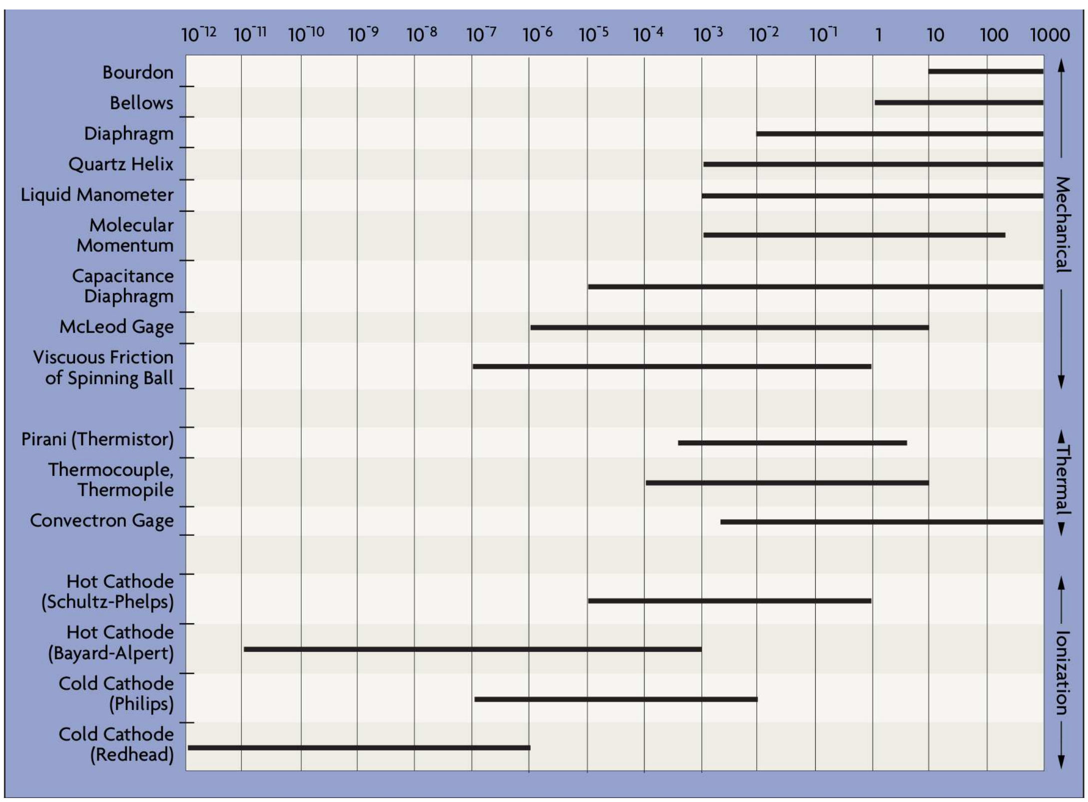

Figure 2: Vacuum Gauge Measurement Ranges

TYPE OF LOAD CELL	WEIGHT RANGE	ACCURACY (FS)	APPLICATIONS	ADVANTAGES	DISADVANTAGES
Mechanical Cells					
Hydraulic	Up to 10,000,000 lb	0.25%	Tanks, bins and hoppers Hazardous areas	Takes high impacts, insensitive to temperature	Expensive, complex
Pneumatic	Wide	High	Food industry, hazardous areas	Intrinsically safe Contains no fluids	Slow response Requires clean, dry air
Strain Gage Cells					
Bending Beam	10-5,000 lb	0.03%	Tanks, platform scales	Low cost, simple construction	Strain gages are exposed, require protection
Shear Beam	10-5,000 lb	0.03%	Tanks, platforms scales, off-center loads	High side load rejection, better sealing and protection	
Canister	to 500,000 lb	0.05%	Truck, tank, track, and hopper scales	Handles load movements	No horizontal load protection
Ring and Pancake	5-500,000 lb		Tanks, bins, scales	All stainless steel	No load movement allowed
Button and Washer	0-50,000 lb 0-200 lb typ.	1%	Small scales	Small, inexpensive	Loads must be centered, no load movement permitted
Other Types					
Helical	0-40,000 lb	0.2%	Platform, forklift, wheel load, automative seat weight	Handles off-axis loads, overloads, shocks	
Fiber Optic		0.1%	Electrical transmission cables, stud or bolt mounts	Immune to RFI/EMI and high temps, intrinsically safe	
Piezoresistive		0.03%		Extremely sensitive, high signal output level	High cost, nonlinear output

Figure 3: Load Cell Performance

Flow Measurement

TYPE OF DESIGN	DIRECT MASS-FLOW SENSOR	DIFFERENTIAL PRESSURE-FLOW SENSOR	VOLUME DISPLACEMENT-FLOW SENSOR	VELOCITY-FLOW SENSOR	EXPECTED ERROR FROM VISCOSITY CHANGE	TRANSMITTER AVAILABLE	LINEAR OUTPUT	RANGEABILITY	PRESSURE LOSS THRU SENSOR	APPROX. STRAIGHT PIPE-RUN REQUIREMENT ⑤ (UPSTREAM DIAM./DOWNSTREAM DIAM.)	FLOW RANGE ⑨
Orifice (plate or integral cell)		✓			H	✓	SR	3:1 ②	H	20/5	gpm—m³/hr; SCFM—Sm³/hr ③
Segmental Wedge		✓			M	✓	SR	3:1 ②	A	20/5	gpm—m³/hr
V-Cone Flowmeter		✓				✓	SR	3:1 to 15:1 ②	M	2/5	gpm—m³/hr; ACFM—Sm³/hr
Target Meters		✓			A	✓	SR	15:1	M	20/5	gpm—m³/hr; SCFM—Sm³/hr ④
Venturi Tubes		✓			H	✓	SR	3:1 ②	M	15/5	gpm—m³/hr; SCFM—Sm³/hr ③
Flow Nozzles		✓			H	✓	SR	3:1 ②	A	20/5	
Pitot Tubes		✓			M	✓	SR	3:1 ②	M	30/5	gpm—m³/hr; SCFM—Sm³/hr ④
Elbow Taps		✓			M	✓	SR	3:1 ②	N	25/10	gpm—m³/hr; SCFM—Sm³/hr ③
Laminar Flowmeters		✓				✓	✓	10:1 ②	H	15/5	gpm—m³/hr; SCFM—Sm³/hr
Magnetic Flowmeters				✓	N	✓	✓	30:1 ⑦	N	5/3	gpm—m³/hr
Positive Displacement Gas Meters			✓			SD	✓	10:1 to 200:1	M	N	SCFM—Sm³/hr
Positive Displacement Liquid Meters			✓		M	SD	✓	10:1 ⑦	A	N	gpm—m³/hr
Turbine Flowmeters			✓	✓	H	✓	✓	10:1 ⑧	A	15/5	gpm—m³/hr; SCFM—Sm³/hr
Ultrasonic Flowmeters											
Time of Flight			✓	✓	N	✓	✓	20:1	N	20/5	gpm—m³/hr ③
Doppler			✓	✓	N	✓	✓	10:1	N	20/5	SCFM—Sm³/hr
Variable Area (Rotamater)					A	✓	✓	10:1	M	N	gpm—m³/hr; SCFM—Sm³/hr
Vortex Shedding	SD			✓	A	✓	✓	10/1 ⑥	A	20/5	gpm—m³/hr
Fluidic Oscillation (Coanda)				✓	A	✓	✓	12/1 ⑥	H	20/5	ACFM—Sm³/hr
Mass Flowmeters Coriolis	✓				N	✓	✓	20:1	M/H	N	lbm—kgm/hr; SCFM—Sm³/hr
Mass Flowmeters Thermal Probe	✓			SD	N	✓	✓	20:1 ⑦	M	20/5	gpm—m³/hr; SCFM—Sm³/hr
Solids Flowmeters	SD					✓	✓	5:1 to 80:1	-	5/3	lbm—kgm/hr
Weirs, Flumes					M	✓	SD	100:1	M	4/1	gpm—m³/hr ③

- - - - = Non-standard Range
L = Limited
SD = Some Designs
H = High
A = Average
M = Minimal
N = None
SR = Square Root

① = The data in this column is for general guidance only.
② = Inherent rangeability of primary device is substantially greater than shown. Value used reflects limitations of differential pressure sensing device when 1% of rate accuracy is desired. With multiple-range intelligent transmitters, rangeability can reach 10:1.
③ = Pipe size establishes the upper limit.
④ = Practically unlimited with probe type design.
⑤ = Varies with upstream disturbance.
⑥ = Can be more with high Reynolds number services.
⑦ = Up to 100:1.
⑧ = More for gas turbine meters.
⑨ = Higher and lower flow ranges may be available. Check several manufacturers.

Figure 4: Flow Sensor Measurement Ranges/Operation

I

FLOWMETER	PIPE SIZE, in. (mm)	STEAM	GASES (VAPORS) CLEAN	GASES DIRTY	GASES PRESS HIGH	GASES PRESS LOW	LIQUIDS CLEAN	LIQUIDS VISCOUS HIGH	LIQUIDS VISCOUS LOW	LIQUIDS DIRTY	LIQUIDS CORROSIVE	LIQUIDS VERY CORROSIVE	LIQUIDS SLURRIES FIBROUS	LIQUIDS SLURRIES ABRASIVE	REVERSE FLOW	PULSATING FLOW	HIGH TEMPERATURE	CRYOGENIC	SEMI-FILLED PIPES	NON-NEWTONIANS	OPEN CHANNEL	TYPICAL Accuracy, uncalibrated (Including transmitter)	TYPICAL Reynolds number ‡ or viscosity	TEMPERATURE °F (°C)	PRESSURE psig (kPa)
SQUARE ROOT SCALE: MAXIMUM SINGLE RANGE 4:1 (Typical)**																									
Orifice																									
Square-Edged	›1.5 (40)	√	√	X	√	√	√	X	?	X	?	X	X	X	SD	?	√	√	X	?	X	±1-4% URV	R_D › 10,000	Process temperature to 1000°F (540°C); Transmitter limited to -30-250°F (-30-120°C)	To 4,000 psig (41,000 kPa)
Honed Meter Run	0.5-1.5 (12-40)	√	√	X	√	√	√	?	?	X	?	X	X	X	SD	?	√	√	X	?	X	±1% URV	R_D › 10,000		
Integrated	‹0.5 (12)	?	√	X	√	√	√	X	?	X	?	?	X	X	SD	?	?	X	X	?	X	±2-5% URV	R_D › 10,000		
Segmental Wedge	‹12 (300)	√	√	√	√	√	√	?	√	?	?	X	?	?	SD	?	√	√	X	?	X	±0.5% URV	R_D › 500		
Eccentric	›2 (50)	?	?	√	√	√	?	X	?	?	?	X	?	X	SD	?	√	√	X	?	X	±2-4% URV	R_D › 10,000		
Segmental	›4 (100)	?	?	√	√	√	?	X	?	?	?	X	?	X	SD	?	√	√	X	?	X	±2-4% URV	R_D › 10,000		
V-Cone	0.5-72 (12-1800)	√	√	?	√	√	√	?	√	?	?	X	?	?	X	?	?	?	X	?	X	±0.5-1% of rate	R_D : 8,000-5,000,000	700 (370)	≤600 (4,100)
Target***	‹0.5(12)	?	√	√	√	√	√	?	√	√	?	X	X	X	?	X	?	?	X	?	X	±0.5-5% URV	R_D › 100	Process temperature to 1000°F (540°C); Transmitter limited to -30-250°F (-30-120°C)	To 4,000 psig (41,000 kPa)
Venturi	›2 (50)	√	√	?	√	√	√	?	√	?	?	X	√	?	X	?	?	?	X	?	X	±0.5-2% URV	R_D › 75,000Ł		
Flow Nozzle	›2 (50)	?	√	?	√	√	√	X	?	?	?	X	X	X	X	?	?	?	X	?	X	±1-2% URV	R_D › 50,000Ł		
Low Loss Venturi	›3 (75)	√	√	X	√	√	√	X	?	X	√	X	X	X	X	?	?	?	X	?	X	±1.25% URV	R_D › 12,800Ł		
Pitot	›3 (75)	X	√	X	√	√	√	X	?	X	?	X	X	X	X	X	?	?	X	X	X	±3-5% URV	R_D › 100,000Ł		
Averaging Pitot	›1 (25)	√	√	SD	√	√	√	X	?	SD	?	X	X	X	SD	X	?	?	X	X	X	±1-2% URV	R_D › 40,000Ł		
Elbow	›2 (50)	X	√	?	√	√	√	X	?	?	?	X	X	X	√	X	?	?	X	?	X	±5-10% URV	R_D › 10,000Ł		
Laminar	0.25-16.6 (6-400)	?	√	X	√	√	√	√	?	X	?	X	X	X	X	√	X	X	X	X	X	±1% of rate	R_D ‹ 500	150 (66)	≤30 (225)
LINEAR SCALE TYPICAL RANGE 10:1 (Or better)																									
Magnetic*	0.1-72 (2.5-1800)	X	X	X	X	X	√	?	√	√	√	√	√	√	√	√	?	X	?	?	?	±0.5% of rate	R_D › 4,500	360 (180)	≤ 1,500 (10,800)
Positive Displacement																									
Gas	‹12 (300)	X	√	X	?	√	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	±1% of rate	-	250 (120)	≤ 1,400 (10,000)
Liquid	‹12 (300)	X	X	X	X	X	√	√	?	X	?	X	X	X	X	X	?	X	X	X	X	±0.5% of rate	No R_D limit ≤ 8,000 cS	600 (315)	≤ 1,400 (10,000)
Turbine																									
Gas	0.25-24 (6-600)	SD	√	X	√	√	X	X	X	X	X	X	X	X	SD	SD	?	?	X	X	?	±0.5% of rate	-	-450-500 (268-260)	≤ 3,000 (21,000)
Liquid	0.25-24 (6-600)	X	X	X	X	X	√	X	?	X	?	X	X	SD	SD	SD	?	?	X	X	?	±0.5% of rate	R_P › 5,000, ≤15 cS	-450-500 (268-260)	≤ 3,000 (21,000)
Ultrasonic																									
Time of Flight	›0.5 (12)	X	SD	SD	SD	SD	√	?	?	X	√	√	?	?	√	√	X	?	X	X	?	±1% of rate to ffl5% URV	R_D › 10,000	-300-500 (-180-260)	Pipe rating
Doppler	›0.5 (12)	X	X	X	X	X	X	?	?	√	√	√	√	√	√	√	X	X	X	?	X	±1% of rate to ffl5% URV	R_D › 4,000	-300-500 (-180-260)	Pipe rating
Variable-Area (Rotameter)	≤3 (75)	?	√	X	X	√	√	X	?	X	?	?	X	X	X	?	?	?	X	X	X	±1% of rate to ffl10% URV	No R_D limit, ‹ 100 cS	Glass: 400 (200) Metal: 1,000 (540)	Glass: 350 (2,400) Metal: 720 (5,000)
Vortex Shedding	1.5-16 (40-400)	√	√	?	√	√	√	X	?	?	?	X	X	X	X	X	?	?	X	X	X	±0.75-1.5% of rate	R_D › 10,000, ‹ 30 cP	400 (200)	≤ 1,500 (10,500)
Vortex Precession (Swirl)	‹16 (400)	√	√	?	√	√	√	X	?	X	?	X	X	X	X	X	?	X	X	X	X	±0.5% of rate	R_D › 10,000, ‹ 5 cP	536 (280)	Pipe rating
Fluidic Oscillation (Coanda)	›1.5 (40)	X	X	X	X	X	√	X	X	?	?	X	X	X	X	?	?	?	X	X	X	±2% of rate	R_D › 2,000, ‹ 80 cS	350 (175)	≤ 720 (5,000)
Mass																									
Coriolis	0.25-6 (6-150)	?	?	?	√	√	√	√	√	√	?	?	?	√	?	?	?	?	X	√	X	±0.15-10% of rate	No R_D limit	-400-800 (-224-427)	≤ 5,700 (39,900)
Thermal Probe	‹72 (1800)	X	√	?	√	√	√	?	√	√	?	?	?	?	X	?	?	X	X	?	X	±1-2% URV	No R_D limit	1,500 (816)	Pipe rating
Solids Flowmeter	‹24 (600)	X	X	X	X	X	X	SD	X	?	X	X	SD	SD	X	SD	SD	X	√	X	X	±0.5% of rate to ffl4% URV	-	750 (400)	≤ 580 (4,000)
Correlation																									
Capacitance	‹8 (200)	X	X	X	X	X	X	√	√	√	√	√	√	√	X	?	?	X	?	?	X	No data available	No data available	300 (149)	≤ 580 (4,000)
Ultrasonic	›0.5 (12)	X	X	X	X	X	X	?	√	√	√	√	√	√	X	?	X	X	X	?	X	±6% of ??	No data available	-300-250 (-180-120)	Pipe rating

cP = centi Poise
cS = centi Stokes
SD = Some designs
? = Normally applicable (worth consideration)
√ = Designed for this application (generally suitable)
URV = Upper Range Value
X = Not applicable
‡ According to other sources, the minimum Reynolds number should be much higher
* Liquid must be electrically conductive
** Range 10:1 for laminar, and 15:1 for target
*** Newer designs linearize the signal

Figure 5: Flow Sensor Operation Comparison

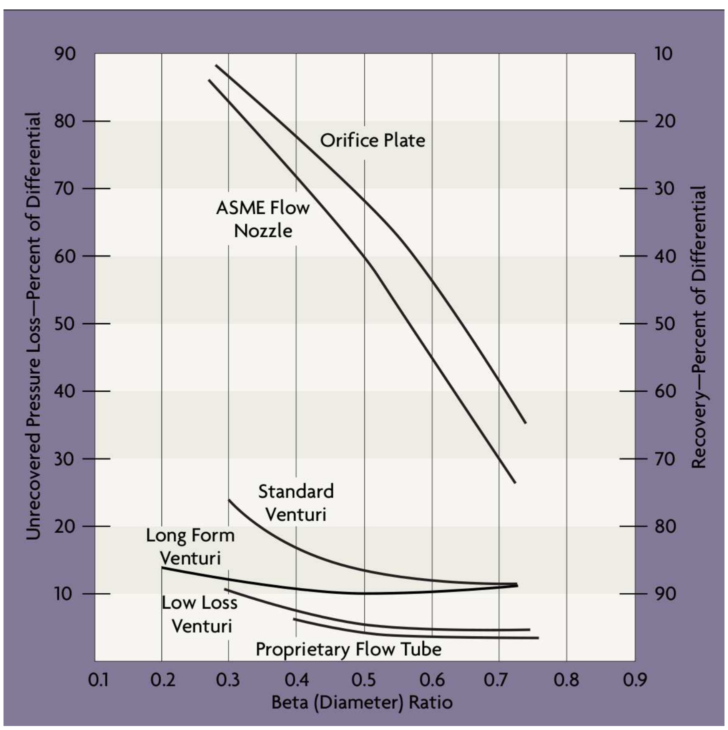

Figure 6: Pressure drop in Flowmeters

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Part II
Mechanical Information

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Contents

User's Guide to Adhesives

Introduction

Almost everything that is made by industry has component pieces, and these have to be fixed together. Often mechanical connections are chosen, such as screws, rivets or spot welds. However, engineers now often choose to use adhesive bonding. This joining technique is well proven and capable of replacing or supplementing mechanical fixing methods and has advantages which include:

- Reduced component and/or assembly costs
- Improved product performance and durability
- Greater design freedom
- Less finishing operations

This guide sets out to remove the reservations that engineers sometimes have about adhesives. It includes a survey of modern adhesives and shows how joints should be designed and pre-treated in order to make best use of adhesive bonding.

The guide comes from the inventors of adhesives capable of bonding metals. Our Araldite® trade name is known world wide in industry and in the home.

A word about adhesives

What are we doing when we seek to use an adhesive? The question is not new. Man has used adhesives or glues since the dawn of history. The ancient Egyptians attached veneers to furniture with glue. These early glues were all natural substances. Nowadays we use synthetic resins and polymers.

When we bond components together the adhesive first thoroughly wets the surface and fills the gap between. Then it solidifies. When solidification is completed the bond can withstand the stresses of use. The strongest adhesives solidify through chemical reaction and have a pronounced affinity for the joint surfaces. Adhesive bonding is sometimes called chemical joining to contrast it with mechanical joining.

Designing to bond.

In order to get the best performance from an adhesive bond it is important to design the component for bonding rather than simply taking a design made for mechanical fixing.

Methods of application of the adhesive and the assembly of the components must always be considered at the design stage, together with the practical curing conditions, which affect the choice of adhesive type to be used.

A quality bond is produced when quality is considered at all stages of the design and production process.

Advantages of adhesive bonding

The bond is continuous: On loading, there is more uniform distribution of stresses over the bonded area. The local concentrations of stresses present in spot welded or mechanically fastened joints are avoided. Bonded stuctures can consequently offer a longer life under load.

Stiffer structures: The bonded joint – being continuous – produces a stiffer structure. Alternatively, if increased stiffness is not needed, the weight of the structure can be decreased while maintaining the required stiffness.

Improved appearance: Adhesive bonding gives a smooth appearance to designs. There are no protruding fasteners such as screws or rivets, and no spot-welds marks.

Complex assemblies: Complex assemblies that cannot be joined together in any other feasible way with adhesives. Composite sandwich structures are a typical example.

Dissimilar materials: Adhesives can join different materials together – materials that may differ in composition, moduli, coefficients of expansion, or thickness.

Reduced corrosion: The continuous adhesive bond forms a seal. The joint is consequently leak proof and less prone to corrosion.

Electrically insulating: The adhesive bond can provide an electrically insulating barrier between the surfaces.

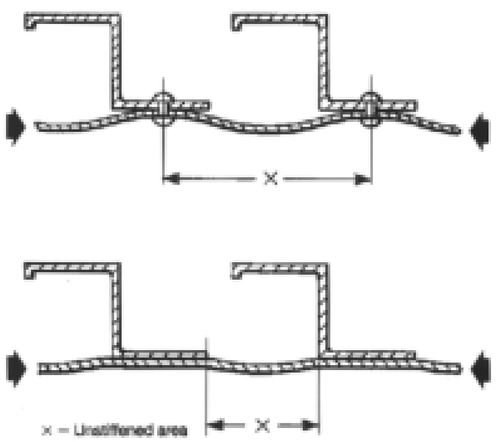

Fig.1 Stiffening effect – bonding and riveting compared

The diagram shows how a joint may be designed to take advantage of the stiffening effect of bonding.
Adhesives form a continuous bond between the joint surfaces. Rivets and spot welds pin the surfaces together only at localised points. Bonded structures are consequently much stiffer and loading may be increased (by up to 30 – 100%) before buckling occurs.

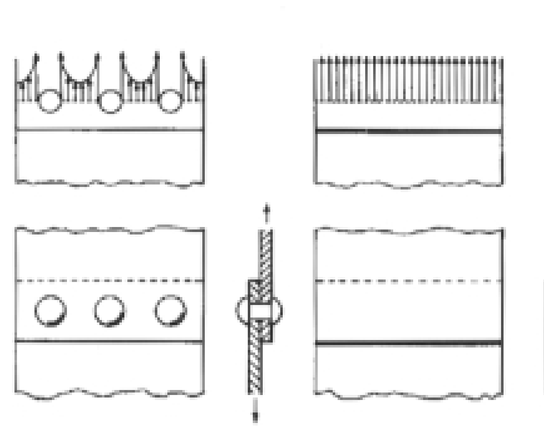

Fig. 2 Stress distribution in loaded joints

The riveted joint on the left is highly stressed in the vicinity of the rivets. Failure tends to initiate in these areas of peak stress. A similar distribution of stress occurs with spot welds and bolts.
The bonded joint on the right is uniformly stressed. A continuous welded joint is likewise uniformly stressed but the metal in the heated zone will have undergone a change in strength.

Reduced stress concentrations: The bonded structure is a safer structure because, owing to the fewer and less severe concentrations of stresses, fatigue cracks are less likely to occur. A fatigue crack in a bonded structure will propagate more slowly than in a riveted structure – or even in a machined profile because the bond-lines act as a crack stopper.

Jointing sensitive materials: Adhesive bonding does not need high temperatures. It is suitable means for joining together heat-sensitive materials prone to distortion or to a change in properties from the heat of brazing or welding.

Vibration damping: Adhesive bonds have good damping properties. The capacity may be useful for reducing sound or vibration.

Simplicity: Adhesive bonding can simplify assembly procedures by replacing several mechanical fasteners with a single bond, or by allowing several components to be joined in one operation.

Adhesive bonding may be used in combination with spot welding or riveting techniques in order to improve the performance of the complete structure. All these advantages may be translated into economic advantages: improved design, easier assembly, lighter weight (inertia overcome at low energy expenditure), longer life in service.

Limitations

Temperature resistance: Adhesives are drawn from the class of materials which we know as 'polymers', 'plastics' or 'synthetic resins'. They have the limitations of that class. They are not as strong as metals. (the difference is offset by the increased surface contact area provided by the bonded joints). With increasing temperature the bond strength decreases, and the strain properties of the adhesive move from elastic to plastic. This transition is usually in the temperature range 70 – 220^{0}C: the transition temperature depends on the particular adhesive.

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Chemical resistance: The resistance of bonded joints to the in-service environment is dependent on the properties of the polymer from which the adhesive is made. Possible exposure of the bonded structure to oxidising agents, solvents, etc., must be borne in mind when selecting the adhesive type to use.

Curing time: With most adhesives maximum bond strength is not produced instantly as it is with mechanical fastening or with welding. The assembled joint must be supported for at least part of the time during which the strength of the bond is building up. The quality of the bond may be adversely affected if, in the bonding process, the surfaces are not readily wetted by the adhesive.

Process controls: Ensuring consistently good results may necessitate the setting up of unfamiliar process controls. A badly made joint is often impossible to correct.

In service repair: Bonded assemblies are usually not easily dismantled for in-service repair.

Modern adhesives: types and main characteristics

Modern adhesives are classified either by the way they are used or by their chemical type. The strongest adhesives solidify by a chemical reaction. Less strong types harden by some physical change. Key types in today's industrial scene are as follows.

Anaerobics: Anaerobic adhesives harden when in contact with metal and air is excluded, e.g. when a screw is tight in a thread. Often known as 'locking compounds' or 'sealants', they are used to secure, seal and retain turned, threaded, or similarly close-fitting parts. They are based on synthetic resins known as acrylics. Due to the curing process, anaerobic adhesives do not have gap-filling capability but have advantage of relatively rapid curing.

Cyanoacrylates: A special type of acrylic, cyanoacrylate adhesives cure through reaction with moisture held on the surfaces to be bonded. They need close-fitting joints. Usually they solidify in seconds and are suited to small plastic parts and to rubber. Cyanoacrylate adhesives have relatively little gap-filling capability but can be obtained in liquid and thixotropic (non-flowing) versions.

Toughened Acrylics/Methacrylates: A modified type of acrylic, these adhesives are fast-curing and offer high strength and toughness. Supplied as two parts (resin and catalyst), they are usually mixed prior to application, but specialised types are available which are applied by separate application: resin to one bond surface, catalyst to the other. They tolerate minimal surface preparation and bond well to a wide range of materials. The products are available in a wide range of cure speeds and as liquids or pastes which will gap-fill up to 5mm.

UV curable adhesives: Specially modified acrylic and epoxy adhesives, which can be cured very rapidly by exposure to UV radiation. Acrylic UV adhesives cure extremely rapidly on exposure to UV but require one substrate to be UV transparent. The UV initiated epoxy adhesives can be irradiated before closing the bondline, and cure in a few hours at ambient temperature or may be cured at elevated temperature.

Epoxies: Epoxy adhesives consist of an epoxy resin plus a hardener. They allow great versatility in formulation since there are many resins and many different hardeners. They form extremely strong durable bonds with most materials. Epoxy adhesives are available in one-part or two-part form and can be supplied as flowable liquids, as highly thixotropic products with gap-filling capability of up to 25mm, or as films.

Polyurethanes: Polyurethane adhesives are commonly one part moisture curing or two-part. They provide strong resilient joints, which are resistant to impacts. They are useful for bonding GRP (glassfibre-reinforced plastics) and certain thermoplastic materials and can be made with a range of curing speeds and supplied as liquids or with gap-filling capability of up to 25mm.

Modified Phenolics: The first adhesives for metals, modified phenolics now have a long history of successful use for making high strength metal-to-metal and metal-to-wood joints, and for bonding metal to brake-lining materials. Modified phenolic adhesives require heat pressure for the curing process.

The above types set by chemical reactions. Types that are less strong, but important industrially, are as follows:

Hot Melts: Related to one of the oldest forms of adhesive, sealing wax, today's industrial hot melts are based on modern polymers. Hot melts are used for the fast assembly of structures designed to be only lightly loaded.

Plastisols: Plastisol adhesives are modified PVC dispersions which require heat to harden, The resultant joints are often resilient and tough.

Rubber adhesives: Based on solutions of latexes, rubber adhesives solidify through loss of solvent or water. They are not suitable for sustained loading.

Polyvinyl Acetates (PVAs): Vinyl acetate is the principal constituent of the PVA emulsion adhesives. They are suited to the bonding of porous materials, such as paper or wood, and general packaging work.

Pressure-sensitive adhesives: Suited to use on tapes and labels, pressure-sensitive adhesives do not solidify but are often able to withstand adverse environments. They are not suitable for sustained loading.

No one company supplies all these types of adhesives. Each supplier specialises in particular types. Huntsman Advanced Materials supplies many industries with epoxy, polyurethane, modified phenolic, toughened methacrylate and UV curable acrylic adhesives under the tradenames Araldite®, Epibond®, Epocast® and Uralane®.

Designing a bonded joint

It is important that bonded articles are designed with bonding in mind, rather than simply bonding a design made for welding or mechanical joining. When designing bonded joints the considerations include:

- Joint geometry
- Adhesive selection
- Mechanical properties of adhesive and adherent
- Stress in the joint
- Manufacturing conditions

Bonded joints may be subjected to tensile, compressive, shear or peel stresses, often in combination. (See Figure 3). Adhesives are strongest in shear, compression and tension. They perform less effectively under peel and cleavage loading. A bonded joint needs to be designed so that the loading stresses will be directed along the lines of the adhesive's greatest strengths.

To indicate the performance of an Araldite, Epibond, Epocast or Uralane adhesive, the Huntsman Advanced Materials Instruction
Sheet for the particular adhesive quotes the shear strengths and peel strengths obtained by standard test methods. For example, the standard test method for shear (ISO4587) uses a simple lap joint made from metal sheet, usually an aluminium alloy, 25mm wide with 12.5mm overlap. The mean breaking stress at room temperature will be in the range 5 to 45 N/mm² depending on the adhesive. At the top end of this breaking stress range, joints made from aluminium alloy sheet of up to 1.5mm thickness will yield or break in the metal. (The lap joint is only one of sveral different types of bonded joint).

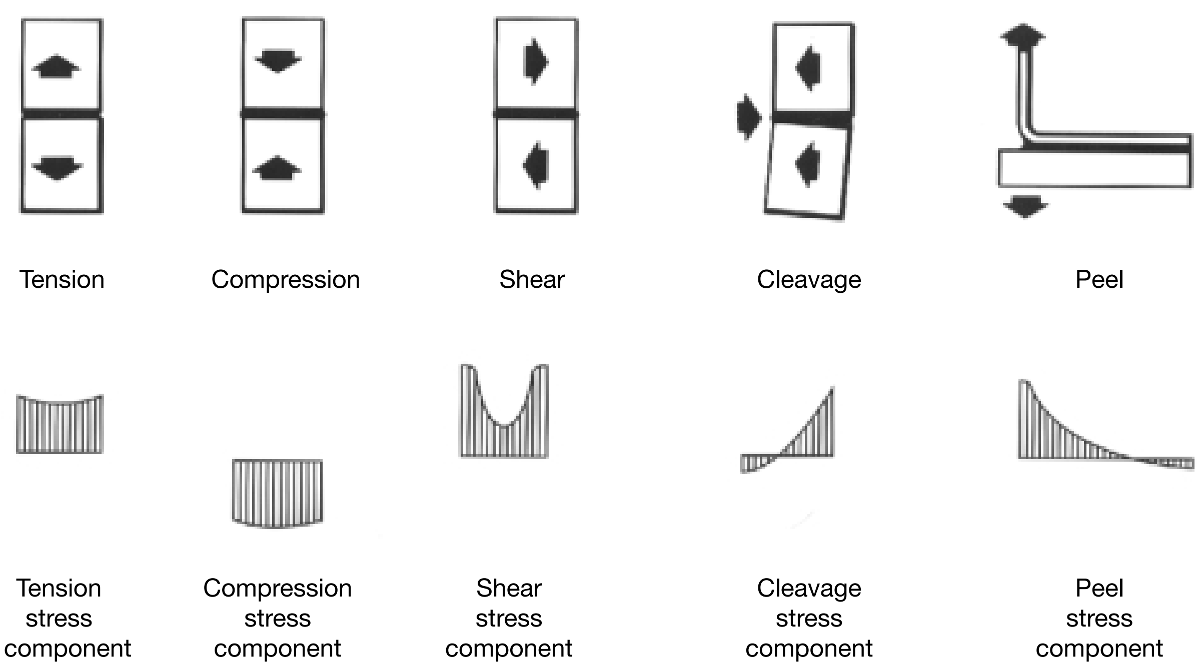
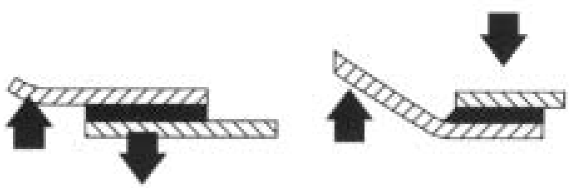


Fig.1 Loading conditions

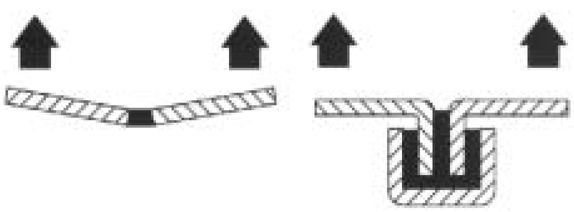
A bonded joint can be loaded in five basic ways (shown in the diagram). Cleavage and peel loading are the most taxing: they concentrate the applied force into a single line of high stress. In practice a bonded structure has to sustain a combination of forces. For maximum strength, cleavage and peel stresses should be as far as possible designed out of the joints.

The breaking load of a lap joint is proportional to its width, but not to its overlap length. Although the breaking load will increase as overlap length is increased, the mean breaking stress will be reduced.
A method of determining the best dimensions for a simple lap joint is described in Simple Lap Joints: Determination of dimensions (page 10).

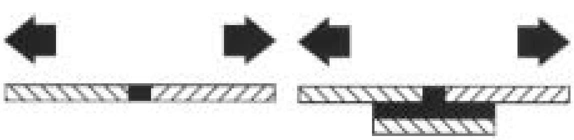
The strength of a joint is a complex function of the stress concentrations set up by the load. In a simple lap joint made from thin metal sheet there are two sorts of stress: shear and peel. Both the shear and peel stresses vary along the length of the joint, with concentrations at the ends. Alternative joint designs are shown in Figure 4 where these stresses are more evenly distributed. The efficiency gained results in joints of greater strength.



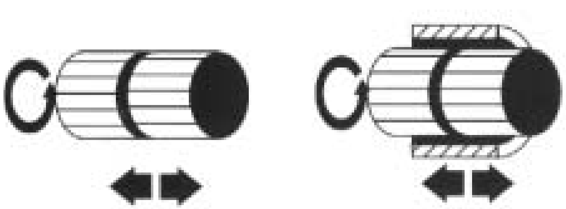
A peel joint can be designed such that the forces acting upon it become compression forces, making a much stronger joint.



Weak cleavage joints can be strengthened through design, in this instance by adding a U-section to the previously bent sheet.



By adding reinforcing plates to this butt joint, the forces run along a much stronger shear joint.



A similar effect is produced by sleeving this cylindrical butt joint.

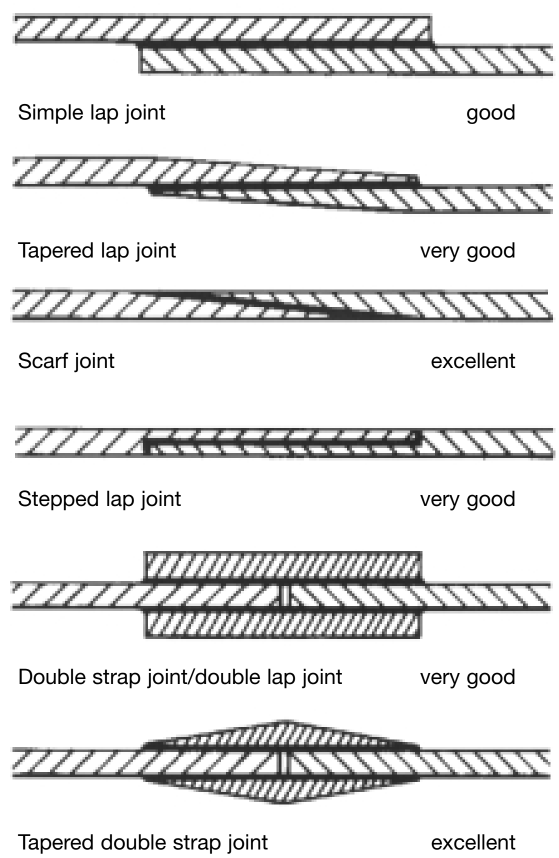


Fig.4 Basic bonded joints between strip/sheet metals

The basic types of bonded joints are shown diagramatically. In practical structures two or more basic types may be used in combination – and the relative dimensions (and areas of bonded surface) of the joints may vary from those shown in the diagrams.
Tapering of the ends of lap joints or scarf joints serves to distribute the stress more uniformly and reduce stress concentration.

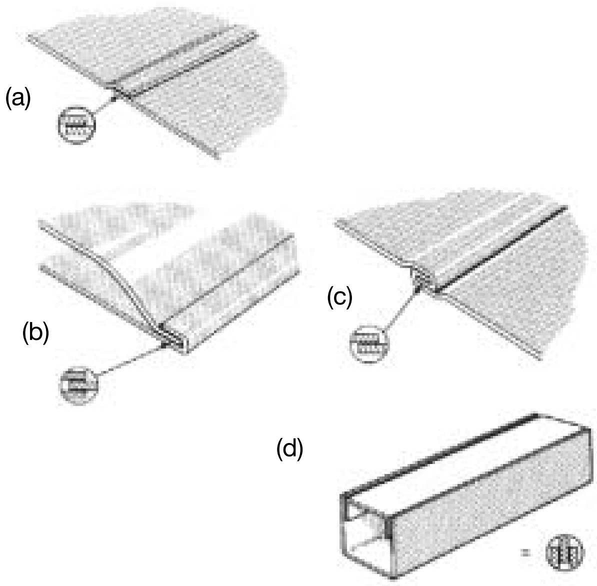


Fig.5 Practical bonded joints between sheet materials

Certain metals, especially mild steel, are easily bent or folded to form advantageous joints. (a) Shows a development from the simple lap joint: a toggled joint. (b) and (c) show further developments.
Closed box structures (d) from formed sheet metal are easily produced using this folding and bonding technique to join the edges.

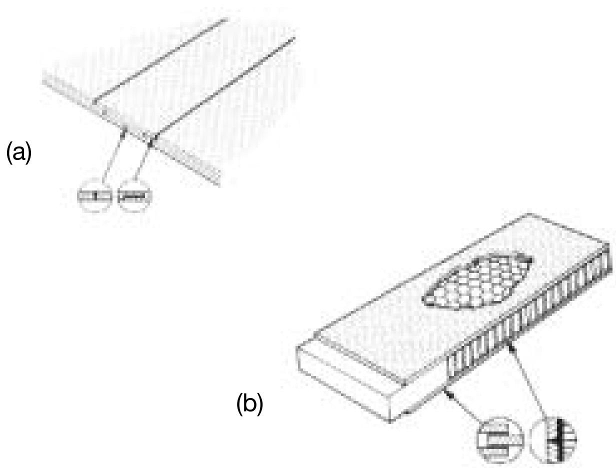


Fig.6 Bonding of multi-layer structures

Multi-layer structures may be built up by adhesive bonding and may also be bonded to other parts. In (a) a multi-layer fibre-reinforced plastics laminate is joined to its neighbour by a multi-stepped lap joint. In (b) an edge member is bonded into a sandwich panel. On loading, the stresses will be transferred into the panel. The honeycomb core is itself assembled and bonded to the facing sheets with adhesives.

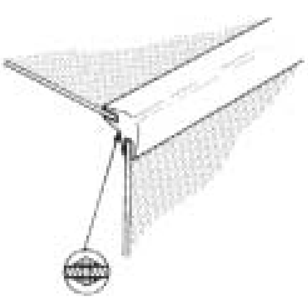


Fig.7 Joints using profiles

Sheets or plates that cannot be bent and folded may be bonded together by means of purpose-made profiles. Tapering removes the high stress concentrations caused by abrupt change in section.

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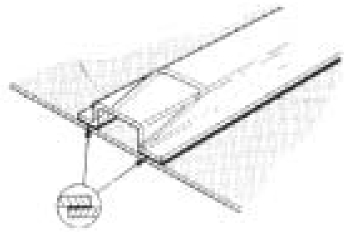
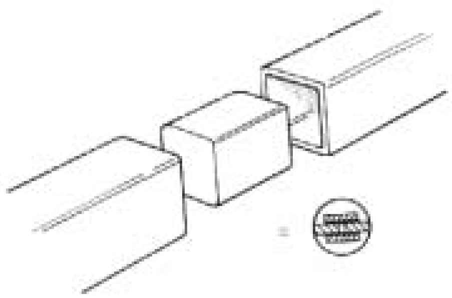


Fig.8 Stiffening of large thin sheets

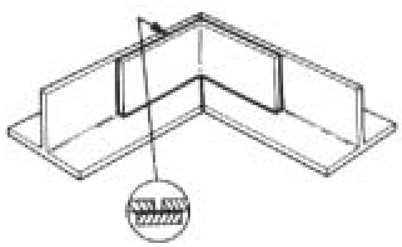
Large sheets of thin-gauge material (metal or plastics) may be stabilised by bonding stiffeners made of the same material in similar gauge. The diagram shows a ‘top hat’ stiffener.
Towards the edge of the sheet, the stiffener may be cut away (as shown) in order to reduce stress concentrations. The effect is similar to that of the scarf joint in Fig.4.



(a)

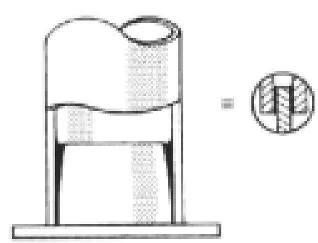
Fig.9 Bonded frameworks

Framework structures of square or round tubes, or simple profiles, may utilise plugs (a), angles (b), or bosses (c) at the joints. Use of these additional pieces greatly increases the area of bond surface at the joint.



(b)

(c)



The durability of a bonded joint

The durability (the long-term performance) of a bonded joint depends on the properties both of the adhesive and of the materials being joined.
The adhesive will be affected by high temperatures, by powerful solvents, or by water. The durability of the joint will also depend on the effects of these agents on the materials being joined. Above all, it will depend on the condition of the joint surfaces when the bond was made. The best joints are made when the surfaces are absolutely clean and have good affinity for the adhesive. This necessitates control of pretreatment of the surfaces. A poor surface condition usually results in a relatively low initial strength and a reduced durability. A thick bond-line gives lower initial strength. (See Figure 10.) With most types of adhesive, the application of heat to complete the curing process improves both initial strength and durability. The user will have to judge the level of control of these factors necessary to produce a bonded joint satisfactory for the expected service conditions. For many applications a good and sufficient durability is obtained with easily attained levels of surface control (or pretreatment), bond-line thickness and curing schedule.

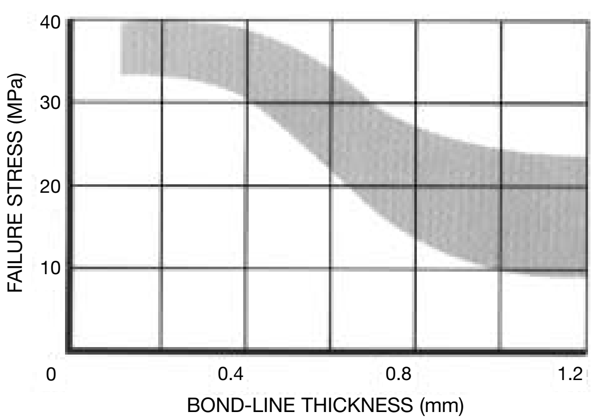


Fig.10 Bond-line thickness v. shear strength

Shear strength decreases if the layer of adhesive is thick. The effect of increasing bons-line thickness in simple lap joints made with hot-cured epoxy adhesives is shown in the diagram.
Adhesive strength at the interface is by its nature greater than the cohesive strength within the adhesive. The diagram shows that in this adhesive the drop in strength occurs in the range 0.4 to 1.0 mm. In thicknesses greater than 1.0 mm shear strength is approximately constant. The exact shape of the curve depends on the characteristics of the adhesive. Toughened adhesives will maintain higher values in thicker bondlines while more rigid adhesives will reduce more quickly. The optimum bond-line thickness is in the range 0.1 to 0.3mm. In very thin bond lines there is risk of incomplete filling of the joint due to contact between high points on the joint surfaces.

The bonded joints may need to resist sustained loads, which are either static or vibrational. Joint designs in which peel stresses are at a minimum give the best durability. The fatigue testing (by standard methods) of simple lap shear joints made with epoxy adhesives will often give failure values of ca 30% of the short-term measured breaking load. (See Figure 11.)

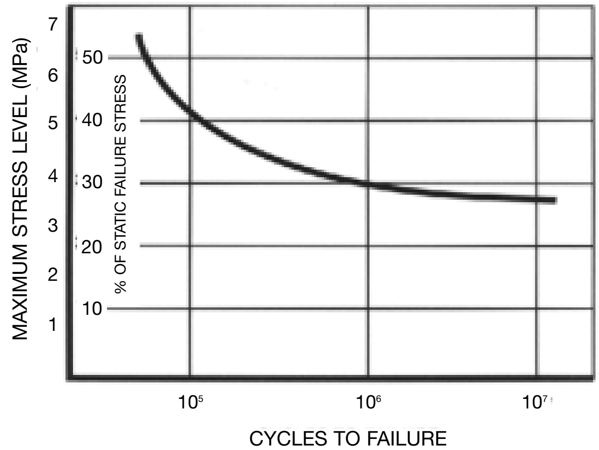


Fig.11 Fatigue strength (tensile) of lap joints

Fatigue strength of simple lap joints made with a cold-cured epoxy adhesive and tested to DIN 53 285. In this test programme the failure stress of control joints under static loading was 13 Mpa. The diagram shows that under fatigue loading the joints required to sustain 106 test cycles should not be stressed higher than 4.1 Mpa per cycle.

Determination of dimensions of simple lap joints

The shear strength of simple lap joint (Fig 12) depends on the nature of the metal, the adhesive, the thickness of the metal and the area of overlap.

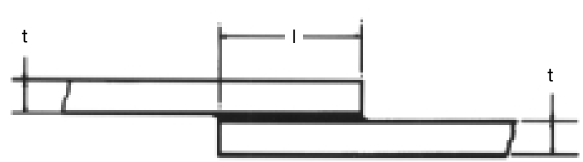


Fig.12 Simple lap shear joint

l = overlap; t = metal thickness

II

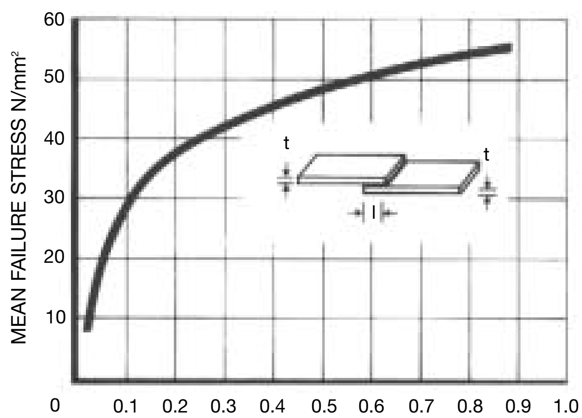


Fig.13 Correlation diagram between shear strength and t/l of simple lap joints

The diagram relates the dimensions of the joint, the shear stress in the adhesive and the tensile stress in the metal*

*The curve shown in Fig.13 was established from a test programme carried out on simple lap joints of BS 1470-HS30 aluminium alloy bonded with hot-cured Araldite epoxy adhesive.

Given the loading required and the metal and adhesive to be used, it is possible to predict:

1. Optimum overlap on metals of given thickness.
2. Optimum metal thickness for given overlap.

This overlap and thickness may be rapidly determined from a diagram based on results from one test programme.

The test – to determine mean shear strengths of joints of various overlaps (**l**) and metal thickness (**t**) – must be sufficient to plot a curve of shear strength against **t/l**. A curve established in this way is shown in Fig.13.

Any particular point on an established curve represents (for lap joints made with metal and adhesive to the same specifications as used in the test programme) the state of stress in a particular joint and shows the relationship between the dimensions of the joint (horizontal axis), the mean shear stress in the adhesive (vertical axis) and the mean tensile stress in the metal (slope of a straight line from the origin to the point).

Optimum overlap (**l**) is determined by using the diagram together with the formula:

$$\tau = \sigma . \frac{\mathbf{t}}{\mathbf{l}}$$

This formula is derived from –

The known design requirements:

P= load per unit width of joint

t = sheet thickness (**t**= thickness of thinner sheet in joints made of sheets of different thickness)

These establish:

σ = mean tensile stress in the metal = $\frac{\mathbf{P}}{\mathbf{t}}$

and by definition:

τ = mean shear stress in the joint = $\frac{\mathbf{P}}{\mathbf{l}}$

Substituting for **P** gives: $\tau = \sigma . \frac{\mathbf{t}}{\mathbf{l}}$

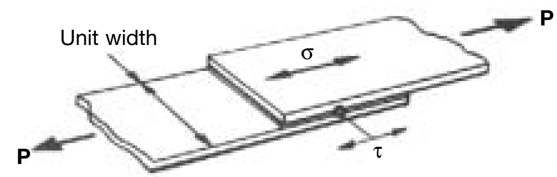


Fig.14 Conventional designs for stresses in a lap joint

Optimum overlap (**l**) is determined as follows:

1. Calculate s from **P** and **t**.
2. Starting form 0, mark on the diagram the straight line whose slope $\left(\tau / \frac{\mathbf{t}}{\mathbf{l}}\right)$ is given by σ.
3. Where the straight line cuts the curve, read off the value for τ
4. Having determined σ and τ, and knowing **t**, substitute these values in:

$$\tau = \sigma . \frac{\mathbf{t}}{\mathbf{l}}$$

and calculate optimum overlap **l**.

Deviation from the optimum overlap reduces the efficiency of the joint. Too small an overlap causes the joint to fail below the required loading, whereas too large an overlap may mean an unnecessarily large joint.

Optimum sheet thickness (**t**) is determined as follows:

1. Calculate τ from **P** and **l**.
2. Where the value of t cuts the curve, read off the value for $\frac{\mathbf{t}}{\mathbf{l}}$
3. Having determined $\frac{\mathbf{t}}{\mathbf{l}}$ and knowing **l**, calculate optimum thickness **t**.

II

Essentials for the bonding process

To make a successful bond, the adhesive must wet the material to be joined, fill the gap between the surfaces, and then fully harden.

With a two-part adhesive this means that resin and hardener must be correctly proportioned and thoroughly mixed together. The right amount of mixed adhesive needs to be placed and spread onto the bond area. Both these steps are aided by using automatic equipment. The simplest equipment dispenses adhesive from pre-filled cartridges (see Figure 15). Typical volumetric proportioning equipment, which meters, mixes and dispenses two-part epoxy adhesives is shown in Figure 16. Where highly viscous or thixotropic components are used, the metering units may be fed by special drum pumps. Similarly for one-part epoxy adhesives there are hand or air operated guns or applicators. Suitable equipment is advantageous in setting up a Quality Assurance Scheme for a bonding process.

Continuous production bonding also necessitates ensuring that the condition of the surfaces to be bonded is always the same. Unknown contaminants must be removed from the surfaces. A particular surface treatment may be needed in order to increase the affinity for the adhesive. Surface preparation can be a multi-step process. T usually includes mechanical abrasion and – to achieve optimum results – chemical etching.

In some cases known surface coverings, such as protective oils, may be absorbed by the adhesive in the bonding process – this ability is a characteristic of specially formulated oil-tolerant Araldite epoxy adhesives. In these cases the known covering material defines the surface condition.

The hardening or curing of reactive adhesives requires time. The time is shortened if heat can be applied. Furthermore, though with many two-part epoxy adhesives strong joints can be obtained by curing at room temperature (for 2 to 24 hours), higher curing temperatures – even a few degrees above room temperature – will raise the bond strength. With certain one-part epoxy adhesives curing temperatures may need to be as high as 180c in order to obtain the best properties. Elevated temperature curing may be carried out using:

Hot air ovens: This is a practical method only when a large number of assemblies are in the oven at the same time or for continuous production lines. Heat transfer is relatively slow and affected by the assembly type and thickness. Infa-red ovens can also be used.

Heated presses: Steam or oil-heated platens can be used in flat bed presses with a rapid and controllable temperature rise. This method is ideal for production of large flat panels, e.g. for insulated container sides.

Induction curing: Magnetic field causes current to flow in a conductive substrate. The resistance to the current generates heat and cures the adhesive. This technique has been used where very fast heat up and cure is required.

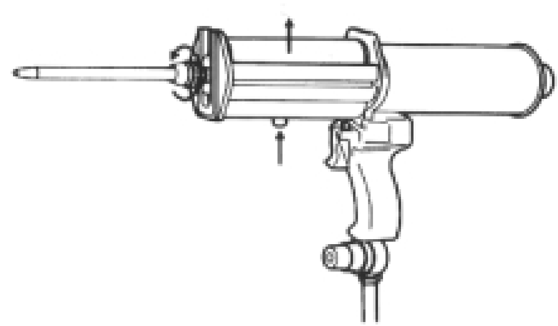


Fig.15 Handgun operated by compressed air

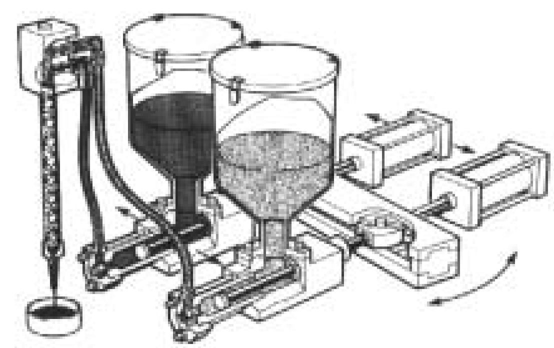


Fig.16 Metering and mixing machine for two-part epoxy adhesives

Combination joints

Adhesives can be used in combination with other joining methods, in particular, riveting or spot welding. Rivets or welds at intervals along the bond-line not only act as locating and holding points during the time the adhesive cures but also increases the peel resistance of the joint. From the other viewpoint, that of the mechanical fastening, the presence of the adhesive improves the stiffness of the joint, distributes the stresses uniformly and it forms a seal. Adhesive bonding also increases the speed and reduces overall the noise of the joining process.

II

Araldite adhesives adhere firmly to most materials. Bonds of great strength are obtained after removal of grease and loose surface deposits, e.g. rust, from the surfaces to be joined, but when maximum strength and long-term durability are required a more thorough mechanical or a chemical pretreatment is recommended.

Surface preparation

Surfaces are prepared by one of the following pretreatment procedures (listed in order of increasing effectiveness):

1. Degrease only.

2. Degrease, abrade and remove loose particles.

3. Degrease and chemically pretreat.

Care must be taken to avoid contaminating the surface during or after pretreatment. Contamination may be caused by finger marking – or by cloths, which are not perfectly clean – or by oil-contaminated abrasives – or by sub-standard degreasing or chemical solutions. Contamination may also be caused by other work processes taking place in the bonding area. Particularly to be excluded are oil vapours from machinery, spraying operation (paint, mould release-agent, etc.) and processes involving powdered materials.

Whatever the pretreatment procedure used, it is good practice to bond the surfaces as soon as possible after completion of the pretreatment – i.e. when surface properties are at their best.

If the scheduling of bonding operations on multi-part assemblies causes delay between pretreatment and bonding, optimum surface properties may be preserved by priming the bond surfaces immediately after pretreatment.

Degreasing

Remove all traces of oil and grease as follows:

(a) Suspend in halocarbon solvent* vapour in a vapour degreasing unit.

or

(b) Immerse successively in two tanks each containing the same liquid halocarbon solvent* acts as a wash, the second as a rinse.

*** Halocarbon solvents** At the time of publication, legislation regarding halogenated solvents was changing. Users should contact the solvent suppliers for advice and must ensure compliance with local and national regulations governing their use.

or

(c) Brush or wipe the joint surfaces with a clean brush or cloth soaked in clean proprietary commercial degreasing solvent. A wide range of proprietary solvent degreasing agents with low hazard ratings are now available.

or

(d) Detergent degreasing. Scrub the joint surface in a solution of liquid detergent. Wash with clean hot water and allow to dry thoroughly – preferably in a stream of hot air.

or

(e) Alkaline degreasing is an alternative method to the detergent degreasing. It is recommended to use proprietary products and follow manufacturer's instructions for use.

or

(f) Ultrasonic degreasing may be employed when appropriate and is generally used for the preparation of small specimens.

Abrading

Lightly abraded surfaces give a better key to adhesives than do highly polished surfaces. Abrasion treatment, if carried out, must be followed by a further treatment to ensure complete removal of loose particles. For example:

(a) Repeat the degreasing operation (degreasing liquids must be clean),

or

(b) Lightly brush with a clean soft brush, or – preferably

(c) Blow with a clean dry (filtered) compressed air-blast. Abrasion can be carried out with abrasive paper, wire brushing or most effectively by grit-blasting.

Pretreatments for particular materials

Most materials likely to require bonding in industrial practice are dealt with individually in detail in Publication No.15 – Guide to surface preparation and pretreatment. The information in this publication is intended only as an overview.

Special pretreatments for maximum bond performance

The surface preparation described above, i.e. degreasing alone or degreasing followed by abrasion and removal of loose particles, is sufficient for most adhesive work, but to obtain maximum strength, reproducibility and long-term resistance to deterioration, a chemical or electrolytic pretreatment may be required.

Metal adherent surfaces are rarely of pure metal, but are a combination of oxides, sulphides, chlorides and other atmospheric contaminants resulting in a surface which is mechanically weak. Acid etching is a well-established method of removing metallic scale, in favour of forming an oxide layer, which is mechanically and chemically compatible with the adhesive. Hence, different acid treatments are applied to different metal adherends, for example, chromic acid for aluminium, sulphuric acid for stainless steel, and nitric acid for copper. Acid pretreatment can also be applied to certain plastics, e.g. chromic acid is used to surface treat polyolefins. (Details are given in Publication No.A15.)

Anodising has been exploited extensively by the aerospace industry as a surface pretreatment for aluminium and titanium alloys. The purpose of anodising is to deposit a porous oxide layer on top of the oxide layer formed after etching. The porous oxide layer enables adhesive (or primer) to penetrate the pores readily to form a strong bond. 'Hard' anodising is not an effective bonding pretreatment.

Application of a primer is another form of surface pretreatment mainly used for materials such as metals and ceramics. Generally, the primer is the final stage of a multistage pretreatment process. Some adherends have 'difficult to bond' surfaces (e.g. copper). The primer, which is formulated such that it represents a solvented version of the adhesive, readily wets the adherend. The adhesive, when applied to the primed surface, being chemically compatible, will establish a strong joint on curing.

Essentials for chemical pretreatments

Care must be taken in the preparation of chemical pretreatment solutions, not only because of the handling hazards, but also because incorrect preparation may lead to bond strengths inferior to those that would have been obtained if there had been no chemical pretreatment.

Time of application is also critical: too short an application does not sufficiently activate the surfaces, whereas overlong application may build up chemical reaction products, which interfere with adhesion.

On completion of chemical pretreatment, thorough washing of the surfaces with plenty of clean water is standard practice. For the final rinse, the use of deionised (demineralised) water is recommended.

Surfaces should be bonded as soon as possible after pretreatment. Stability of the pretreated surfaces is limited.

Metals

The wide range of individual alloy (and the variety of surface structures caused by different heat treatments) within each metal group precludes standardising on one pretreatment for each. The pretreatments listed in Publication No.A15 are well established but on occasions a different pretreatment may prove more effective. This can be shown only by comparative trials – using material from the batch of metal components to be bonded and the type of adhesive specified for the work. Additional data on pretreatment of metals is given in ISO 4588 and DEF standard 03-2/2.

Thermosetting plastics

Mouldings, castings, laminates, etc. can usually be bonded without difficulty. To ensure good bond strength, all soil and residual release agents must be removed from the joint surfaces before the adhesive is applied. The surface must either be abraded with emery cloth or grit-blasted, or they must be cleaned with a solvent such as acetone, methyl ethyl ketone, etc. Abrading or grit-blasting is recommended for mouldings since their surfaces may otherwise repel the adhesive.

Thermoplastics

These are often difficult to bond. Certain types permit only moderately successful bonding, and one and the same material may show considerable variation in properties, which determine the strength of a bond. Special adhesives have been developed, but they usually prove to be unserviceable when thermoplastics have to be bonded to materials such as wood, metal, etc. Araldite adhesives can be very useful in such cases even though their suitability for bonding thermoplastics is only limited. Pretreated thermoplastics for special applications (e.g. ski 'skins') are easily bonded with Araldite.

The grade of plastic and the manufacturing process used to make the component may influence the effectiveness of the pretreatment. It is advisable to establish by trial whether the pretreatment is improved by adjusting the specified time.

In addition to the normal mechanical and chemical methods of pretreatment, certain plastics can be pretreated using the following methods, all of which cause a change in the surface texture of the adherend. The change is brought about by the interaction of highly energised species with the adherend surface. These pretreatment methods have been applied to metals and in particular composites and plastics.

A low pressure plasma is an excited gas generated by applying a high frequency and high voltage between electrodes in a low pressure chamber. The advantage of this method is that it allows treatment of adherends by different plasmas of argon, ammonia, oxygen or nitrogen making the process suitable for a range of substrate types. Plasmas are generally used to activate the surfaces of adherends.

If instead a plasma is created in air at atmospheric pressure, the air when isolated appears as a blue/purple glow with faint sparking, and is termed a **corona**. Corona treatments are usually applied for preparing thin polymer films and composite laminates.

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The effect of a **flame treatment** is to oxidise the adherend, which produces polar groups creating a surface better suited to wetting by the adhesive. This method of surface pretreatment has been applied successfully to polyethylene/polypropylene. The variables of flame treatment include type of gas, gas/air (oxygen) ratio, the rate of flow of mixture, exposure time and distance between flame and adherend.

All these methods have limited stability due to adsorption of airborne contaminants and vary from hours to weeks according to substrate. Further information can be found in ISO 13895.

Araldite adhesives are simple to use, but to ensure successful bonding the directions given in the instructions supplied with the adhesive must be strictly observed.

In particular:

1. Joint surfaces must be degreased and when necessary, pretreated.
2. Resin and hardener must be correctly proportioned and thoroughly mixed together.
3. Adhesive must be applied in the correct controlled thickness.
4. Jigs or other fixtures must be used to prevent the bond surfaces from moving relative to one another during the curing process.
5. Though only light pressure is needed, it should be applied as evenly as possible over the whole bond area. Excessive pressure leaves the joint starved of adhesive.
6. Curing temperatures and curing time must be correct (in accordance with the supplier's recommendations).

Caution

Acids, caustic soda etc.

Concentrated acids, oxidising agents (e.g. chromium trioxide, dichromates) and caustic soda are highly corrosive chemicals. Spillages and splashes can cause severe damage to eyes and skin, and attack ordinary clothing where these chemicals are used.

The manufacturer's handling precautions must be observed.

Araldite, Epocast, Epibond and Uralane

Araldite, Epocast, Epibond and Uralane resins and hardeners are generally quite harmless to handle provided that certain precautions normally taken when handling chemicals are observed. The uncured materials must not, for instance, be allowed to come into contact with foodstuffs or food utensils, and measures should also be taken to prevent the uncured materials from coming in contact with the skin, since people with particularly sensitive skin may be affected. The wearing of impervious rubber or plastic gloves will normally be necessary; likewise the use of eye protection. The skin should be thoroughly cleansed at the end of each working period by washing with soap and warm water. The use of solvents is to be avoided. Disposable paper towels – not cloth towels – should be used to dry the skin. Adequate ventilation of the working area is recommended. These precautions are described in greater detail in Publication No.24264* and in the Safety Data Sheets* for the individual products.

The Araldite UV range is likewise generally harmless to handle provided that direct contact with the adhesive is avoided and good ventilation is maintained.

* These publications are available on request and should be referred to.

All recommendations for use of our products, whether given by us in writing, verbally, or to be implied from the results of test carried out by us, are based on the current state of our knowledge. Notwithstanding any such recommendations the Buyer shall remain responsible for satisfying himself that the products as supplied by us are suitable for his intended process or purpose. Since we cannot control the application, use or processing of the products, we cannot accept responsibility therefor. The Buyer shall ensure that the intended use of the products will not infringe any third party's intellectual rights. We warrant that our products are free from defects in accordance with and subject to our general conditions of supply.

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Araldite® Bonding

II

Surface preparation and pretreatments

Working directions for the surface preparation essential for optimum adhesion between structural materials bonded with Araldite adhesives.

Araldite adhesives form extremely strong and durable bonds with metals, glass, rigid plastics, rubber and many other materials. Designers in every sphere of industry increasingly find that bonding with Araldite provides the answer to production problems posed by new materials, new uses of existing materials, and new techniques and manufacturing methods.

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Introduction

Araldite resins adhere firmly to most materials. Bonds of great strength are obtained after removal of grease and loose surface deposits, e.g. rust, from the surfaces to be joined, but when maximum strength is required a more thorough mechanical or a chemical pretreatment is recommended.

Materials to be bonded
Listed on the back cover of this manual are the materials for which pretreatments are given. Bonding these materials comprises the main part of high-strength adhesive work and consequently their pretreatments are the most in demand. Materials less commonly used and not specifically dealt with in this manual may require only simple degreasing and abrading (as described below) but if other pretreatments appear necessary advice should be obtained from Huntsman Advanced Materials.

Surface preparation
Surfaces are prepared by one of the following pretreatment procedures (listed in order of increasing effectiveness).

1. Degrease only.
2. Degrease, abrade and remove loose particles.
3. Degrease and chemically pretreat

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Care must be taken to avoid contaminating the surfaces during or after pretreatment. Wear clean gloves. Contamination may be caused by finger marking – or by cloths which are not perfectly clean – or by oil – contaminated abrasives – or by sub-standard degreasing or chemical solutions. Contamination may also be caused by other work processes taking place in the bonding area. Particularly to be excluded are oil vapours from machinery, spraying operations (paint, mould release-agent, etc.) and processes involving powdered materials.

This instruction manual is divided into the following parts:

Whatever the pretreatment procedure used, it is good practice to bond the surfaces as soon as possible after completion of the pretreatment – i.e. when surface properties are at their best.

Note If the scheduling of bonding operations on multi-part assemblies causes delay between pretreatment and bonding, optimum surface properties may be preserved by priming the bond surfaces immediately after pretreatment.

Part 1 - Degreasing

The removal of all traces of oil and grease from the surfaces to be bonded is essential. Degreasing by one of the four methods given below should be carried out even when the surfaces to be bonded appear clean.

Degreasing methods
Remove all traces of oil and grease as follows:

(a) Suspend in halocarbon solvent *vapour in a vapour degreasing unit. The unit may include a compartment to enable initial washing in the liquid solvent.

or

where a vapour degreasing unit is not available:

(b) Immerse successively in two tanks each containing the same liquid halocarbon solvent. The first tank acts as a wash, the second as a rinse. When the solvent in the wash tank becomes heavily contaminated, the tank is cleaned out and refilled with fresh solvent.

***Halocarbon solvents** Trichloroethylene is the dominant halocarbon solvent for vapour degreasing. Its toxicity necessitates the use of suitably designed plant. At the time of publication, legislation regarding halogenated solvents was changing. Users should contact the solvent suppliers for advice and must ensure compliance with local and national regulations governing their use. See Part 5 – Caution.

This tank is then used for the rinse, and the former tank for the wash.

or

(c) Brush or wipe the joint surfaces with a clean brush or cloth soaked in clean halocarbon solvent or other proprietary commercial degreasing solvent. (For fine work, washing down with solvent applied by aerosol spray may be a more suitable alternative; this technique also ensures that the solvent used is clean.) Allow to stand for a minute or two to permit complete evaporation from the joint surfaces.
A wide range of proprietary solvent degreasing agents with low hazard ratings are now available. These should be used according to the manufacturers' instructions

Note Certain plastics and rubbers are attacked by solvents such as trichloroethylene. These plastics may be degreased with detergent solutions, alcohols such as isopropanol, ketone solvents, or proprietary solvent degreasing agents, depending on the type of plastic.

or

(d) **Detergent degreasing** Scrub the joint surfaces in a solution of liquid detergent. Wash with clean hot water and allow to dry thoroughly – preferably in a stream of hot air from, e.g. a domestic forced-air heater.

Note Non-ionic detergents give generally good results.

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(e) **Alkaline degreasing** is an alternative method to the detergent degreasing.
The ingredients may be selected from a wide range of compounds including sodium or potassium hydroxide, carbonates, phosphates, borates, complexing agents and organic surfactants. They can be used hot or cold with or without applied current. There should be very thorough washing, and possibly neutralisation to remove residual traces of alkaline cleaners. It is recommended to use proprietary products and follow manufacturers' instructions for use.

(f) **Ultrasonic degreasing** may be employed when appropriate and is generally used for the preparation of small specimens.

Warning Safety precautions must be observed where halocarbon solvents are in use. See Part 5 – Caution.

Test for a clean bond surface

The water-break test is a simple method to determine whether the surface to be bonded is clean. It is best suited to metals. If a few drops of distilled water applied to the surface, wet the surface and spread – or if, on drawing the surface from distilled water, the water film does not break up into droplets – then the surface may be assumed to be free of contamination. Uniform wetting of the surface by distilled water indicates that it will probably be likewise wetted by adhesive.
It must be borne in mind that certain plastics, even when clean, may not be wetted by distilled water, but will be wetted by adhesive. Furthermore, satisfactory wetting gives no information as to the potential bond strength. At most it is a necessary – but not sufficient – requirement for the achievement of high bond strengths.

Part 2 - Abrading

Lightly abraded surfaces give a better key to adhesives than do highly polished surfaces. Abrasion treatment, if carried out, must be followed by a further treatment to ensure complete removal of loose particles. For example:
(a) Repeat the degreasing operation (degreasing liquids must be clean), or
(b) Lightly brush with a clean soft brush, or-preferably
(c) Blow with a clean dry (filtered) compressed-air blast.

Metal surfaces

Remove surface deposits, e.g. tarnish, rust or mill scale, preferably by blasting with sharp grit*. If grit-blasting equipment is not available or the metal is too thin to withstand blast treatment, then clean the joint surfaces with a wire brush, or with abrasive cloth or water-proof abrasive paper (alumina or silicon carbide abrasive, from 46 to 120 mesh). Wetting the wire brush – or the abrasive cloth or paper – assists removal of contaminants and reduces dust. Dry, if necessary, and remove all loose particles.

Note Painted surfaces should be stripped of paint; otherwise the strength of the joint may be limited by comparatively low adhesion to metal.

*For most materials the preferred grits are fused alumina and, less commonly, silicon carbide (ferrous grits such as chilled iron must be restricted to mild steels and cast irons; their use on other metals may promote corrosion). Fused alumina is the abrasive almost invariably used for aluminium alloys and stainless steels. Silicon carbide is sharper, but it is more expensive and also more friable. Silicon carbide is used on certain special alloys liable to react adversely with any residual fused alumina at temperatures they may encounter in service. The use of silicon carbide can be advantageous when the materials to be abraded are either soft or extremely hard. Choice of grit size depends on various factors: the metal to be grit-blasted, the type of grit-blasting equipment, the pressure and angle of blast impact, and the time of treatment. Grits in the range of 46 to 120 mesh are suitable, but the optimum grit size for the work in hand can be determined only by trials. In general for soft materials the optimum grit size will be towards the fine end of the range.

Plastics and glass surfaces
Remove the surface layer of plastics surfaces to ensure elimination of all traces of release agent. As with metals, abrasion by grit-blasting (see notes on page 3) is in general the best method; the alternative is to use abrasive cloth or paper. After abrasion, remove all loose particles.

Note Removal of loose particles from plastics surfaces is best carried out by methods (b) or (c) above. Use of degreasing liquids on certain plastics may impair the key produced by the abrasion treatment.

Since plastics are poor heat conductors, care must be taken to keep blasting times as short as possible.

For pretreatment of composite materials cryoblasting may also be used which involves use of solid carbon dioxide pellets as the blasting medium.

Part 3 - Pretreatments for particular materials

Most materials likely to require bonding in industrial practice are dealt with individually in the following pages – for index, see page 20. Engineers contemplating the bonding of materials not covered by this manual are invited to submit enquiries concerning appropriate pretreatments to our technical staff.

Special pretreatments for maximum bond performance
The surface preparation described above, i.e. degreasing alone or degreasing followed by abrasion and removal of loose particles, is sufficient for most adhesive work.

But to obtain maximum strength, reproducibility and long-term resistance to deterioration, a chemical or electrolytic pretreatment may be required – and examples of these special pretreatments are printed in blue in the following pages.

Metal adherend surfaces are rarely of pure metal, but are a combination of oxides, suphides, chlorides and other atmospheric contaminants resulting in a surface which is mechanically weak. Acid etching is a well-established method of removing metallic scale, in favour of forming an oxide layer which is mechanically and chemically compatible with the adhesive. Hence, different acid treatments are applied to different metal adherends, for example, chromic acid for aluminium, sulphuric acid for stainless steel, and nitric acid for copper. Acid pretreatment can also be applied to certain plastics, e.g. chromic acid is used to surface treat polyolefins.

Anodising has been exploited extensively by the aerospace industry as a surface pretreatment for aluminium and titanium alloys. The purpose of anodising is to deposit a porous oxide layer on top of the oxide layer formed after etching. The porous oxide layer enables adhesive (or primer) to penetrate the pores readily to form a strong bond.

Application of a primer is another form of surface pretreatment mainly used for materials such as metals and ceramics. Generally, the primer is the final stage of a multistage pretreatment process. Some adherends have 'difficult to bond' surfaces (e.g. copper). The primer, which is formulated such that it represents a solvented version of the adhesive, readily wets the adherend. The adhesive, when applied to the primed surface, being chemically compatible, will establish a strong joint on curing.

Essentials for chemical pretreatments
Care must be taken in the preparation of chemical pretreatment solutions, not only because of the handling hazards*, but also because incorrect preparation may lead to bond strengths inferior to those that would have been obtained if there had been no chemical treatment.
Time of application is also critical: too short an application does not sufficiently activate the surfaces, whereas overlong application may build up chemical reaction products which interfere with adhesion.
On completion of a chemical pretreatment, thorough washing of the surfaces with plenty of clean water is standard practice. For the final rinse, the use of deionised (demineralised) water is recommended.

*Safety precautions must be strictly observed where chemical solutions are in use. See Part 5 – Caution.

Metals

The wide range of individual alloys (and the variety of surface structures caused by heat treatments) within each metal group precludes standardising on one pretreatment for each. The following pretreatments are well established but on occasion a different petreatment (not given here) may prove more effective. This can be shown only by comparative trials – using materials from the batch of metal components to be bonded and the type of Araldite adhesive specified for the work.
Additional data on pretreatment of metals is given in ISO 4588 and DEF standard 03-2/2. The recommendations given in this brochure for pretreatment of metals are in compliance with the above.

Aluminium and aluminium alloys

Anodised material Aluminium alloy anodised by the normal chromic acid or sulphuric acid methods and sealed, may be bonded after degreasing and light abrasion. Chromic acid or phosphoric acid anodised material has the optimum surface properties for bonding directly after completion of the anodising process. No pretreatment is needed, but the material must be bonded within a few hours anodising.
Hard anodised aluminium alloy requires stripping either by abrasive blasting or by etching in the sulphuric acid + sodium dichromate (or chromiun trioxide) solution given below. Note: the unstripped metal is unsuitable for bonding.

Non-anodised material Degrease according to Part 1 – Degreasing. Then either abrade according to Part 2 – Abrading.

Or anodise with chromic or phosphoric acid as above

Or etch in a sulphuric acid + sodium dichromate solution made up as follows:

Pour 10-15 litres of clean water into a container with a 50 litre calibration mark. While stirring the water, add 7.5 litres of concentrated sulphuric acid (S.G. 1.84) in a slow steady stream – with continued stirring add 3.75kg of sodium dichromate ($Na_2Cr_2O_7.2H_2O$) – or 2.5kg of chromium trioxide (CrO_3)* – and fill to the 50 litre calibration mark with clean cold water. For best results the aluminium content of the etch should be 1.5-14.0g/l and the copper content 0.06-1.0g/l. (see notes below).**
Immerse the part for 30 minutes at 60-65^{0}C, then wash with clean cold running water, followed by clean hot water, and dry with hot air. The temperature of the hot water and air must not be greater than 65^{0}C.

***This procedure + preliminary degreasing with halocarbon solvent conforms to Process O of DEF Standard 03-2/2. Method O includes a procedure for controlling the composition of etch solution in continual use.**

****European standard EN2334 describes in detail acid chromate pickling procedures and the control of solutions. Similarly DT9915B and DEF STAN 03-2/2 describe control procedures.**

ISO 4588 describes and equally suitable etching process based on a solution of 30 pbw water, 10 pbw concentrated sulphuric acid and 2 pbw sodium dichromate. The parts are immersed for 10 minutes at 65-70^{0}C and then rinsed and dried as above.

> **Warning** Chromium trioxide is an exceptionally powerful oxidising agent. Particular care is essential when handling this chemical. Likewise, particular care is needed when handling concentrated sulphuric acid. – see Part 5 – Caution.

Accomet C*, a solution containing chromium compounds, is an alternative to the chemical pretreatment for aluminium and aluminium alloys. Accomet C in diluted form is applied as a primer coating. Instructions for use are given on page 9 under Primer for Metals.
***British Patent 1,234,181 (Albright & Wilson Ltd).**

Cadmium
Degrease according to Part 1 – Degreasing (page 2). Then either abrade according to Part 2 – Abrading (page 3), **or electro-plate with silver or nickel.**

Cast iron
Degrease according to Part 1 – Degreasing (page 2). Then abrade according to Part 2 – Abrading (page 3).

Chromium
Degrease according to Part 1 – Degreasing (page 2). Then either abrade according to Part 2 – Abrading (page 3), **or etch in a solution of:**

Concentrated hydrochloric acid (S.G. ca 1.18)	**4.25 litres**
Water	**5 litres**

Immerse for 1-5 minutes at 90-95^{0}C, wash with clean cold running water, followed by clean hot water, and dry with hot air.

Note Prepare the solution according to the sequence specified on page 14 under Concrete – method 3.

> **Warning** Concentrated hydrochloric acid is highly corrosive. Special care is required. See Part 5 – Caution.

II

Copper and copper alloys
Degrease according to Part 1-Degreasing (page 2). Then either abrade according to Part 2-Abrading (page 3),
or etch for 30 seconds at room temperature in a solution of:

Concentrated nitric acid (S.G. ca 1.42)	**5 litres**
Water	**15 litres**

Wash with clean cold running water. Do not allow to dry. Immerse for 2-3 minutes at 95-100ºC in a solution of:

Sodium hydroxide	**0.1kg**
Sodium chlorite ($NaC10_2$ technical)	**0.6kg**
Trisodium phosphate (Na_3 P_4O anhydrous)	**0.2kg**
Water	**20 litres**

Wash with plenty of clean cold water and dry promptly with a room temperature air stream. (The use of *hot* air may cause staining of the surfaces.)

The above two-stage chemical pretreatment gives, in general, better bond strengths than the ammonium persulphate pretreatment below. This however offers the advantage of simplicity and the strengths obtained may be adequate for the work in hand.

II

Etch in a 25% solution of: Ammonium persulphate
Immerse for 30 seconds at room temperature, wash with plenty of clean cold water and dry promptly with a room temperature air stream. (The use of hot air may cause staining of the surfaces.)
Note Preparation of 25% ammonium persulphate solution: pour about 700ml of deionised water into a container with a 1,000ml calibration mark. Add 250 grammes of ammounium persulphate. Stir until the powder dissolves, then fill to the 1,000ml calibration mark with deionised water.

Warning Concentrated nitric acid is highly corrosive. Special care is required when handling. See Part 5 – Caution.

Araldite Primer DZ 80-1, a hot-setting one-part resin solution, is an alternative to the chemical pretreatments for copper and copper alloys. The use of Primer DZ 80-1 improves bond strength retention during long-term service. Drying and curing schedules for Primer DZ 80-1 are given in the Huntsman Advanced Materials Instruction Sheet No. A.27, available on request.

Galvanised surfaces
Pretreat as for Zinc and Zinc Alloys (page 9).

Gold
Degrease according to Part 1-Degreasing (page 2).

Lead
Degrease according to Part1-Degreasing (page 2). Then either abrade according to Part 2-Abrading (page 3),
or etch in a solution of:

Concentrated nitric acid (S.G. ca 1.42)	**1 litre**
Water	**9 litres**

Immerse for 10 minutes at 45-55ºC, wash with clean running water, followed by clean hot water, and dry with hot air.

Warning Concentrated nitric acid is highly corrosive. Special care is required when handling. See Part 5-Caution.

Magnesium and Magnesium alloys
Degrease according to Part 1-Degreasing (page 2). Then abrade according to Part 2-Abrading (page 3), and apply the adhesive immediately.
The following chemical pretreatment for magnesium alloys produces bond strengths only slightly below those obtainable on the grit-blasted metal. The chemical pretreatment provides an alternative to grit-blasting for metal too thin to withstand the grit-blasting treatment and it is more effective than cleaning by wire brush or abrasive paper.

Degrease according to Part 1-Degreasing (page 2). Then etch in a solution of

Chromium trioxide (CrO_3)	**24 gms**
Water	**123 gms**
Sodium Sulphate (anhydrous)	**1.8 gms**
Calcium Nitrate	**2.1 gms**

Immerse for 3 minutes at room temperature, wash with clean cold running water, followed by clean hot water, dry with hot air and apply the adhesive immediately.

Warning Chromium trioxide is an exceptionally powerful oxidising agent. Particular care is essential when handling this chemical. See Part 5-Caution.

Nickel and nickel alloys

Degrease according to Part 1-Degreasing (page 2). Then either abrade according to Part 2-Abrading (page 3), **or etch for 5 seconds in: Concentrated nitric acid (S.G. ca 1.42). Wash with clean cold running water, followed by clean hot water, and dry with hot air.**

II

Warning Concentrated nitric acid is highly corrosive. Special care is required when handling. See Part 5-Caution.

Silver

Degrease according to Part 1-Degreasing (page 2). Then abrade according to Part 2-Abrading (page 3).

Steel-mild

Degrease according to Part 1-Degreasing (page 2). Then either abrade according to Part 2-Abrading (page 3), **or etch in a solution of:**

Orthophosphoric acid (S.G. ca 1.7)	**10 litres**
Industrial methylated spirit	**20 litres**

Immerse for 10 minutes at 60ºC, remove from the solution and then, under clean cold running water, brush off the black deposit with a stiff-bristle nylon brush. Absorb residual water by wiping with a clean cloth soaked with clean industrial methylated spirit or isopropanol. Heat for 1 hour at 120ºC.

Accomet C*, a solution containing chromium compounds, is an alternative to the chemical pretreatment for mild steel. Accomet C in diluted form is applied as a primer coating. Instructions for use are given on page 9 under Primer for Metals.

Warning Orthophosphoric acid is corrosive and requires special care in use. Refer to Part 5 - Caution.

Steel-stainless

Degrease according to Part 1-Degreasing (page 2). Then either abrade according to Part 2-Abrading (page 3), **or etch for 5-10 minutes at 55-65ºC in a solution of:**

Oxalic acid ($(COOH_2)2H_20$)	**5kg**
Concentrated sulphuric acid (S.G. ca 1.83)	**16 litres**
Water	**35 litres**

Note: Prepare solution according to the sequence specified on page 5 under Aluminium and Aluminium Alloys. The oxalic acid will dissolve completely at the immersion temperature.

Prior conditioning (e.g. passivation) of the steel surface may delay the reaction between steel and etch solution. The etch treatment should be timed from the onset of the reaction.

Wash with clean cold running water, then remove the black deposit* by immersing for 5-20 minutes at 60-65ºC in the sulphuric acid + sodium dichromate (or chromium trioxide) etch specified on page 5 under Aluminium and Aluminium Alloys.

Note Trials are recommended with the particular stainless steel to establish the optimum immersion conditions and proportions of the solution constituents. Baths in use for the pretreatment of aluminium alloys must not be used concurrently for the pretreatment of steel.

*Alternatively, remove the black deposit by brushing, under clean *cold* running water, with a stiff-bristle nylon brush, and dry with hot air. Highest bond strengths, however, are obtained after desmutting by the chemical treatment given above.

Warning Concentrated sulphuric acid and chromic acid are highly corrosive. Special care is needed when handling these chemicals. See Part 5-Caution

Accomet C*, a solution containing chromium compounds, is an alternative to the chemical pretreatments for stainless steels. Accomet C in diluted form is applied as a primer coating.
Instructions for use are given on page 9 under Primer for Metals.

Titanium and titanium alloys

Degrease according to Part 1-Degreasing (page 2). Then either abrade according to Part 2-Abrading (page 3),

or etch for 1-2 minutes at room temperature in a solution* of:

Concentrated nitric acid (S.G. ca 1.42)	**9.5 litres**
Hydroflouric acid (S.G. ca 1.17)	**0.85 litres**
Water	**40 litres**

Wash with clean cold running water, then immerse for 2-3 minutes at room temperature in a solution* of:

Trisodium phosphate ($NA_3\ P0_4\ .12H_20$)	**1.75kg**
Potassium flouride ($KF.2H_20$)	**0.68kg**
Hydroflouric acid (S.G. ca 1.17)	**1 litre**
Water	**40 litres**

Wash with clean cold running water, immerse in clean deionised water† at 55-65°C for 15-20 minutes, remove, wash with clean cold running water (brush off any remaining deposit with a clean stiff-bristle nylon brush) and dry with hot air. The temperature of the hot water and air must not be greater than 65°C.

† Frequent renewing of the deionised water is recommended. Renewing is essential if turbidity appears.

The above chemical pretreatment is used mainly in the bonding of titanium alloy structures for aircraft.
Likewise used for this purpose is:

Etch for 10-20 minutes at room temperature in a solution* of:

Concentrated nitric acid (S.G ca 1.42)	**4.5 litres**
Hydroflouric acid (S.G ca 1.17)	**0.45 litres**
Water	**10 litres**

Wash with clean cold running water (brush off any deposit with a clean stiff-bristle nylon brush), then anodise to give a blue surface film:

Solution: chromium trioxide CrO (60-80 grammes per litre of deionised water). Anode: titanium alloy part to be bonded. Cathode: mild steel (for example). Anode-to-cathode area ratio: 3:1. Potential: increase at 4 volts/minute to 20 volts and maintain for 5-30 minutes, depending on the particular alloy type. Temperature of solution: 38-40°C.

Wash with clean cold running water, followed by clean hot water, and dry with hot air. The temperature of the hot water must not be greater than 65°C.

* Use a polythene or polypropylene container. Mixing procedure: add the acids to the water in a slow and steady stream with continuous stirring.

> **Warning** Chromium trioxide is an exceptionally powerful oxidising agent. Hydroflouric acid and nitric acid are highly corrosive. Particular care is essential when handling these chemicals. See Part 5-Caution.

Tungsten and tungsten carbide

Degrease according to Part 1-Degreasing (page 2). Then either abrade according to Part 2-Abrading (page 3),
or etch in a solution* of:

Caustic soda (sodium hydroxide)	**15 kg**
Water	**35 litres**

Immerse for 10 minutes at 80-90°C, wash with clean cold running water, followed by clean hot water, and dry with hot air.

* Use a stress-relieved mild-steel container.) Aluminium, tin and zinc-coated, galvanised or tinned ware are unsuitable for caustic soda.) Mixing procedure: slowly sprinkle while stirring, flake or pearl caustic soda onto the cold water. Continue stirring until the soda is dissolved.

Zinc and zinc alloys

Degrease according to Part 1 – Degreasing (page 2). Then either abrade according to Part 2 – Abrading (page 3), and apply the adhesive immediately.

Or treat with Bonderite 255 solution according to the supplier's directions:

Degrease, wash with clean cold water, dip (or spray) for 30-45 seconds in Bonderite 255 solution, wash with clean cold water, and dry with hot air.

Primer for metals – Accomet C

The proprietary product Accomet C is highly effective in diluted form as a primer for the bonding of steels and aluminium alloys with Araldite adhesives. The metal surfaces are prepared as follows:

II

Surface preparation Degrease according to Part 1 – Degreasing (page 2). The best surface condition for wetting out with Accomet C is obtained by a final degreasing with alkaline cleaner or detergent. See Part 1 – Degreasing (page 3, section d).

Application Apply a thin coating of diluted Accomet C solution (by brushing, or by dipping and draining) to both the surfaces to be bonded. Dry in hot air and cure.

Note The optimum weight of coating is dependant on the degree of dilution and should be established by trials. The dilution usually suitable is 1Y part by volume of Accomet C to 4 parts by volume of clean cold water.

Warning: Care is needed to avoid build-up of an over-thick coating – e.g. in recessed areas or complex shapes. This is detrimental to bond strength; moreover it raises a health hazard through dust formation. Accomet C contains hexavalent chromium compounds. Any splashes on the skin should be washed off immediately with water. Dust from dried-off Accomet C must not be inhaled. Refer to Part 5 – Caution.

Curing Curing of the primer depends on the temperature at which the adhesive itself will be cured.

Araldite adhesives cured at temperatures below 100ºC

Cure the film of Accomet C primer for at least 20 seconds at 100ºC – 250ºC. A typical curing schedule is 30 seconds at 200ºC. Allow to cool before applying the adhesive.

Araldite adhesives cured at temperatures above 100ºC

The film of primer may be cured as above – i.e. prior to application of the adhesive. Alternatively apply the adhesive to the dried film, assemble the joint and cure the adhesive and primer in one operation.

Plastics

Thermosetting plastics: Mouldings, castings, laminates, etc can usually be bonded without difficulty. To ensure good bond strength, all soil and residual release agent must be removed from the joint surfaces before the Araldite adhesive is applied. The surfaces must either be abraded with emery cloth or grit-blasted, or they must be cleaned with a solvent such as acetone, methyl ethyl ketone, etc. Abrading or grit-blasting is recommended for mouldings since their surfaces may otherwise repel the adhesive.

Thermoplastics: These are often difficult to bond. Certain types permit only moderately successful bonding, and one and the same material may show considerable variation in properties determining the strength of a bond. Special adhesives have been developed, but they usually prove to be unserviceable when thermoplastics have to be bonded to materials such as wood, metal, etc. Araldite adhesives can be very useful in such cases even though their suitability for bonding thermoplastics is only limited. Pretreated thermoplastics for special applications (e.g. ski 'skins') are easily bonded with Araldite.

The grade of plastic and the manufacturing process used to make the component may influence the effectiveness of the chemical pretreatment. It is advisable to establish by trial whether the pretreatment is improved by adjusting the specified immersion time.

Note: Certain plastics are attacked by one or more of the halocarbon solvents listed in Part 1 – Degreasing. These plastics have suitable degreasing agents specified in the procedures given below. If the procedure does not specify particular degreasing agents, the halocarbon solvents listed in Part 1 are safe to use for the short times normally sufficient to carry out degreasing.

In addition to the normal mechanical and chemical methods of pretreatment, certain plastics can be pretreated using the following methods, all of which cause a change in the surface texture of the adherend. The change is brought about by the interaction of highly energised species with the adherend surface. These pretreatment methods have been applied to metals and in particular composites and plastics.

A low pressure plasma is an excited gas generated by applying a high frequency and high voltage between electrodes in a low pressure chamber. The advantage of this method is that it allows treatment of adherends by different plasmas of argon, ammonia, oxygen or nitrogen making the process suitable for a range of substrate types. Plasmas are generally used to activate the surfaces of adherends.
If instead, a plasma is created in air at atmospheric pressure, the air when ionised appears as a blue/purple glow with faint sparking, and is termed **a corona**. Corona treatments are usually applied for preparing thin polymer films and composite laminates.
The effect of **a flame treatment** is to oxidise the adherend, which produces polar groups creating a surface better suited to wetting by the adhesive. This method of surface pretreatment has been applied successfully to polyethylene/polypropylene. The variables of flame treatment include type of gas, gas/air (oxygen) ration, the rate of flow of mixture, exposure time and distance between flame and adherend.
All these methods have limited stability and vary from hours to weeks according to substrate. Suppliers of specialist equipment are listed on pages 18 and 19. Further information can be found in ISO 13895.

ABS plastics (acrylonitrile-butadiene-styrene)

Degrease according to Part 1 – Degreasing (page 2). Ketone solvents can be used advantageously for degreasing ABS plastics. Then either abrade according to Part 2 – Abrading (page 3),

II

or etch in a solution of:

Concentrated sulphuric acid (S.G. ca 1.83)	**10 litres**
Sodium dichromate ($Na_2Cr_2O_7.2H_2O$) or	
Potassium dichromate ($K_2Cr_2O_7$)	**140 gms**
Water	**3.3 litres**

Immerse for 15 minutes at room temperature, wash with clean cold running water, followed by clean hot water, and dry with hot air.

Note Prepare the solution according to the sequence specified on page 5 under Aluminium and Aluminium Alloys.

Warning Warning Concentrated sulphuric acid and chromic acid are highly corrosive and require special handling precautions. See Part 5 – Caution.

Acetal plastics (e.g. 'Delrin', 'Hostaform')

Degrease according to Part 1 – Degreasing (page 2). Ketone solvents can be used advantageously for degreasing Acetal plastics. Then either abrade according to Part 2 – Abrading (page 3),

or etch in a solution* of:

Concentrated sulphuric acid (S.G. ca 1.83)	**10 litres**
Sodium or potassium dichromate	**140gms**
Water	**3.3 litres**

Immerse for 5 minutes at room temperature, wash with clean cold running water, followed by clean hot water, and dry with hot air.

Note Prepare solution according to the sequence specified on page 5 under Aluminium and Aluminium Alloys.

Warning Warning Concentrated sulphuric acid and chromic acid are highly corrosive and require special handling precautions. See Part 5 – Caution.

The following* † solution is an alternative to the dichromate solution described above. It is more effective than the dichromate solution but its use necessitates carefully controlled ventilation of the work-area to remove the acrid fumes given off by the hot solution.

Para-Toluenesulphonic acid	**50 gms**
Dioxan	**0.5 litre**
Perchlorethylene	**10 litres**

Immerse for 5 minutes at 120^{o}C, wash with clean cold running water, followed by clean hot water, and dry with hot air.

† British Patent 1,025,675 (Celanese Corporation of America).

* Stresses due to moulding, machining etc, should be relieved by a suitable heat-treatment prior to acid etching. Advice concerning stress relieving should be sought from the manufacturer.

Acetal plastics can also be pretreated using Plasma treatment. The articles should ideally be bonded as soon as possible after Plasma treatment, but the treatment has a long effective shelf life enabling parts to be bonded several days after treatment.

Cellulose plastics
Degrease with a chlorinated solvent or detergent solution – according to Part 1 – Degreasing (page 2). Then abrade according to Part 2 – Abrading (page 3).

Warm preferably for 1 hour at 100ºC and apply the adhesive before the plastic cools completely to room temperature.

Decorative and industrial laminates
Degrease according to Part 1 – Degreasing (page 2). Then abrade according to Part 2 – Abrading (page 3).

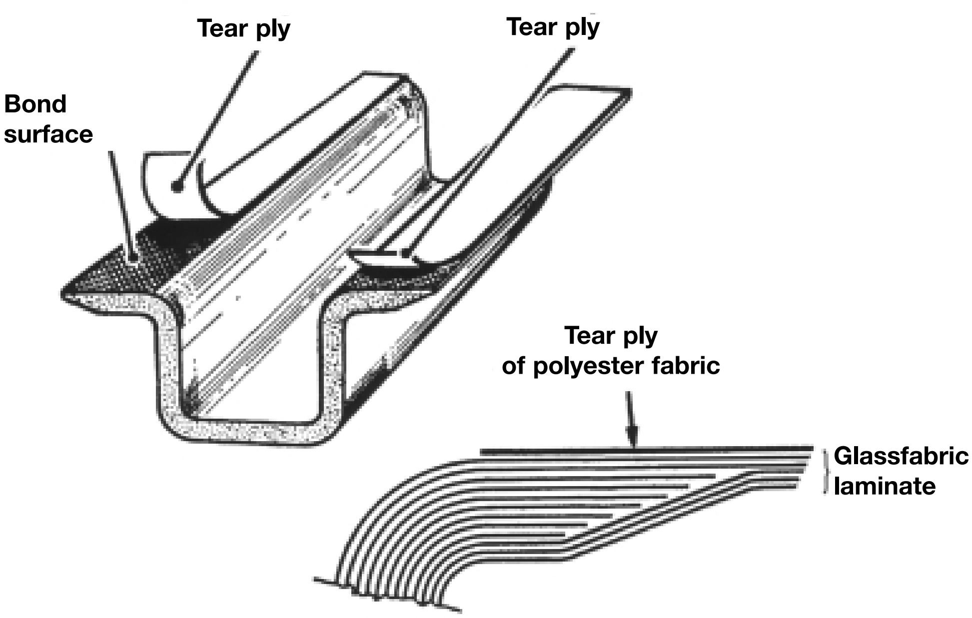
Note: Certain grades of decorative laminates are supplied sanded and need no abrasion.

or Pretreat using Corona/Plasma treatment (see Part 6 – Suppliers).

Glassfabric laminates
Degrease according to Part 1 – Degreasing (page 2). Then abrade according to Part 2 – Abrading (page 3).

II

Alternatively, design the laminate so that a 'tear ply' of fine closeweave polyester fabric is placed at the surface to be bonded. (The ply becomes part of the laminate on curing.) Just prior to bonding, tear off the ply to expose a fresh clean bond surface on the laminate.



Note Fineweave polyester sailcloths are a suitable tear-ply material.

Polyamides (Nylon)
Degrease according to Part 1 – Degreasing (page 2). Ketone solvents can be used advantageously to degrease polyamide. Then either abrade according to Part 2 – Abrading (page 3),

or Prime with the two-part solution of Redux® 101 primer described on page 13 under Primer for Thermoplastics.

or Pretreat using Corona/Plasma treatment (see Part 6 – Suppliers).

Polyacrylics (e.g. Perspex)
Degrease with alcohol solvent or detergent solution – according to Part 1 - Degreasing (page 2). Then abrade according to Part 2 – Abrading (page 3), and remove dust with alcohol solvent.

For optimal results, it is recommended to stress relieve the plastic by annealing.

Polycarbonate (e.g. Makrolon; Lexan)
Degrease with alcohol solvent (e.g. isopropanol) or detergent solution - according to Part 1 - Degreasing (page 2). Then abrade according to Part 2 - Abrading (page 3),

or Pretreat using Corona or Plasma treatment (see Part 6 – Suppliers).

Polyesters

Thermosetting (unsaturated) polyester resins – see Thermosetting Plastics.

Thermoplastic (polyteraphthlate) polyester mouldings ('Crastine', 'Kelanex') and films ('Melinex', 'Mylar'): Degrease according to Part 1 – Degreasing (page 2), using ketone solvents. Then either abrade according to Part 2 – Abrading (page 3),

or Pretreat by Corona or Plasma treatment. (See Part 6 – Suppliers.)

or Prime with the two-part solution described on page 13 under Primer for Thermoplastics,

or Etch in a solution of:

Caustic soda (2kg) in water	**8 litres**

Immerse for 6 minutes at 75-85°C, wash with clean running cold water, followed by clean hot water, and dry with hot air. This treatment will give the best bond strengths

Note: Prepare the solution according to the sequence specified on page 8 under Tungsten and Tungsten Carbide.

Polyolefines (polythene, polypropylene)

Either lightly flame with a waving motion in an oxidising (i.e. blue – not yellow) gas flame until the surface is shiny. Natural gas is particularly suitable, but care must be taken to avoid overheating and melting. (see also page 10).

II

or Pretreat by Corona or Plasma treatment. See Part 6 – Suppliers.

The following chemical treatment is in general an alternative to flame or corona treatment. Certain grades of polypropylene are, however, not affected by the chemical solution and for these it is not efficaceous as a pretreatment. Preliminary trials are essential when using the chemical solution as a pretreatment for polypropylene.

Degrease according to Part 1 – Degreasing* (page 2). Then etch in a solution of:

Concentrated sulphuric acid (S.G. ca 1.83)	**10 litres**
Sodium dichromate ($Na_2Cr_2H_2O$) or	
Potassium dichromate ($K_2Cr_2O_7$)	**0.3 kg**
Water	**2 litres**

Immerse for 15 minutes at room temperature, wash with clean cold running water, followed by clean hot water, and dry with hot air.

Note Prepare the solution according to the sequence specified on page 5 under Aluminium and Aluminium Alloys.

Warning: Concentrated sulphuric acid awnd chromic acid are highly corrosive and require special handling precautions. See Part 5 – Caution.

Proprietary primers for polypropylene are available which provide an alternative to flame, corona and chemical pretreatments described above. Almost as effective as these pretreatments is degreasing according to Part 1 – Degreasing (page 2), then priming with ISF Polypropylene Primer (according to the supplier's directions), and allowing to dry in air.

*Certain grades of polythene are attacked by trichloroethylene. It is advisable to use an alternative solvent. See page 2 – when degreasing polythene.

Polyphenylene oxide and similar plastics

Degrease according to Part 1 – Degreasing (page 2), using alcohol solvents. Then either abrade according to Part 2 – Abrading (page 3),

or etch in sulphuric acid/dichromate solution at 70°C for 15 seconds and rinse in clean running water. Dry with hot air.

Polystyrene

Degrease with alcohol solvent or detergent solution according to Part 1 – Degreasing (page 2). Then either abrade according to Part 2 – Abrading (page 3).

The following alternative procedure is more effective than the above but the solution is considerably less convenient to handle.

Or etch in a solution of:

Concentrated sulphuric acid (S.G. ca 1.83)	**10 litres**
Sodium dichromate or potassium dichromate	**0.3 kg**
Water	**2 litres**

Immerse for 15 minutes at 68-72°C, wash with clean cold running water, followed by clean hot water, and dry with hot air.

Note Prepare the solution according to the sequence specified on page 5 under Aluminium and Aluminium Alloys.

Warning Concentrated sulphuric acid and chromic acid are highly corrosive and require special handling precautions. See Part 5 – Caution.

Polyurethane

Degrease according to Part 1 – Degreasing (page 2). Then abrade according to Part 2 – Abrading (page 3).

or Pretreat with Corona/Plasma treatment (see Part 6 – Suppliers).

PTFE and similar fluorocarbon plastics*

Fluorocarbon plastics such as PTFE ('Fluon', 'Teflon') cannot normally be bonded in the untreated condition. There are, however, specialised processes (involving flame oxidisation or exposure to dispersions of metallic sodium) for treating the surfaces of fluorocarbon plastics. PTFE already treated by such processes is available from suppliers listed in Part 6.

II

An etch solution can be made up as follows:
Pour 2 litres tetrahydrofurane into a three-necked flask fitted with a calcium chloride tube and a mixer. In it dissolve 256 grammes naphthalene, then add 46 grammes fragmented metallic sodium. The sodium will take about 2 hours to react with the naphthalene. The solution will then be brownish black in colour and ready for etching PTFE; it will keep for 2 to 3 months in a sealed container.
Immerse the PTFE surfaces for about 15 minutes at room temperature, then wash them with acetone, followed with clean running water, and dry thoroughly. The dry, etched PTFE will be brown in colour.

PTFE pretreated for bonding with Araldite is available in foil and sheet form from various firms. Names and addresses are available on request.

Suppliers of proprietary etches include:

R. D. Taylor, 240 Edmiston Drive, Ibrox, Glasgow GS1 2YT
(tel. 0141 427 5103).

C.Huth & Söhne, Bietigheim/Württ, Germany. (Teflon-Ätzmittel Typ AM 92).

* The principal flourocarbon plastics are: PTFE (polytetrafluoroethylene), PCTFE (polychlorotrifluoroethylene), FEP (fluorinatedethylene-propylene) and PVF (polyvinylfluoride).

Warning: Metallic sodium is dangerous in contact with air or water, special care is needed when handling this material. See Part 5 – Caution.

PVC

Degrease according to Part 1 – Degreasing (page 2), using Ketone or chlorinated solvents. Then abrade according to Part 2 – Abrading (page 3).

Thermosetting plastics (amino, diallyl phthalate, epoxy, phenolic, unsaturated polyester)

Degrease according to Part 1 – Degreasing (page 2), using ketone solvents. Then abrade according to Part 2 – Abrading (page 3).

Note Laminated thermosetting plastics. See page 11.

Primer for thermoplastics

The primer described below is a two part low-viscosity cold-setting solution. The primer markedly improves the bond strengths of Araldite adhesives to nylon-type plastics (nylon and similar polyamide products). The use of the primer is recommended when bonding nylon-type plastics to themselves – or to other materials such as metal or glass. The primer likewise improves the bond strengths of Araldite adhesives to certain different plastics, such as Perspex, polycarbonate, polystyrene and thermoplastic polyesters.

The primer is also effective as a surface protection solution for these plastics types when there is to be a delay between pretreatment and bonding. The primed surfaces – kept in a dust-free atmosphere (for example, in polythene bags or covered by polythene sheet) in a clean dry place – remain effective for 2-3 months.

The primer is a resin + hardener solution mixture consisting of:

Resin solution	Redux 101	100	pbw
Hardener solution	para-Toluenesulphonic acid	2.9	pbw
	Industrial methylated spirit	25	pbw

The two solutions must be stored separately. Prior to use: mix the resin and hardener solutions together at room temperature, stirring thoroughly. The correct ratio must be ensured.

A 2 litre quantity of mixture has a usable life of ca 6 hours at room temperature.

Apply a thin even coating of the mixture by spatula, brush or roller to the bond surfaces of the thermoplastic only. Diluting the mixture with methylated spirit facilitates spray or dip application.

Dry and partially cure the coated surfaces before applying the Araldite adhesive. Typical times are:
4 hours at 20ºC or 1 hour at 40ºC or 10 mins at 60ºC or 5 mins at 80ºC or 3 mins at 100ºC.

Apply the Araldite adhesive to the primed surfaces, assemble the joint and, while maintaining light contact pressure, cure the primer plus adhesive in one operation.

Miscellaneous materials

Asbestos board

Degrease according to Part 1 – Degreasing (page 2). Allow the board to stand a few minutes to ensure all the degreasing agent evaporates out.

Warning: Loose particles and dust must be removed with extreme care – asbestos is a serious hazard by inhalation of fibres. Refer to local regulations.

Bricks and other fired non-glazed building materials

Degrease according to Part 1 – Degreasing (page 2). Brush with a wire brush and remove dust.

Carbon

Degrease according to Part 1 – Degreasing (page 2). Abrade with fine abrasive cloth or paper, and remove dust.

Ceramics

Degrease according to Part 1 – Degreasing (page 2). Abrade with a slurry of silicon carbide powder and water.

Concrete

Remove heavy grime and laitance by wire brushing. Degrease with detergent solution, according to Part 1 – Degreasing (page 3, section d).

Note: Where concrete is deteriorated and weak, the surface must be removed until sound concrete is exposed.

Even where concrete is sound, it should be pretreated wherever practicable by one of the following methods. Method 1 is more effective than 2, and 2 is more effective than 3.

1. **Remove by mechanical scarification 3mm – or more – of all surfaces to be bonded, then remove dust preferably by vacuum cleaner; or**
2. **Sand-blast about 1.5mm off all surfaces to be bonded, then remove dust preferably by vacuum cleaner; or**
3. **Etch with 12% hydrochloric acid or sulphamic acid solution (1litre per square metre, spread by stiff-bristle brooms) until bubbling subsides (about 15 minutes). Wash with clean water by high-pressure hose until all slush is removed and the surface is neutral to litmus. Final rinsing with 1% ammonia solution followed by clean water is good practice – this ensures thorough neutralisation. Allow the surface to dry thoroughly. Remove dust preferably by vacuum cleaner.**

Note: Preparation of 12% hydrochloric acid solution: pour 2 litres of clean cold water into a clean polythene or earthenware container. While stirring the water, add 1 litre of concentrated hydrochloric acid (S.G. ca 1.18) in a slow steady stream. Preparation of 12% sulphamic acid solution: fill a calibrated clean polythene or earthenware container to the 8litre mark with clean warm water. Slowly sprinkle, with stirring, 1kg of sulphamic acid crystals onto the water. Continue stirring until the acid is completely dissolved. (The crystals do not dissolve in cold water).

Warning: Concentrated hydrochloric acid is a highly corrosive chemical. Particular care is needed when handling the acid. See Part 5 – Caution (page 17).

Earthenware

Degrease according to Part 1 – Degreasing (page 2). Then abrade according to Part 2 – Abrading (page 3).

Glass

Degrease according to Part 1 – Degreasing (page 2)*. Then abrade according to Part 2 – Abrading (page 3).
Either warm for ½ hour at 100ºC and apply the adhesive before the glass cools completely to room temperature,
Or, for bonds with improved long-term resistance to water:
Prime the glass surfaces with 5% solution of Silane A-187. Allow to dry in air – or dry with hot air.

Note: Preparation of 5% Silane A-187 solution: pour 85ml of methylated spirit into a container with a 100ml calibration mark. Add 5ml of Silane A-187. Stir thoroughly, then fill to the 100ml calibration mark with water. (This 9: 1 methylated spirit + water mixture imparts 2-3 days storage stability to the solution).

Warning Silane A-187 is toxic: it is harmful if taken internally or absorbed through the skin. Goggles and protective clothing must be worn when Silane A-187 is handled.

* Treatment with the sulphuric acid + sodium dichromate solution specified on page 5 under Aluminium and Aluminium Alloys is a highly effective method for degreasing glass. Immerse for 30 seconds at 60-65c, then wash with clean cold running water, followed by clean hot water, and dry with hot air.

Graphite

Degrease according to Part 1 - Degreasing (page 2). Abrade with fine abrasive paper or cloth, and remove dust.

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Jewels

Degrease according to Part 1 – Degreasing (page 2).

Leather

Degrease according to Part 1 – Degreasing (page 2). Roughen with abrasive paper and remove loose particles.

Plaster

Allow the surface to dry thoroughly. Smooth with fine abrasive paper or cloth, and remove dust.

Rubber

Degrease with trichlorotrifluoroethane or detergent solution – according to Part 1 – Degreasing (page 2). Then etch with modified bleach solution, with concentrated sulphuric acid depending on the type of rubber.

Note: Degreasing with clean cold *trichloroethylene* is effective as a pretreatment for certain rubber types – see the comparison table on page 16. Particular care must be taken when handling trichloroethylene and trichlorotrifluoroethane. See Part 5 – Caution (page 17). Owing to the increased toxic hazard raised, the manual use of trichloroethylene is in general not an approved practice.

Modified bleach solution

Household bleach (standard type)	300ml
Concentrated hydrochloric acid (S.G. ca 1.18)	50ml
Water	10 litres

Immerse for 1-3 minutes at room temperature, wash with cold clean water, followed by clean hot water, and dry with hot air.

Note: Concentrated hydrochloric acid is a highly corrosive chemical. Particular care is needed when handling the acid. See Part 5 – Caution (page 17).

Prepare the modified bleach solution by pouring the clean water into a clean container made of plastic, glass or similar inert ware. While stirring the water, add the concentrated hydrochloric acid in a slow steady stream. Then add the household bleach, stirring it thoroughly into the diluted acid. Never pour the household bleach into the acid (or the other way round) without adding the water first.

Fresh solution should be made up each day. The solution gives off chlorine: good ventilation is essential.

Sulphuric acid etch

Concentrated sulphuric acid (S.G. ca 1.83)

Immerse for 2-10 minutes at room temperature, wash with clean cold running ware, followed by clean hot water, and dry with hot air.

Note: Immersion time depends on the rubber type and grade. For optimum surface properties, immersion should continue only until flexing the rubber produces fine crazing over the joint surfaces. Particular care is needed when handling concentrated sulphuric acid. See Part 5 – Caution (page 17).

Vertical surfaces may be treated with a paste prepared by adding sufficient barytes powder to the acid to prevent it from flowing.

The modified bleach solution is however considerably less hazardous to handle than concentrated sulphuric acid.

Relative effectiveness of pretreatment liquids

	Trichloroethylene solvent	Modified bleach solution	Sulpuric acid
		Effectiveness of solvent or solution	
Butyl	**R**	**R**	-
Ethylene propylenediene monomer (EPDMR)	**R**	E	E
Natural	E	**R**	-
Neoprene	-	-	**R**
Nitrile	**R**	**R***	-
Styrene-butadiene	-	-	E

Key: **R = Recommended** E = Effective - = Not Effective

Joints pretreated with a liquid rated as *Recommended* for the particular rubber type give the highest bond strengths. The use of a liquid rated as *Effective* gives substantially improved bond strengths; they are however, less than the strengths obtained after pretreatment with a *Recommended* liquid.

* *Nitrile rubber joints* Highest bond strengths are obtained after pretreatmnent with modified bleach and bonding with Araldite AY 103 + Hardener HY 991. Nitrile rubber joints bonded with different Araldite adhesives give lower bond strengths. The best bond strengths given by these alternative adhesives are obtained after pretreatment with trichloroethylene.

Rubber-silicone
Silicone rubbers, by their chemical nature, are unsuitable for bonding with Araldite adhesives.

Stonework
All the surfaces to dry thoroughly. Brush with a wire brush and remove dust.

Wood
Ensure the wood is dry. Plane – or abrade with glass paper and remove dust.

Note The moisture content of the wood should not exceed 16%. Some hardwoods can be difficult to bond.

Part 4 – Essentials for maximum bond strength

Araldite adhesives are simple to use, but to ensure successful bonding the directions given in the instructions supplied with the adhesive must be strictly observed.

In particular:

1. Resin and hardener must be correctly proportioned and thoroughly mixed together.
2. Joint surfaces must be degreased and, when necessary, pretreated.
3. Curing temperature and curing time must be correct.
4. Jigs or other fixtures must be used to prevent the bond surfaces from moving relative to one another during the curing process.
5. Though only light pressure is needed, it should be applied as evenly as possible over the whole bond area. Excessive pressure leaves the joint starved of adhesive.

II

General
This manual lists many chemicals that require cautionary labelling under local legislation in many countries, e.g. UK legislation – Chemicals (Hazard Information on Packages) Regulations 1993. It is important to read, and fully understand, suppliers' technical and safety data sheets, making sure all precautions are in place before commencing work.

Halocarbon solvents
As halocarbon solvents remove the natural grease from the skin, contact with the hands should be avoided as far as possible. Suitable gloves – e.g. nitrile, should be worn.

The place of work should be well ventilated with an efficient extraction system. Information on safe working concentrations of vapour is given in the current edition of *Occupational Exposure Limits – Guidance EH40**.
The vapours from halocarbon solvents have an anaesthetic effect and consequently cause drowsiness if inhaled in quantity. Any person so affected should lie down in the open air and be kept quiet and warm (given no exercise) while a doctor is called.

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*Guidance Note EH40 is available from HSE Books, PO Box 199, Sudbury, Suffolk CO10 6FS. Outside the UK, advice should be sought from the Health and Safety Authority in the user's country.

Where halocarbon vapour is present, open flames and smoking must be prohibited – they cause the vapour to form poisonous gases.

Acids, caustic soda, etc.
Concentrated acids, oxidising agents e.g. dichromates, and hot caustic soda solution are highly corrosive chemicals. Spillages and splashes can cause severe damage to eyes and skin, and attack ordinary clothing. Fine-particle mists resulting from the stirring or agitation of the solutions can present a severe respiratory hazard. Operators must wear personal protection e.g. goggles, protective clothing, respirators.

Important Never pour water into acids. Always pour the acid in a slow steady stream into the water, with continuous stirring. Bear in mind that the handling hazard is intensified when the acid is hot.

Chromium Compounds
These materials have health hazards ranging from harmful to those that are highly toxic and carcinogenic. Ensure that the latest safety data is available for the particular compound chosen.

Sodium
Pieces of sodium react violently and may explode on contact with water, emitting flammable hydrogen gas. Sodium burns spontaneously in air and vapours ignite at room temperature. It also reacts explosively with many aqueous solutions and some organic solvents – mainly chlorinated hydrocarbons – and reacts vigorously with many others on heating. Sodium reacts incandescently with some (mostly halogenated) compounds. Mixtures of sodium and metal halides are sensitive to mechanical shock. Sodium is highly toxic and corrosive, causing severe thermal and caustic burns to tissues in the presence of moisture.

Araldite products
Araldite resins and hardeners are generally quite harmless to handle provided that certain precautions normally taken when handling chemicals are observed. The uncured materials must not, for instance, be allowed to come in contact with foodstuffs or food utensils, and measures should be taken to prevent the uncured materials from coming in contact with the skin, since people with particularly sensitive skin may be affected. The wearing of impervious rubber or plastic gloves will normally be necessary; likewise the use of eye protection. The skin should be thoroughly cleansed at the end of each working period by washing with soap and warm water. The use of solvents is to be avoided. Disposable paper towels – not cloth towels – should be used to dry the skin. Adequate ventilation of the working area is recommended. These precautions are described in greater detail in the Huntsman Advanced Materials Manual *Hygienic Precautions for Handling Plastics Products of Huntsman Advanced Materials* and in the Huntsman Advanced Materials Product Safety Data Sheets for the individual products. These publications are available on request and should be referred to for fuller information.

Selected suppliers of pretreatment materials and adhesives processing equipment

Abrasives and blasting equipment

Matrasur département Vocublast, Z.A. Les Glaises, 36 avenue du 1er Mai, 91124 Palaiseau Cedex, France (tel. (1) 69 19 17 25)

Wheelabrator Sisson Lehmann, Z.I. de Mahon, 24 rue Camille Didier, B.P. 39, 08001 Charleville Mezieres Cedex, France (tel. (?) 24 33 63 00)

RAGA GmbH, Kriegsbergstr. 12, D-71336 Waiblingen, Germany (tel. 07151-98901-0)

Werkzeug-und Industrieausrüstungs GmbH, Niederstr. 24, D-40789 Mannheim, Germany (tel. 02173-52029)

Abrasive Developments Limited, Norman House, Henley-in-Arden, Solihull, West Midlands B95 5AH, UK (tel. 01564-792231)

Guyson International Limited, PO Box 18, West Yorkshire LS21 1RD, UK (tel. 01756 69911)

UCF, Woodson House, Ajax Avenue, Slough SL1 4DJ, UK (tel. 01753 526511)

Solvents and chemicals

Accomet C

Albright & Wilson Ltd, Surface Technologies European Headquarters, PO Box 3, 210-222 Hagley Road West, Oldbury, Warley, West Midlands B68 ONN, UK (tel. 0121-429-4942)

Bonderite 255

Brent Europe, Adrox Pyrene Limited, Ridgeway, Iver, Buckinghamshire (tel. 01753 630200)

ISF Polypropylene Primer

International Shoe Findings Ltd, Thurmaston Boulevard, Leicester LE4 7HS, UK (tel. 01162 742222)

Redux 101

Hexcel Composites, Duxford, Cambridge CB2 4QA (tel. 01223 833141)

Silane A-187

Albright and Wilson Limited, 210-222 Hagley Road West, Warley, West Midlands B68 0NN (tel. 0121 429 4942)

Trichloroethylene

Lambert & Riviere, 17 ave Louison Bobet, Val de Fontenay, 94132 Fontenay Sous Bois Cedex, France (tel. (1) 49 74 80 80)

S.P.C.I. (Sté de Produits Chimiques Industriels), 43 rue Cristino Garcia, B.P. 43, 93212 La Plaine Saint Denis, France (tel. (1) 49 33 31 31)

Biesterfeld, Gertrudenstraße 14, D-20095 Hamburg, Germany (tel. 040-30080)

Brenntag AG & Co, Humboldring 15, D-45472 Mülheim a.d. Ruhr, Germany (tel. 0208-4940)

Deutsche Solvay-Werke GmbH, Langhansstr 6, D-42697 Solingen, Germany (tel. 0212-704-0)

ICI Chemicals & Polymers Ltd, Solvents Sales Department, PO Box 14, Runcorn, Cheshire WA7 4QG, UK (tel. 01928 511190)

Samuel Banner & Co Ltd, 59-61 Sandhills Lane, Liverpool L5 9XL, UK (tel. 0151 944 7000)

Plasma and Corona Pretreatment equipment

Sherman Treaters N.A. Inc, 964, Westport Crescent, Unit 7, Mississauga, Ontario, L5T 1GI, Canada (tel. (416) 670 9117)

RR Print A/S, Rodovrevej 245, DK-2610 Rodovre, Denmark (tel. 010 45 31 41 36 66)

Cogeplast, 39 rue Rouvrel, 80110 Mailly Raineval, France (tel. (3) 22 09 82 43)

Polymix Z.I., 6 rue de l'Industrie, 68126 Bemlihr Sare, France (tel. (3) 89 20 13 80)

Perkin Elmer (Division Instruments), 1 rue Franklin, B.P. 304, 78054 Saint Quentin en Yvelynes Cedex, France (tel. (1) 30 85 63 63)

Marcatec International, 10 rue du Noyer, F-67203 Oberschaeffolsheim, France (tel. (3) 8878 5880)

Soreprind, 2 rue Condorcet 4200 Saint Etienne, France (tel. (4) 77 74 20 01)

Arcotec Oberflächentechnik GmbH, D-7251 Wimsheim, Mönsheimer Straße 20, Postfach 1101, Germany (tel. 07044 5952/4077)

BMP Plasmatechnologie GmbH, Dr. Guido Bell, Weissenfelder Straße 4, D-85551, Kirchheim, Germany (tel. 089-9039903)

Buck Technologien, Plasma Electronic, Buck Plasma Electronic GmbH, Fabrikstr. 17, D-70794 Filderstadt, Germany (tel. 0711-77903-0)

Haug GmbH & Co. KG, Friedrich-List-Strasse 18, 7022 Leinfelden-Echterdingen 2, Germany (tel. 0711 94980)

Plasma-electronic, Fabrikstraße 17, D-7024 Filderstadt 4 (Bonlanden), Germany (tel. 0711-7775517)

II

Plasmatechnologie Blersch GmbH, Eichenweg 13, D-70771 Leinfelden-Echterdingen, Germany (tel. 0711-791106)

Technics Plasma GmbH, Dieselstr. 22a, D-85551 Kirchheim (bei München) Germany (tel. 089-905030)

Tigres Dr. Gerstenberg GmbH i.G., Zum Fürstentor 11, D-21079 Hamburg, Germant (tel. 040-79012300)

Plasonic Oberflächentechnik GmbH, Carl Zeiss Straße 9, D-70839 Gerlingen, Germany (tel. 07156-23711)

Webber Brennertechnik, Landmannweg 18, 4600 Dortmund – Oespel 1, Germany (tel. 0231 652 438)

The Aerogen Company Limited, Newman Lane, Alton, Hampshire GU34 2QW, UK (tel. 01420 83744)

Combustion Engineering Consultancy, Weydale Rise, Alton, Hampshire GU34 2TY, UK

Sherman Treaters Ltd, Dormer Road, Thame Industrial Estate, Thame, Oxon OX9 3UW, UK (tel. 01844 213686)

Acovent Srl, Via Cavour, 81-83, 20030 Senago, Milan, Italy (tel. (2) 998 61 9192)

F.D.M., S.R.L., Corco Sempione 80/a, I-20015 Parabiago, Milano, Italy (tel. 331 554 826)

Logopak 8.V, Prof. Lorentzweg 6 B, NL-5140 Waalwijk (tel. 4160 44144)

Tantec Inc, 630 Estes Avenue, Schaumburg, IL 60193, USA (tel. (708) 529 5506)

Flame pretreatment equipment

Sherman Treaters N.A. Inc., 964, Westport Crescent, Unit 7, Mississauga, Ontario, L5T 1GI, Canada (tel. (416) 670 9117)

RR Print A/S, Rodovrevej 245, DK-2610 Rodovre, Denmark (tel. 31 41 36 66)

Boussey Control, 10 rue d'Abrantés, 21500 Montbard, France (tel. (3) 80 89 1 10)

Donze, La Flie, B.P. 51, 54460 Liverdun, France (tel. (3) 83 24 49 95)

Soreprind, 2 rue Condorcet 4200 Saint Etienne, France (tel. 77 74 20 01)

Webber Brennertechnik, Landmannweg 18, 4600 Dortmund – Oespel 1, Germany (tel. 0231 652 438)

The Aerogen Company Limited, Newman Lane, Alton, Hampshire GU34 2QW, UK (tel. 01420 83744)

Sherman Treaters Ltd, Dormer Road, Thame Industrial Estate, Thame, Oxon, OX9 3UW, UK (tel. 01844 213686)

Flynn Controls B.V., Industrial Control Systems, Zonnebaan 27-29, 3606 CH Maarssen, Holland (tel. 30 41 4199)

Acovent Srl, Via Cavour, 81-83, 20030 Senago, Milan, Italy (tel. (2) 998 61 9192)

Flame Treaters esse CI s.r.l, Via Flaminia 386 I-05035 Narni TR, Italy (tel. (0744) 72 67 41)

Matsuzaka Co.Ltd, S.Otsuka, Nishimatsu Bldg, 20-10 Toranomon, 1 Chome Minato-Ku, Tokyo 105 Japan (tel. 011/81 33 502 1251)

Flynn Burner Corporation, 425 Fifth Avenue, New Rochelle, NY 10802, USA (tel. (914) 636 1320)

The individual materials covered by this instruction manual are mainly those in common use in industry. Engineers contemplating the bonding of particular materials not listed below are invited to submit enquiries concerning appropriate pretreatments to our technical staff.

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www.araldite.com

HUNTSMAN

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Open type

Shielded type

Sealed type (non-contact)

Expansion Compensating Bearing

II

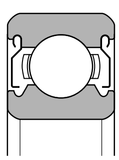
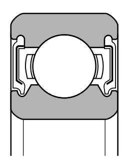
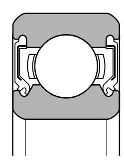
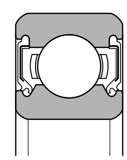
1. Design features and special characteristics

Deep groove ball bearings are very widely used. A deep groove is formed on each inner and outer ring of the bearing enabling them to sustain radial and axial loads in either direction as well as well as the complex loads which result from the combination of these forces. Deep groove ball bearings are suitable for high speed applications.

In addition to unsealed bearings, deep groove ball bearings include ball bearings with greased sealed inside (sealed or shielded) and bearings with a snap ring that simplify structure around the bearing and design.

Table 1 shows the construction and special characteristics of various sealed deep groove ball bearings.

Table 1 Sealed ball bearings: construction and characteristics

Type, code no.		Shielded type	Sealed type		
		Non-contact type ZZ	Non-contact type LLB	Contact type LLU	Low torque type LLH
Construction					
		● Metal shield plate is affixed to outside ring; inner ring incorporates a V-groove and labyrinth clearance.	● Outer ring incorporates synthetic rubber molded to a steel plate; seal edge is aligned with V-groove along inner ring surface with labyrinth clearance.	● Outer ring incorporates synthetic rubber molded to a steel plate; seal edge contacts V-groove along inner ring surface.	● Basic construction the same as LU type, but specially designed lip on edge of seal prevents penetration by foreign matter; low torque construction.
Performance comparison	Torque	Low	Low	Rather high	Medium
	Dust proofing	Very good	Better than ZZ-type	Excellent	Much better than LLB-type
	Water proofing	Poor	Poor	Very good	Very good
	High speed capacity	Same as open type	Same as open type	Limited by contact seals	Much better than LLU-type
	Allowable temp.range ❶	Depends on lubricant	-25 ℃～120 ℃	-25 ℃～110 ℃	-25 ℃～120 ℃

❶ Please consult NTN Engineering about applications which exceed the allowable temperature range of products listed on this table.
Note : This chart lists double shielded and double sealed bearings, but single shielded (Z) and single sealed (LB, LU, LH) are also available.
Grease lubrication should be used with single shielded and single sealed bearings.

2. Standard cage types

As shown in **Table 2**, pressed cages are generally used in deep groove ball bearings. Machined cages are however used for large bearings and high-speed bearings.

Table 2 Standard cage for deep groove ball bearings

Bearing series	Pressed cages	Machined cages
67	6700～ 6706	—
68	6800～ 6834	6836～ 68／600
69	6900～ 6934	6936～ 69／500
160	16001～16052	16056～ 16072
60	6000～ 6052	6056～ 6084
62	6200～ 6244	—
63	6300～ 6344	—
64	6403～ 6416	—

3. Other bearing types

3. 1 Bearings with snap rings

Some bearings accommodate a snap ring which is attached along the outer diameter of the outer ring. By using snap rings, positioning in the axial direction is possible and housing installation is simplified. In addition to open type, shielded and sealed types are also manufactured. Consult NTN Engineering.

3. 2 Expansion compensating bearings (creep prevention bearings)

The boundary dimensions of expansion compensating deep groove ball bearings are the same as for standard bearings, but formed high polymer material with a high expansion rate is provided in the grooves on the outer circumference of the outer ring (see **Diagram 1**) .

Due to the extremely small difference of thermal expansion attained between the fitted surfaces of the high polymer equipped outer ring and the light alloy bearing housing, a good interference fit can be achieved with stable performance across a wide temperature range. Another advantage is a large reduction in the occurrence of outer ring creeping.

(1) Allowable load

Maximum allowable load C_p (refer to the table of boundary dimensions) has been determined in accordance with outer ring strength; therefore, **it is necessary to select a bearing with a maximum allowable load greater than the largest anticipated bearing load.**

(2) Housing and bearing fit

Table 3 shows the recommended fits for bearings with light metal alloy housings.

In cases where the bearing is going to be interference fit with the housing, it is very important not to damage the high polymer material. Therefore it is essential that the lip of the housing diameter be given a 10°–15° chamfer as shown in **Diagram 2**.

Furthermore, as shown in **Diagram 2**, it is also advisable to apply the interference fit using a press in order not force the bearing into the housing in a misaligned position. (**Diagram 2**)

(3) Radial internal clearance

Regulations for radial internal clearance are the same as those for standard deep groove ball bearings. For standard fit and application conditions, a C3 clearance is used with

Table 3 Recommended fits for outer ring and housing bore

Conditions		Suitable bearing	Housing bore tolerance class
Load type, etc.	Housing material		
Rotating outer ring load Rotating inner ring load; light load Direction indeterminate load; ordinary load	Al alloy Mg alloy Other light alloys	Deep groove ball bearing Cylindrical roller bearing	H6
Rotating outer ring load; heavy load Direction indeterminate load; shock load	Al alloy Mg alloy Other light alloys	Thick-walled type deep groove ball bearing	N6

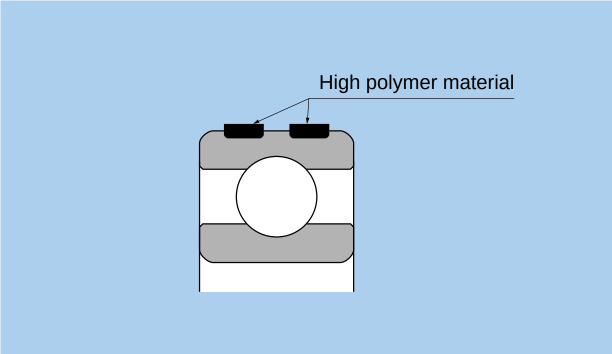


Diagram 1. Expansion compensating bearings

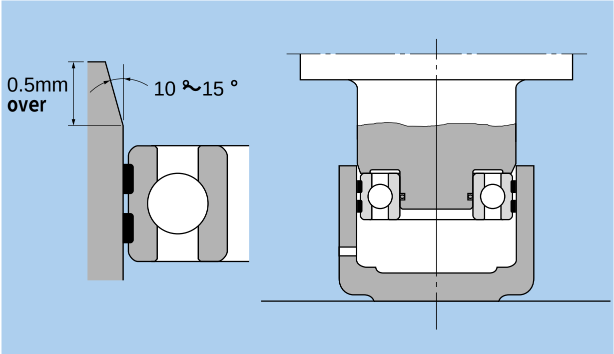


Diagram 2. Fitting method and housing inner diameter chamfer

this bearing.

For more detailed information concerning this bearing and the availability of roller bearings contact NTN Engineering.

(4) Allowable temperature range

-20～120˚C

3.3 Long-life bearings (TMB/TAB bearings)

Boundary dimensions of long-life bearings are the same as those of standard deep groove ball bearings, but the bearings have undergone special heat treatment that considerably extends wear life.

These bearings are especially effective in countering reduced wear life due to the effects of infiltration by dust and other foreign matter.

Features are as follows:

- Rated load is the same as standard bearings, but shaft characteristics factor is a_2 = 2.2 for TMB bearings and a_2 = 3.6 for TAB.
- TMB 62 series bearings can be used in place of standard 63 series bearings enabling lighter weight, more compact designs
- Greater resistance to reduced wear life due to infiltration by dust and other foreign matter

Dimensions for these bearings are not provided in the dimensions table. For details, please contact NTN Engineering.

3.4 AC bearings (creep prevention bearings)

AC bearings have the same boundary dimensions as standard bearings with the addition of two O-rings imbedded in the outside circumference of the outer ring. (**Diagram 3**)

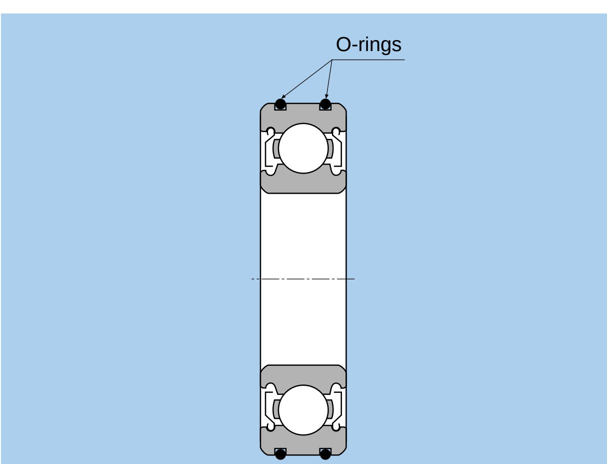


Diagram 3. AC bearing

This bearing has a steel housing, can withstand rotating outer ring loads, and is suitable for applications where a "tight fit" is not possible but the fear of creeping exists. With its capacity for axial load displacement, an AC bearing can also be installed as a floating side bearing to accommodate shaft fluctuations. Before installing the bearing into the housing, high viscosity oil (base oil viscosity, 100 mm²/s or more) or grease should be applied to the space between the two O-rings. This lubricant forms a thin oil layer inside the bearing which prevents contact between the outer ring and housing, lowers the coefficient of friction, and is still able to prevent creeping by utilizing the friction force of the O-rings.Outer ring spin is prevented by friction force of the O-ring and housing.

For dimensional specifications, handling procedures, and other detailed information concerning AC bearings, contact NTN Engineering.

(1) Allowable load

Because allowable load C_p that takes outer ring strength into account (see dimensions table) is established, selection must be made so that maximum load on the bearing does not exceed C_p.

(2) Fit with housing

Table 4 gives recommended fit with steel housing.

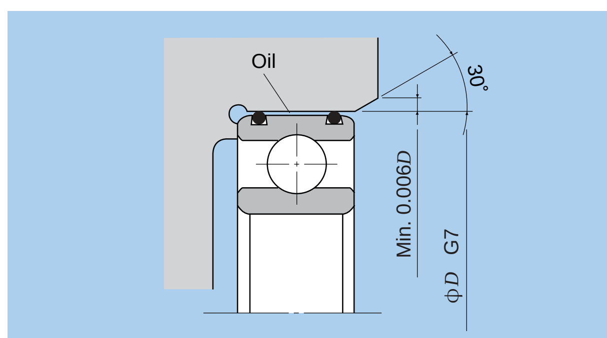


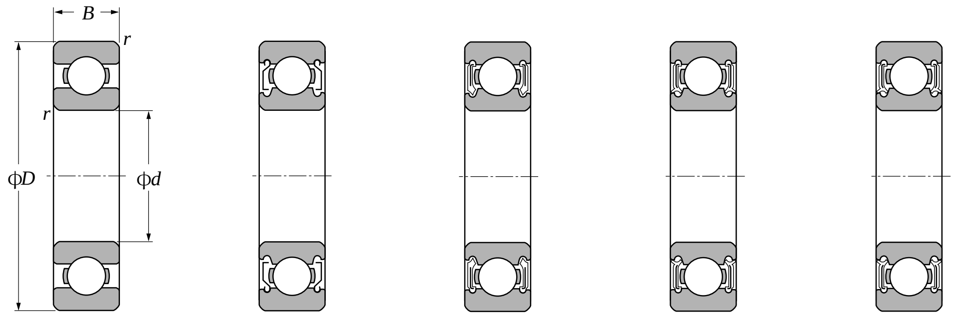
Diagram 4. Housing

Table 4 dimensions and shape

Housing bore tolerance	G7
Housing bore entrance chamfer	Max. 30˚C
Housing bore chamfer grinding undercut	Min. 0.006D
Housing bore finish roughness	2.5 μm Ra
Housing bore roundness	1/2 bearing housing dimension tolerance

(3) Allowable temperature range

-25 ～120˚C



Open type | Shielded type (ZZ) | Non-contact sealed type (LLB, LLF) | Low torque sealed type (LLH) | Contact sealed type (LLU)

II

d 10～20mm

Boundary dimensions (mm) *d*	*D*	*B*	$r_{s\,min}$ 1)	r_{NS} min	Basic load ratings dynamic (kN) C_r	static (kN) C_{or}	dynamic (kgf) C_r	static (kgf) C_{or}	Factor f_o	Limiting speeds (min⁻¹) grease open type ZZ LLB	oil open type Z LB	LLH	LLU	Bearing numbers open type	shielded type	non-contact sealed type	low torque sealed type	contact sealed type
10	15	3	0.1	—	0.855	0.435	87	44	15.7	10 000	12 000	—	—	6700	—	—	—	—
	19	5	0.3	—	1.83	0.925	187	94	14.8	32 000	38 000	—	24 000	6800	ZZ	LLB	—	LLU
	22	6	0.3	0.3	2.7	1.27	275	129	14.0	30 000	36 000	—	21 000	6900	ZZ	LLB	—	LLU
	26	8	0.3	—	4.55	1.96	465	200	12.4	29 000	34 000	25 000	21 000	6000	ZZ	LLB	LLH	LLU
	30	9	0.6	0.5	5.10	2.39	520	244	13.2	25 000	30 000	21 000	18 000	6200	ZZ	LLB	LLH	LLU
	35	11	0.6	0.5	8.20	3.50	835	355	11.4	23 000	27 000	20 000	16 000	6300	ZZ	LLB	LLH	LLU
12	18	4	0.2	—	0.930	0.530	95	54	16.2	8 300	9 500	—	—	6701	—	LLF	—	—
	21	5	0.3	—	1.92	1.04	195	106	15.3	29 000	35 000	—	20 000	6801	ZZ	LLB	—	LLU
	24	6	0.3	0.3	2.89	1.46	295	149	14.5	27 000	32 000	—	19 000	6901	ZZ	LLB	—	LLU
	28	7	0.3	—	5.10	2.39	520	244	13.2	26 000	30 000	—	—	16001	—	—	—	—
	28	8	0.3	—	5.10	2.39	520	244	13.2	26 000	30 000	21 000	18 000	6001	ZZ	LLB	LLH	LLU
	32	10	0.6	0.5	6.10	2.75	620	280	12.7	22 000	26 000	20 000	16 000	6201	ZZ	LLB	LLH	LLU
	37	12	1	0.5	9.70	4.20	990	425	11.1	20 000	24 000	19 000	15 000	6301	ZZ	LLB	LLH	LLU
15	21	4	0.2	—	0.940	0.585	96	59	16.5	6 600	7 600	—	—	6702	—	LLF	—	—
	24	5	0.3	—	2.08	1.26	212	128	15.8	26 000	31 000	—	17 000	6802	ZZ	LLB	—	LLU
	28	7	0.3	0.3	3.65	2.00	375	204	14.8	24 000	28 000	—	16 000	6902	ZZ	LLB	—	LLU
	32	8	0.3	—	5.60	2.83	570	289	13.9	22 000	26 000	—	—	16002	—	—	—	—
	32	9	0.3	0.3	5.60	2.83	570	289	13.9	22 000	26 000	18 000	15 000	6002	ZZ	LLB	LLH	LLU
	35	11	0.6	0.5	7.75	3.60	790	365	12.7	19 000	23 000	18 000	15 000	6202	ZZ	LLB	LLH	LLU
	42	13	1	0.5	11.4	5.45	1 170	555	12.3	17 000	21 000	15 000	12 000	6302	ZZ	LLB	LLH	LLU
17	23	4	0.2	—	1.00	0.660	102	67	16.3	5 000	6 700	—	—	6703	—	LLF	—	—
	26	5	0.3	—	2.23	1.46	227	149	16.1	24 000	28 000	—	15 000	6803	ZZ	LLB	—	LLU
	30	7	0.3	0.3	4.65	2.58	475	263	14.7	22 000	26 000	—	14 000	6903	ZZ	LLB	—	LLU
	35	8	0.3	—	6.80	3.35	695	345	13.6	20 000	24 000	—	—	16003	—	—	—	—
	35	10	0.3	0.3	6.80	3.35	695	345	13.6	20 000	24 000	16 000	14 000	6003	ZZ	LLB	LLH	LLU
	40	12	0.6	0.5	9.60	4.60	980	465	12.8	18 000	21 000	15 000	12 000	6203	ZZ	LLB	LLH	LLU
	47	14	1	0.5	13.5	6.55	1 380	665	12.2	16 000	19 000	14 000	11 000	6303	ZZ	LLB	LLH	LLU
	62	17	1.1	—	22.7	10.8	2 320	1 100	11.1	14 000	16 000	—	—	6403	—	—	—	—
20	27	4	0.2	—	1.04	0.730	106	74	16.1	5 000	5 700	—	—	6704	—	LLF	—	—
	32	7	0.3	0.3	4.00	2.47	410	252	15.5	21 000	25 000	—	13 000	6804	ZZ	LLB	—	LLU
	37	9	0.3	0.3	6.40	3.70	650	375	14.7	19 000	23 000	—	12 000	6904	ZZ	LLB	—	LLU
	42	8	0.3	—	7.90	4.50	810	455	14.5	18 000	21 000	—	—	16004	—	—	—	—
	42	12	0.6	0.5	9.40	5.05	955	515	13.9	18 000	21 000	13 000	11 000	6004	ZZ	LLB	LLH	LLU
	47	14	1	0.5	12.8	6.65	1 310	680	13.2	16 000	18 000	12 000	10 000	6204	ZZ	LLB	LLH	LLU
	52	15	1.1	0.5	15.9	7.90	1 620	805	12.4	14 000	17 000	12 000	10 000	6304	ZZ	LLB	LLH	LLU

1) Smallest allowable dimension for chamfer dimension *r*.

				
Angular contact ball bearing	High speed use angular contact ball bearing	Ultra high speed use angular contact ball bearing	Four-point contact ball bearing	Double row angular contact ball bearing

1. Design features and special characteristics

1.1 Angular contact ball bearing

Angular contact ball bearings are non-separable bearings which have a certain contact angle in the radial direction relative to the straight line that runs through the point where each ball makes contact with the inner and outer rings (see **Diagram 1**). **Table 1** gives contact angle and contact angle symbol.

In addition to radial loads, single direction axial loads can also be accommodated by angular contact ball bearings.

Furthermore, since an axial load is generated from a radial force, these bearings are generally used in pairs facing each other. Standard type, high speed use type and ultra high speed varieties of angular contact ball bearings are available through NTN, and there are also many duplex varieties. A bearing accuracy of JIS Class 5 or higher is applied to duplex type angular contact ball bearings, and in many cases they are given a preload, in compliance with standard preload levels, before being used in an application. **Table 2** shows information concerning angular contact ball bearings, and **Table 3** shows similar information for duplex angular contact ball bearings.

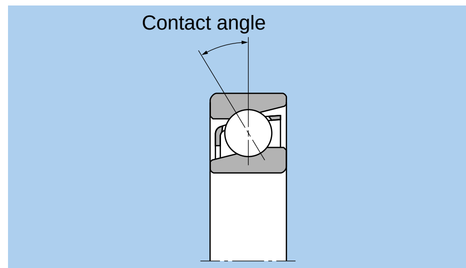


Diagram 1.

Table 1 Contact angle and contact angle codes

Contact angle	15˚	30˚	40˚
Code	C	A❶	B

❶ Contact angle symbol A is omitted.

Table 2 Angular contact ball bearing types and characteristics

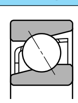
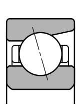
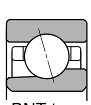
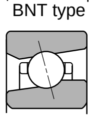
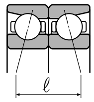
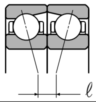
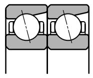
Type	Design	Characteristics
Standard type		● Available in bearing series 79, 70, 72, 72B, 73, and 73B. ● Contact angles: 30˚and 40˚ (with B) available. ● Standard bearing cage type differs depending on bearing no. (Refer to **Table 4**)
High speed use		● Available in bearing series 78C, 79C, 70C, 72C, and 73C. ● Contact angles: 15˚ ● All bearing accuracies JIS Class 5 or higher. ● Standard bearing cage type differs depending on bearing no. (Refer to **Table 4**)
Ultra high speed use	 BNT type  HSB type	● Available in bearing series HSB9C, HSB0C, BNT0, and BNT2; all boundary dimensions agree with JIS series dimensions. ● Contact angles: 15˚; HSB type HSB9 and HSB0: 15˚and 30˚. ● All bearing accuracies JIS Class 5 or higher. ● BNT type internal design can be altered; suitable for higher speed applications than high speed use bearings. ● HSB series bearings have smaller diameter of balls than high speed use type bearings, so benefit by less torque for high precision, high speed applications. ● The inner ring bore diameter and outer ring inner diameter of the HSB series have a ground undercut on one side enabling easy oil flow. ● For even higher speed applications, there is a bearing in this series equipped with ceramic ball bearings. ● For standard cage types refer to **Table 4**; molded resin cages are also available for some varieties.

Table 3 Duplex angular contact ball bearings — types and characteristics

Duplex type		Characteristics
Back-to-back duplex (DB)		● Can accommodate radial loads and axial loads in either direction. ● Has a large distance ℓ between the acting load center of the bearing, and therefore a large momentary force load capacity. ● Allowable misalignment angle is small.
Face-to face duplex (DF)		● Can accommodate radial loads and axial loads in either direction. ● Has a smaller distance ℓ between the acting load center of the bearing, and therefore a smaller momentary force load capacity. ● Has a larger allowable misalignment angle than back-to-back duplex type.
Tandem duplex (DT)		● Can accommodate radial loads and single direction axial loads. ● Axial loads are received by both bearings as a set, and therefore heavy axial loads can be accommodated.

II

Note: 1. Duplex bearings are manufactured in a set to specified clearance and preload values, therefore they must be assembled together with identically numbered bearings and not mixed with other arrangements.
2. Triplex arrangements of angular contact bearings are also available. Consult NTN Engineering for details.

1.2 Four-point angular contact ball bearings

Four-point angular contact ball bearings have a contact angle of 30˚ and inner rings which are separated in half. As shown in **Diagram 2**, when the inner and outer rings receive a radial load the ball bearings contact the inner and outer rings at four points. This construction enables a single bearing to accommodate axial loads from either direction, and when generally under a simple axial load or heavy axial load, the bearing functions in reliance on two contact points like ordinary bearings.

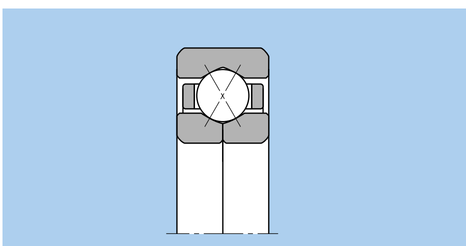


Diagram 2.

1.3 Double row angular contact ball bearings

The structure of double row angular contact ball bearings is designed by arranging two single row angular contact bearings back-to-back in duplex (DB) to form one united bearing with a contact angle of 25˚.

These bearings are capable of accommodating radial loads, axial loads in either direction, and have a high capacity for momentary loads as well.

As shown in **Diagram 3**, sealed and shielded type double row angular contact ball bearings are also available. Standard loads vary from those of open type bearings.

■Flush ground

"Flush ground" is the name given to the finishing method shown in **Diagram 4** where the offset of the front and back faces of the bearing are ground to the same value. By doing this, a stated clearance or preload value can be achieved by using bearings with identical codes for these values, in other words by combining either DB or DF series bearings. DT series bearings can also be used in various arrangements to achieve uniform load distribution.

All BNT type bearings are flush ground, but other angular contact ball bearing series are not. If it is necessary to flush grind any of these other bearings, please consult NTN Engineering.

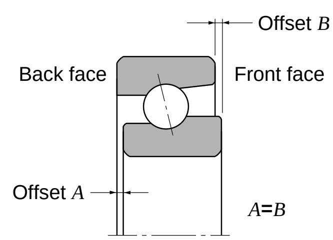


Diagram 4.

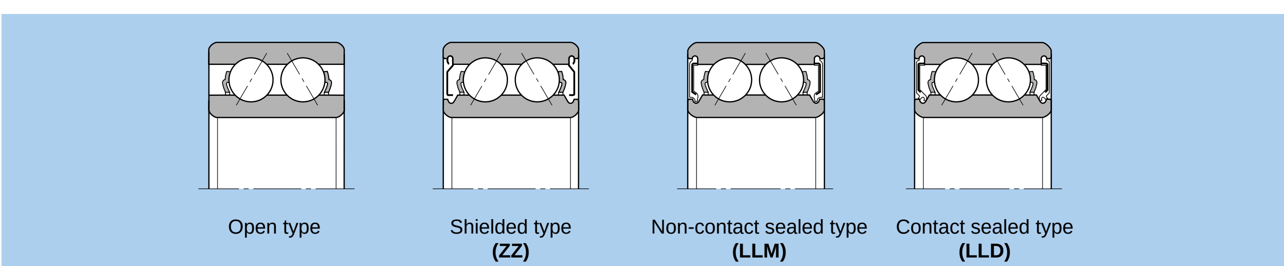


Diagram 3.

II

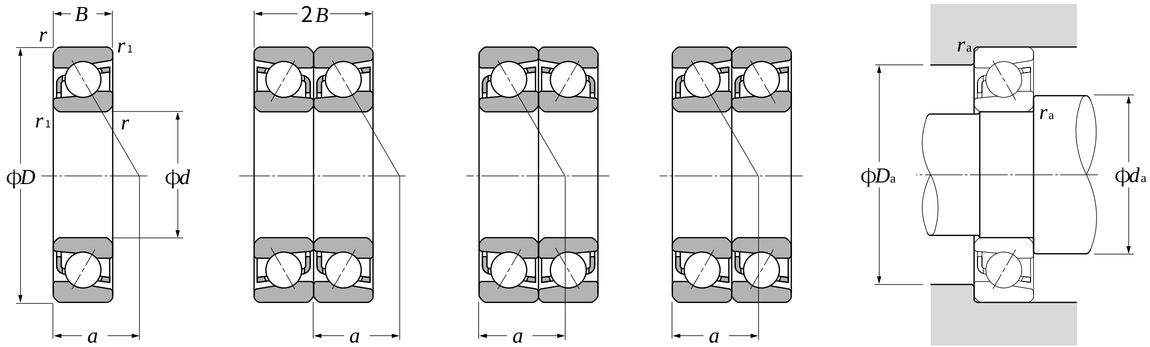
2. Standard cage types

Table 4 lists the standard cage types for angular contact ball bearings. For high speed use angular contact ball bearings, molded resin cages and machined cages are widely used.

Table 4 Standard cages for angular contact ball bearings

Type	Bearing series	Molded resin cage	Pressed cage	Machined cage
Standard	79 70 72 73 72B 73B	7904～7913 7000～7024 — — — —	— — 7200 ～7222 7300 ～7322 7200B～7222B 7300B～7322B	7914 ～7960 7026 ～7040 7224 ～7240 7324 ～7340 7224B ～7240B 7324B ～7340B
High speed use	78C 79C 70C 72C 73C	— 7904C～7913C 7000C～7024C 7200C～7220C 7303C～7312C	— — — — —	7805C ～7834C 7914C ～7934C 7026C ～7040C 7221C ～7240C 7300C ～7302C 7313C ～7340C
Ultra high speed use	BNT0 BNT2 HSB9C HSB0C	— — — HSB010C～HSB032C	— — — —	BNT000 ～BNT009 BNT200 ～BNT209 HSB910C～HSB934C HSB034C
4-point contact	QJ2 QJ3	— —	— —	QJ208～QJ224 QJ306～QJ324
Double row	52 53	— —	5200S～5217S 5302S～5314S	— —

Note: 1. Standard cages for 5S-BNT and 5S-HSB type bearings are the same as cages for BNT and HSB type bearings.
2. Due to the material characteristics of molded resin cages, use at application temperatures in excess of 120˚C is not possible.



Single | Back-to-back arrangement (DB) | Face-to-face arrangement (DF) | Tandem arrangement (DT)

d 10～30mm

Boundary dimensions mm						Basic load ratings dynamic kN	static kN	dynamic kgf	static kgf	Limiting speeds[1] min⁻¹		Bearing[2] numbers	Load center mm	Mass kg single (approx.)
d	*D*	*B*	2*B*	$r_{s\,min}$[3]	$r_{1s\,min}$[3]	C_r	C_{or}	C_r	C_{or}	grease	oil		*a*	
10	26	8	16	0.3	0.15	4.65	2.07	470	212	29 000	39 000	**7000**	9	0.023
	30	9	18	0.6	0.3	5.45	2.74	555	279	28 000	37 000	**7200**	10.5	0.029
	30	9	18	0.6	0.3	5.00	2.52	510	257	24 000	32 000	**7200B**	13	0.029
	35	11	22	0.6	0.3	10.1	4.95	1 030	500	26 000	34 000	**7300**	12	0.04
	35	11	22	0.6	0.3	9.50	4.60	970	470	22 000	29 000	**7300B**	15	0.041
12	28	8	16	0.3	0.15	5.05	2.46	515	251	26 000	35 000	**7001**	10	0.025
	32	10	20	0.6	0.3	7.60	3.95	775	405	25 000	33 000	**7201**	11.5	0.035
	32	10	20	0.6	0.3	7.00	3.65	775	405	21 000	28 000	**7201B**	14	0.036
	37	12	24	1	0.6	11.2	5.25	1 140	535	23 000	30 000	**7301**	13	0.044
	37	12	24	1	0.6	10.5	4.95	1 080	505	19 000	26 000	**7301B**	16.5	0.045
15	32	9	18	0.3	0.15	5.80	3.15	590	320	23 000	31 000	**7002**	11.5	0.035
	35	11	22	0.6	0.3	9.05	4.70	925	480	22 000	29 000	**7202**	12.5	0.046
	35	11	22	0.6	0.3	8.35	4.35	855	445	18 000	25 000	**7202B**	16	0.046
	42	13	26	1	0.6	13.5	7.20	1 370	735	19 000	26 000	**7302**	15	0.055
	42	13	26	1	0.6	12.5	6.65	1 270	680	17 000	22 000	**7302B**	19	0.057
17	35	10	20	0.3	0.15	7.15	3.85	730	390	21 000	28 000	**7003**	12.5	0.046
	40	12	24	0.6	0.3	12.0	6.60	1 220	675	19 000	26 000	**7203**	14.5	0.064
	40	12	24	0.6	0.3	11.0	6.10	1 120	625	17 000	22 000	**7203B**	18	0.066
	47	14	28	1	0.6	15.9	8.65	1 630	880	18 000	24 000	**7303**	16	0.107
	47	14	28	1	0.6	14.8	8.00	1 510	820	15 000	20 000	**7303B**	20.5	0.109
20	42	12	24	0.6	0.3	9.70	5.60	990	570	19 000	25 000	**7004**	15	0.08
	47	14	28	1	0.6	14.5	8.40	1 480	855	17 000	23 000	**7204**	17	0.1
	47	14	28	1	0.6	13.3	7.70	1 360	785	15 000	20 000	**7204B**	21.5	0.102
	52	15	30	1.1	0.6	18.7	10.4	1 910	1 060	16 000	21 000	**7304**	18	0.138
	52	15	30	1.1	0.6	17.3	9.65	1 770	985	13 000	18 000	**7304B**	22.5	0.141
25	42	9	18	0.3	0.15	7.15	4.95	730	505	17 000	22 000	**7905**	14	0.05
	47	12	24	0.6	0.3	10.7	6.85	1 100	700	16 000	21 000	**7005**	16.5	0.093
	52	15	30	1	0.6	16.2	10.3	1 650	1 050	14 000	19 000	**7205**	19	0.125
	52	15	30	1	0.6	14.8	9.40	1 510	960	12 000	16 000	**7205B**	24	0.129
	62	17	34	1.1	0.6	26.4	15.8	2 690	1 610	13 000	17 000	**7305**	21	0.23
	62	17	34	1.1	0.6	24.4	14.6	2 490	1 490	11 000	15 000	**7305B**	27	0.234
30	47	9	18	0.3	0.15	7.55	5.75	770	585	14 000	19 000	**7906**	15.5	0.058
	55	13	26	1	0.6	13.9	9.45	1 410	965	13 000	18 000	**7006**	19	0.135

1）This value achieved with machined cages; when pressed cages are used, 80% of this value is acceptable.
2）Bearing numbers appended with the code "**B**" have a contact angle of 40˚; bearings with this code have a contact angle of 30˚.
3）Smallest allowable dimension for chamfer dimension *r*.

1. Design features and characteristics

The outer ring raceway of self-aligning ball bearings forms a spherical surface whose center is common to the bearing center. The inner ring of the bearing has two raceways. The balls, cage, and inner ring of these bearings are capable of a shifting in order to compensate for a certain degree of misalignment with the outer rings. As a result, the bearing is able to align itself and compensate for shaft / housing finishing unevenness, bearing fitting error, and other sources of misalignment as shown in **Diagram 1**.

However, since axial load capacity is limited, self-aligning ball bearings are not suitable for applications with heavy axial loads.

Furthermore, if an adapter is used on the tapered bore of the inner diameter, installation and disassembly are much simpler and for this reason adapters are often used on equipment with drive shafts.

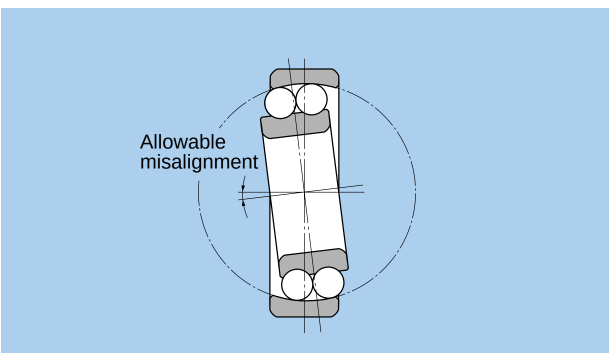


Diagram 1.

2. Standard cage types

All bearing series are equipped with a pressed cage, except 2322S, which is equipped with a machined cage.

3. Ball protrusion

Bearings with part numbers listed in **Diagram 2** below have balls which protrude slightly from the bearing face.

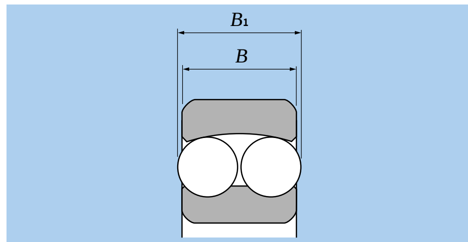


Diagram 2.

their degree of protrusion is listed below

Units mm

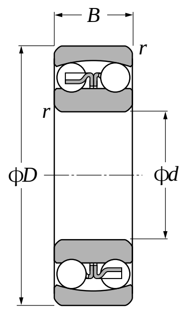
Bearing number	Width dimension B	Total width dimension B_1
2222S (K)	53	54
2316S (K)	58	59
2319S (K)	67	68
2320S (K)	73	74
2321S	77	78
2322S (K)	80	81
1318S (K)	43	46
1319S (K)	45	49
1320S (K)	47	53
1321S	49	55
1322S (K)	50	56

4. Allowable misalignment angle

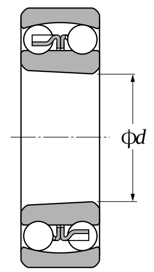
Listed below are the allowable misalignment angles for bearings with self-aligning characteristics when placed under normal load conditions. This degree of allowable misalignment may be limited by the design of structures around the bearing.

Allowable misalignment under normal loads (loads equivalent to 0.09 C_r): 0.07 rad (4˚)

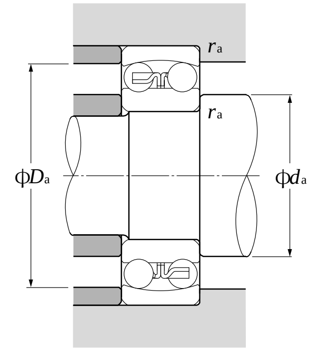
Self-Aligning Ball Bearings



Cylindrical bore



Tapered bore



II

d 10～35mm

Boundary dimensions mm				Basic load ratings dynamic kN	static kN	dynamic kgf	static kgf	Limiting speeds min⁻¹		Bearing numbers		Abutment and fillet dimensions mm		
d	D	B	$r_{s\,min}$ 1)	C_r	C_{or}	C_r	C_{or}	grease	oil	cylindrical bore	tapered 2) bore	d_a min	D_a max	r_{as} max
10	30	9	0.6	5.55	1.19	570	121	22 000	28 000	**1200S**	—	14.0	26.0	0.6
	30	14	0.6	7.45	1.59	760	162	24 000	28 000	**2200S**	—	14.0	26.0	0.6
	35	11	0.6	7.35	1.62	750	165	20 000	24 000	**1300S**	—	14.0	31.0	0.6
	35	17	0.6	9.20	2.01	935	205	18 000	22 000	**2300S**	—	14.0	31.0	0.6
12	32	10	0.6	5.70	1.27	580	130	22 000	26 000	**1201S**	—	16.0	28.0	0.6
	32	14	0.6	7.75	1.73	790	177	22 000	26 000	**2201S**	—	16.0	28.0	0.6
	37	12	1	9.65	2.16	985	221	18 000	22 000	**1301S**	—	17.0	32.0	1
	37	17	1	12.1	2.73	1 240	278	17 000	22 000	**2301S**	—	17.0	32.0	1
15	35	11	0.6	7.60	1.75	775	179	18 000	22 000	**1202S**	—	19.0	31.0	0.6
	35	14	0.6	7.80	1.85	795	188	18 000	22 000	**2202S**	—	19.0	31.0	0.6
	42	13	1	9.70	2.29	990	234	16 000	20 000	**1302S**	—	20.0	37.0	1
	42	17	1	12.3	2.91	1 250	296	14 000	18 000	**2302S**	—	20.0	37.0	1
17	40	12	0.6	8.00	2.01	815	205	16 000	20 000	**1203S**	—	21.0	36.0	0.6
	40	16	0.6	9.95	2.42	1 010	247	16 000	20 000	**2203S**	—	21.0	36.0	0.6
	47	14	1	12.7	3.20	1 300	325	14 000	17 000	**1303S**	—	22.0	42.0	1
	47	19	1	14.7	3.55	1 500	365	13 000	16 000	**2303S**	—	22.0	42.0	1
20	47	14	1	10.0	2.61	1 020	266	14 000	17 000	**1204S**	**1204SK**	25.0	42.0	1
	47	18	1	12.8	3.30	1 310	340	14 000	17 000	**2204S**	**2204SK**	25.0	42.0	1
	52	15	1.1	12.6	3.35	1 280	340	12 000	15 000	**1304S**	**1304SK**	26.5	45.5	1
	52	21	1.1	18.5	4.70	1 880	480	11 000	14 000	**2304S**	**2304SK**	26.5	45.5	1
25	52	15	1	12.2	3.30	1 250	335	12 000	14 000	**1205S**	**1205SK**	30.0	47.0	1
	52	18	1	12.4	3.45	1 270	350	12 000	14 000	**2205S**	**2205SK**	30.0	47.0	1
	62	17	1.1	18.2	5.00	1 850	510	10 000	13 000	**1305S**	**1305SK**	31.5	55.5	1
	62	24	1.1	24.9	6.60	2 530	675	9 500	12 000	**2305S**	**2305SK**	31.5	55.5	1
30	62	16	1	15.8	4.65	1 610	475	10 000	12 000	**1206S**	**1206SK**	35.0	57.0	1
	62	20	1	15.3	4.55	1 560	460	10 000	12 000	**2206S**	**2206SK**	35.0	57.0	1
	72	19	1.1	21.4	6.30	2 190	645	8 500	11 000	**1306S**	**1306SK**	36.5	65.5	1
	72	27	1.1	32.0	8.75	3 250	895	8 000	10 000	**2306S**	**2306SK**	36.5	65.5	1
35	72	17	1.1	15.9	5.10	1 620	520	8 500	10 000	**1207S**	**1207SK**	41.5	65.5	1
	72	23	1.1	21.7	6.60	2 210	675	8 500	10 000	**2207S**	**2207SK**	41.5	65.5	1
	80	21	1.5	25.3	7.85	2 580	800	7 500	9 500	**1307S**	**1307SK**	43.0	72.0	1.5
	80	31	1.5	40.0	11.3	4 100	1 150	7 100	9 000	**2307S**	**2307SK**	43.0	72.0	1.5

1) Smallest allowable dimension for chamfer dimension r. 2) "**K**" indicates bearings have tapered bore with a taper ratio of 1: 12.


Cylindrical roller bearing


E Type cylindrical roller bearing


Double row cylindrical roller bearing


Four row cylindrical roller bearing

1. Types, design features, and characteristics

Since the rolling elements in cylindrical roller bearings make line contact with raceways, these bearings can accommodate heavy radial loads. The rollers are guided by ribs on either the inner or outer ring, therefore these bearings are also suitable for high speed applications. Furthermore, cylindrical roller bearings are separable, and relatively easy to install and disassemble even when interference fits are required.

Among the various types of cylindrical roller bearings, Type E has a high load capacity and its boundary dimensions are identical to standard type. HT type has a large axial load capacity, and HL type provides extended wear life in conditions where the development of a lubricating film inside the bearing is difficult.

Double and multiple row bearing arrangements are also available.

For extremely heavy load applications, the non-separable full complement SL type bearing offers special advantages.

Table 1 shows the various types and characteristics of single row cylindrical roller bearings. **Table 2** shows the characteristics of non-standard type cylindrical roller bearings.

Table 1 Cylindrical roller bearing types and characteristics

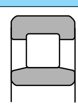
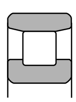
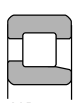
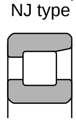
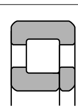
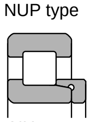
Type code	Design	Characteristics
NU type **N type**	 NU type  N type	● NU type outer rings have double ribs; outer ring and roller as well as cage can be separated from inner ring. N type inner ring have double ribs; inner ring and roller as well as cage can be separated from outer ring. ● Unable to accommodate even the lightest axial loads. ● This type is extremely suitable for, and widely used as, the floating side bearing.
NJ type **NF type**	 NJ type  NF type	● NJ type has double ribs on outer ring, single rib on inner ring; NF type has single rib on outer ring, and double rib on inner ring. ● Can receive single direction axial loads. ● When there is no distinction between the fixed side and floating side bearing, can be used as a pair in close proximity.
NUP type **NH type (NJ + HJ)**	 NUP type  NH type	● NUP type has a collar ring attached to the ribless side of the inner ring; NH type is NJ type with an L type collar ring attached. All of these collar rings are separable, and therefore it is necessary to fix the inner ring axially. ● Can accommodate axial loads in either direction. ● Widely used as the shaft's fixed-side bearing.

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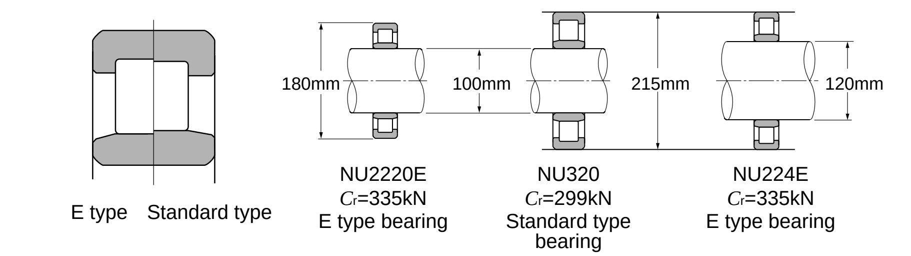
Table 2 Non-standard type cylindrical roller bearing characteristics

Bearing type	Characteristics
E Type cylindrical roller bearing	● Boundary dimensions are the same as the standard type, but the diameter, length and number of the rollers have been increased, as well as load capacity. ● Identified by addition of "E" to end of basic roller number. ● Enables compact design due to its high load rating. ● Rollers' inscribed circle diameter differs from standard type rollers and therefore cannot be interchanged.  Remarks: In the dimension tables, both E type and standard type are listed, but in the future JIS will change to E type.
Large axial load use cylindrical roller bearings (HT type)	● Can accommodate larger axial loads than standard type thanks to improved geometry of the rib roller end surface. ● Please consult NTN Engineering concerning the many factors which require consideration, such as load, lubricant, and installation conditions.
Double row cylindrical roller bearings	● NN type and NNU type available. ● Widely used for applications requiring thin-walled bearings, such the main shafts of machine tools, rolling machine rollers, and in printing equipment. ● Internal radial clearance is adjusted for the spindle of machine tools by pressing the tapered bore of the inner ring on a tapered shaft.
Four row cylindrical roller bearings	● Used mainly in the necks of rolling machine rollers; designed for maximum rated load to accommodate the severely limited space in the roller neck section of such equipment. ● Many varieties exist, including sealed types, which have been specially designed for high speed use, to prevent creeping, provide dust and water proofing properties, etc. Contact NTN Engineering.
SL type cylindrical roller bearings	● Full complement roller bearing capable of handling heavy loads. ● Consult NTN Engineering regarding special application designs for SL type cylindrical roller bearings.

2. Standard cage types

Table 3 shows the standard varieties for cylindrical roller bearings.

Table 3 Standard cage types

Bearing series	Molded resin cage	Pressed cage	Machined cage
NU10	—	—	1005〜10/500
NU 2 **NU2E**	— 204E〜218E	208〜230 —	232〜264 219E〜240E
NU22 **NU22E**	— 2204E〜2218E	2208〜2230 —	2232〜2264 2219E〜2240E
NU3 **NU3E**	— 304E〜314E	308〜324 —	326〜356 315E〜332E
NU23 **NU23E**	— 2304E〜2311E	2308〜2320 —	2322〜2356 2312E〜2332E
NU4	—	405〜416	—

The basic road ratings listed in the dimension charts correspond to values achieved with the standard cages listed in **Table 3**. Furthermore, please note that even for the identical bearing, in cases where the number of rolling elements or the cage type differs, the basic rated load will also differ from the values listed in the dimension charts.

Note: 1) Within the same bearing series, cage type is identical even if the type code (NJ, NUP, N, NF) differs.
2) For high speed and other special applications, machined cages can be manufactured when necessary.
Consult NTN Engineering.
3) Among E type bearings (those using molded resin cages), certain varieties may also use pressed cages.
Consult **NTN** Engineering.
4) Although machined cages are standard for two row and four row cylindrical roller bearings, molded resin cages may also be used in some of these bearings for machine tool applications.
5) **Due to their material properties, molded resin cages cannot be used in applications where temperatures exceed 120˚C. #04 - #07 however use resin material with superior ability to withstand heat and high temperatures, which are capable of withstanding temperatures up to 150˚C.**
6) Formed resin cages capable of withstanding temperatures up to 150˚C can be manufactured by request for type E (formed resin cage) of #08 or greater. For information, please contact NTN Engineering.

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3. Allowable misalignment

Although values vary somewhat depending on bearing type and internal specifications, under general load conditions, to avoid the occurrence of edge loading, allowable misalignments have been set as follows:

Bearing width series 0 or 1:	0.001 rad (3.5')
Bearing width series 2:	0.0005 rad (1.5')
Double row cylindrical roller bearings ❶ :	0.0005 rad (1.5')

❶ Does not include high precision bearings for machine tool main shaft applications.

4. Allowable axial load for cylindrical roller bearings

Cylindrical roller bearings with ribs on the inner and outer rings are capable of simultaneously bearing a radial load and an axial load of a certain degree. Unlike basic road ratings based on rolling fatigue, allowable axial load is determined by heat produced on the sliding surface between the ends of the rollers and rib, seizure and wear. Allowable axial load when center axial load is applied is approximately determined by formula (1), which is based upon experience and testing.

$$P_t = k \cdot d^2 \cdot P_z \quad \cdots\cdots (1)$$

Where:

P_t : Allowable axial load when rotating N {kgf}

k : Factor determined by internal design of bearing (see **Table 4**)

d : Bearing bore mm

P_z : Allowable surface pressure of rib MPa {kgf/mm^2} (see **Diagram 1**)

If axial load is greater than radial load, the rollers will not rotate properly. The allowable axial load therefore must not exceed the value for $F_{a\,max}$ given in **Table 4**.

The following are also important to operate the bearing smoothly under axial load:

(1) Do not make the internal radial clearance any larger than necessary.

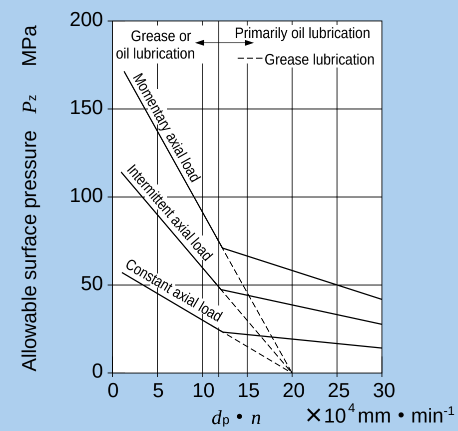
(2) Use lubricant with extreme pressure additive.

(3) Make the shoulder of the housing and shaft high enough for the rib of the bearing.

(4) If the bearing is to support an extreme axial load, mounting precision should be improved and the bearing should rotate slowly before actual use.

If large cylindrical roller bearings (bore of 300 mm or more) are to support an axial load or moment load simultaneously, please contact NTN Engineering.

NTN Engineering also offers cylindrical roller bearings for high axial loads (HT type). For details, please contact NTN Engineering.

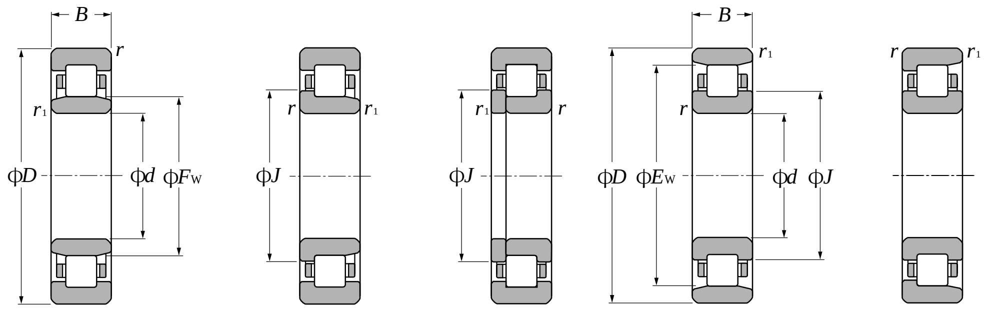


d_p : Roller pitch diameter mm
d_p (bearing bore + bearing outside diameter)/2 mm
n : Rotational speed min^{-1}

Diagram 1 Allowable surface pressure of rib

Table 4 Factor k values and allowable axial load ($F_{a\,max}$)

Bearing series	k	$F_{a\,max}$
NJ, NUP10 NJ, NUP, NF, NH2, NJ, NUP, NH22	0.040	0.4F_r
NJ, NUP, NF, NH3, NJ, NUP, NH23	0.065	0.4F_r
NJ, NUP, NH2E, NJ, NUP, NH22E	0.050	0.4F_r
NJ, NUP, NH3E, NJ, NUP, NH23E	0.080	0.4F_r
NJ, NUP, NH4,	0.100	0.4F_r
SL01-48	0.022	0.2F_r
SL01-49	0.034	0.2F_r
SL04-50	0.044	0.2F_r



Type NU Type NJ Type NUP Type N Type NF

d 20～40mm

Boundary dimensions mm					Basic load ratings dynamic kN	static kN	dynamic kgf	static kgf	Limiting speeds[1] $\min^{-1}$		Bearing numbers[2]			
d	D	B	$r_{s\,min}$[3]	$r_{1s\,min}$[3]	C_r	C_{or}	C_r	C_{or}	grease	oil	type NU	type NJ	type NUP	type N
20	47	14	1	0.6	25.7	22.6	2 620	2 310	15 000	18 000	NU204E	NJ	NUP	—
	47	18	1	0.6	30.5	28.3	3 100	2 890	14 000	16 000	NU2204E	NJ	NUP	—
	52	15	1.1	0.6	31.5	26.9	3 200	2 740	13 000	15 000	NU304E	NJ	NUP	—
	52	21	1.1	0.6	42.0	39.0	4 300	3 950	12 000	14 000	NU2304E	NJ	NUP	—
25	47	12	0.6	0.3	15.1	14.1	1 540	1 430	16 000	19 000	NU1005	NJ	NUP	N
	52	15	1	0.6	29.3	27.7	2 990	2 830	13 000	15 000	NU205E	NJ	NUP	—
	52	18	1	0.6	35.0	34.5	3 550	3 550	11 000	13 000	NU2205E	NJ	NUP	—
	62	17	1.1	1.1	41.5	37.5	4 250	3 800	11 000	13 000	NU305E	NJ	NUP	—
	62	24	1.1	1.1	57.0	56.0	5 800	5 700	9 700	11 000	NU2305E	NJ	NUP	—
	80	21	1.5	1.5	46.5	40.0	4 750	4 050	8 500	10 000	NU405	NJ	NUP	N
30	55	13	1	0.6	19.7	19.6	2 000	2 000	14 000	16 000	NU1006	NJ	NUP	N
	62	16	1	0.6	39.0	37.5	4 000	3 800	11 000	13 000	NU206E	NJ	NUP	—
	62	20	1	0.6	49.0	50.0	5 000	5 100	9 700	11 000	NU2206E	NJ	NUP	—
	72	19	1.1	1.1	53.0	50.0	5 400	5 100	9 300	11 000	NU306E	NJ	NUP	—
	72	27	1.1	1.1	74.5	77.5	7 600	7 900	8 300	9 700	NU2306E	NJ	NUP	—
	90	23	1.5	1.5	62.5	55.0	6 400	5 600	7 300	8 500	NU406	NJ	NUP	N
35	62	14	1	0.6	22.6	23.2	2 310	2 360	12 000	15 000	NU1007	NJ	NUP	N
	72	17	1.1	0.6	50.5	50.0	5 150	5 100	9 500	11 000	NU207E	NJ	NUP	—
	72	23	1.1	0.6	61.5	65.5	6 300	6 650	8 500	10 000	NU2207E	NJ	NUP	—
	80	21	1.5	1.1	71.0	71.0	7 200	7 200	8 100	9 600	NU307E	NJ	NUP	—
	80	31	1.5	1.1	99.0	109	10 100	11 100	7 200	8 500	NU2307E	NJ	NUP	—
	100	25	1.5	1.5	75.5	69.0	7 700	7 050	6 400	7 500	NU407	NJ	NUP	N
40	68	15	1	0.6	27.3	29.0	2 780	2 950	11 000	13 000	NU1008	NJ	NUP	N
	80	18	1.1	1.1	43.5	43.0	4 450	4 350	9 400	11 000	NU208	NJ	NUP	N
	80	18	1.1	1.1	55.5	55.5	5 700	5 650	8 500	10 000	NU208E	NJ	NUP	—
	80	23	1.1	1.1	58.0	62.0	5 950	6 300	8 500	10 000	NU2208	NJ	NUP	N
	80	23	1.1	1.1	72.5	77.5	7 400	7 900	7 600	8 900	NU2208E	NJ	NUP	—
	90	23	1.5	1.5	58.5	57.0	6 000	5 800	8 000	9 400	NU308	NJ	NUP	N
	90	23	1.5	1.5	83.0	81.5	8 500	8 300	7 200	8 500	NU308E	NJ	NUP	—
	90	33	1.5	1.5	82.5	88.0	8 400	8 950	7 000	8 200	NU2308	NJ	NUP	N
	90	33	1.5	1.5	114	122	11 600	12 500	6 400	7 500	NU2308E	NJ	NUP	—
	110	27	2	2	95.5	89.0	9 750	9 100	5 700	6 700	NU408	NJ	NUP	N

1) This value achieved with machined cages; when pressed cages are used, 80% of this value is acceptable.
2) Production switched to E type only for bearing number for which there is no standard form.
3) Minimal allowable dimension for chamfer dimension r or r_1.

Open type

Shielded type

Shielded type with snap ring

II

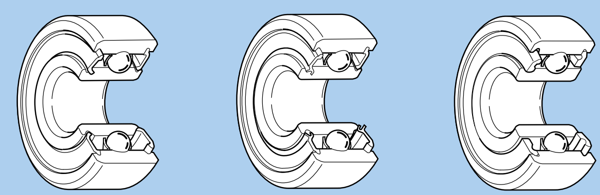
1. Design features and special characteristics

The dimensional range of miniature and extra small ball bearings is given in **Table 1**. Boundary dimensions for both metric and inch systems are in accordance with the internationally specified ISO and ANSI/ABMA standards. The most widely used sealed and shielded type ball bearings have a 1–2 mm wider width dimension than open type bearings.

The main variations of these bearings are shown in **Table 2**. Bearings with snap rings, which simplify the bearing housing construction and design, have also been serialized and are listed in dimension tables. Among the most generally used sealed and shielded bearings are standard ZZ and ZZA type which incorporate non-contact steel shield plates. **Diagram 1** also shows non-contact type rubber sealed LLB and resin sealed SSA type bearings, and the contact-type rubber sealed LLU bearing.

Table 1 Dimensional range

Bearing	Dimensional range
Miniature ball bearings	Nominal outer diameter D<9mm
Extra small ball bearings	Nominal bore diameter d<10mm Nominal outer diameter D≧9mm



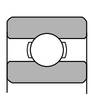
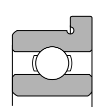
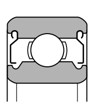
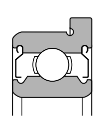
Rubber sealed (non-contact type) **LLB**

Rubber sealed (contact type) **LLU**

Resin sealed (non-contact type) **SSA**

Diagram 1.

Table 2 Main types and construction

Type	Standard type code			Flange-attached type code		
	Construction	Metric series	Inch series	Construction	Metric series	Inch series
Open type		6 BC	R		FL6 FLBC	FLR
Shielded type		6 x x ZZ W6 x x ZZ WBC x x x ZZ	RA x x ZZ		FL6 x x x ZZ FLW6 x x x ZZ FLWBC x x ZZ	FLRA x x ZZ

Note: 1. Representative type codes are shown. For further details, please refer to dimension tables.
2. May change to ZA or SA for shielded type bearings, according to the bearing number.

2. Standard cage types

Pressed cage are standard for these bearings. However, molded resin cage are used for some bearings depending on the application.

3. Dimensional and rotational accuracy

The accuracy of miniature and extra small ball bearings complies with JIS standards. Accuracy standards are listed in the Bearings Tolerances clause on page A-35. Flange accuracies are listed in **Table 3**.

Table 3 Tolerance and tolerance values for outer ring flange

Units μm

Accuracy class		Outer diameter dimensional tolerance Δ_{D1s} or Δ_{D2s}		Outer ring surface runout for rear surface S_{D1}	Back face axial runout S_{ea1}	Width dimension tolerance Δ_{C1s} or Δ_{C2s}		Width unevenness V_{C1s} or V_{C2s}
		Upper	Lower	Max.	Max.	Upper	Lower	Max.
ISO standard	Class 0	∗ (see table below)		—	—	Identical to same bearing's inner ring V_{Bs}		Identical to same bearing's inner ring V_{Bs}
	Class 6			—	—			
	Class 5			8	11			5
	Class 4			4	7			2.5
	Class 2			1.5	3 ❶ 4			1.5

❶ Nominal outer diameter, 18 mm or less.

∗ Units μm

Flange nominal outer diameter D_1 or D_2 mm		Outer diameter dimensional tolerance Δ_{D1s} or Δ_{D2s}	
over	incl.	Upper	Lower
—	10	+220	−36
10	18	+270	−43
18	30	+330	−52
30	50	+390	−62

4. Radial internal clearance

Radial internal clearance values should be applied as listed in the table regarding the Bearing Internal Clearance and Preload clause on page A-58.

However, for miniature and extra small bearings, the radial clearance values for high precision bearings given in **Table 4** are applied in many cases.

For more specific selection information, please refer to the NTN Miniature and Extra Small Ball Bearings Catalog, or contact NTN Engineering.

II

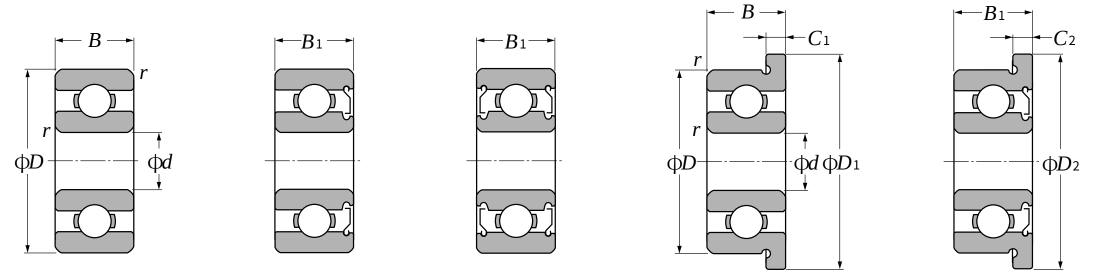
Table 4 Radial internal clearance for high precision bearings

Units μm

MIL Standard	Tight				Standard						Loose		Extra Loose	
Code	C2S		CNS		CNM		CNL		C3S		C3M		C3L	
Internal clearance	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
	0	5	3	8	5	10	8	13	10	15	13	20	20	28

Note: 1. These standards are specified in accordance with MIL B-23063. However, NTN codes are shown.
2. Clearance values do not include compensation for measuring load.

Metric series



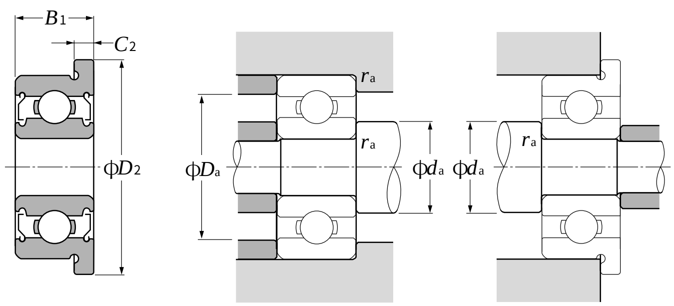
Open type | With single shield (Z) | With double shield (ZZ) | Open type with flange (FL) | With flanged outer ring and single shield (FL…Z)

II

d 1.5 ～ 5mm

Boundary dimensions									Basic load ratings				Factor	Limiting speeds	
									dynamic	static	dynamic	static			
mm									N		kgf			min^{-1}	
d	D	B	B_1	D_1	D_2	C_1	C_2	$r_{s\,min}$ 1)	C_r	C_{or}	C_r	C_{or}	f_o	grease	oil
1.5	4	1.2	2	5	5	0.4	0.6	0.15	102	29.0	10.0	3.00	13.6	88 000	100 000
	5	2	2.6	6.5	6.5	0.6	0.8	0.15	171	51.0	17.0	5.00	13.3	79 000	93 000
	6	2.5	3	7.5	7.5	0.6	0.8	0.15	274	86.0	28.0	9.00	12.3	71 000	84 000
2	4	1.2	2	—	—	—	—	0.05	104	37.0	11.0	4.00	14.8	83 000	98 000
	5	1.5	2.3	6.1	6.1	0.5	0.6	0.08	171	51.0	17.0	5.00	13.3	74 000	87 000
	5	2	2.5	—	—	—	—	0.1	171	51.0	17.0	5.00	13.3	74 000	87 000
	6	2.3	3	7.5	7.5	0.6	0.8	0.15	279	89.0	28.0	9.00	12.8	67 000	79 000
	6	2.5	—	7.2	—	0.6	—	0.15	279	89.0	28.0	9.00	12.8	67 000	79 000
	7	2.5	—	—	—	—	—	0.15	390	120	40.0	12.0	11.9	59 000	70 000
	7	2.8	3.5	8.5	8.5	0.7	0.9	0.15	380	125	39.0	13.0	12.4	62 000	73 000
2.5	5	1.5	2.3	—	—	—	—	0.08	153	59.0	16.0	6.00	15.0	70 000	82 000
	6	1.8	2.6	7.1	7.1	0.5	0.8	0.08	209	73.0	21.0	7.50	14.2	65 000	76 000
	7	3	3	—	8.2	—	0.6	0.15	284	96.0	29.0	10.0	13.8	59 000	70 000
	7	2.5	3.5	8.5	8.5	0.7	0.9	0.15	284	96.0	29.0	10.0	13.8	59 000	70 000
	8	2.5	2.8	9.2	—	0.6	—	0.15	430	152	44.0	16.0	13.2	56 000	66 000
	8	2.8	4	9.5	9.5	0.7	0.9	0.15	550	174	56.0	18.0	11.5	56 000	66 000
3	6	2	2.5	7.2	7.2	0.6	0.6	0.08	242	94.0	25.0	9.50	14.7	60 000	71 000
	7	2	3	8.1	8.1	0.5	0.8	0.1	390	130	40.0	13.0	13.0	58 000	68 000
	8	2.5	—	9.2	—	0.6	—	0.15	560	180	57.0	18.0	11.9	54 000	63 000
	8	3	4	9.5	9.5	0.7	0.9	0.15	560	180	57.0	18.0	11.9	54 000	63 000
	9	2.5	4	10.2	10.6	0.6	0.8	0.15	635	219	65.0	22.0	12.4	50 000	59 000
	9	3	5	10.5	10.5	0.7	1	0.15	635	219	65.0	22.0	12.4	50 000	59 000
	10	4	4	11.5	11.5	1	1	0.15	640	224	65.0	23.0	12.7	50 000	58 000
4	7	2	2.5	8.2	8.2	0.6	0.6	0.08	222	88.0	23.0	9.00	15.3	54 000	63 000
	8	2	3	9.2	9.2	0.6	0.6	0.08	395	140	40.0	14.0	13.9	52 000	61 000
	9	2.5	4	10.3	10.3	0.6	1	0.15	640	224	65.0	23.0	12.7	49 000	57 000
	10	3	4	11.2	11.6	0.6	0.8	0.15	650	235	66.0	24.0	13.3	46 000	55 000
	11	4	4	12.5	12.5	1	1	0.15	715	276	73.0	28.0	13.7	45 000	52 000
	12	4	4	13.5	13.5	1	1	0.2	970	360	99.0	36.0	12.8	43 000	51 000
	13	5	5	15	15	1	1	0.2	1 310	490	134	50.0	12.4	42 000	49 000
	16	5	5	—	—	—	—	0.3	1 760	680	179	69.0	12.4	37 000	44 000
5	8	2	2.5	9.2	9.2	0.6	0.6	0.08	217	91.0	22.0	9.50	15.8	49 000	57 000
	9	2.5	3	10.2	10.2	0.6	0.6	0.15	500	211	51.0	21.0	14.6	46 000	55 000
	10	3	4	11.2	11.6	0.6	0.8	0.15	715	276	73.0	28.0	13.7	45 000	52 000

1）Smallest allowable dimension for chamfer dimension r.



II

With flanged outer ring and double shield (FL···ZZ)

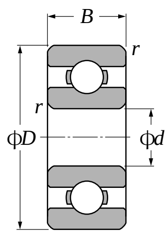
Dynamic equivalent radial load
$P_r = XF_r + YF_a$

$\frac{f_0 \cdot F_a}{C_{or}}$	e	$\frac{F_a}{F_r} \leqq e$		$\frac{F_a}{F_r} > e$	
		X	Y	X	Y
0.172	0.19	1	0	0.56	2.30
0.345	0.22				1.99
0.689	0.26				1.71
1.03	0.28				1.55
1.38	0.30				1.45
2.07	0.34				1.31
3.45	0.38				1.15
5.17	0.42				1.04
6.89	0.44				1.00

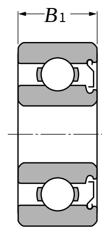
Static equivalent radial load
$P_{or} = 0.6F_r + 0.5F_a$
When $P_{or} < F_r$ use $P_{or} = F_r$

Bearing numbers						Abutment and fillet dimensions mm				Mass (approx.) g	
open	with single shield	with double shield	unsealed type with flange	with flanged OR and single shield	with flanged OR and double shield	d_a min	d_a max[2]	D_a max	r_{as} max	open	unsealed type with flange
68/1.5	**W68/1.5SA**	**SSA**	**FL68/1.5**	**FLW68/1.5SA**	**SSA**	2.3	2.4	3.2	0.05	0.07	0.09
69/1.5A	**W69/1.5ASA**	**SSA**	**FL69/1.5A**	**FLW69/1.5ASA**	**SSA**	2.7	2.9	3.8	0.15	0.18	0.24
60/1.5	**W60/1.5ZA**	**ZZA**	**FL60/1.5**	**FLW60/1.5ZA**	**ZZA**	2.7	3.0	4.8	0.15	0.35	0.42
672	—	—	—	—	—	2.5	2.6	3.5	0.05	0.06	—
682	**W682SA**	**SSA**	**FL682**	**FLW682SA**	**SSA**	2.8	2.9	4.2	0.08	0.13	0.17
BC2-5	**WBC2-5SA**	**SSA**	—	—	—	2.8	2.9	4.2	0.10	0.16	—
692	**W692SA**	**SSA**	**FL692**	**FLW692SA**	**SSA**	3.2	3.3	4.8	0.15	0.31	0.38
BC2-6	—	—	**FLBC2-6**	—	—	3.2	3.3	4.8	0.15	0.32	0.38
BC2-7A	—	—	—	—	—	3.2	3.6	5.8	0.15	0.44	—
602	**W602ZA**	**ZZA**	**FL602**	**FLW602ZA**	**ZZA**	3.2	3.7	5.8	0.15	0.54	0.64
67/2.5	**W67/2.5ZA**	**ZZA**	—	—	—	3.1	3.3	4.4	0.08	0.11	—
68/2.5	**W68/2.5ZA**	**ZZA**	**FL68/2.5**	**FLW68/2.5ZA**	**ZZA**	3.1	3.6	4.8	0.08	0.22	0.26
—	**WBC2.5-7ZA**	**ZZA**	—	**FLWBC2.5-7ZA**	**ZZA**	3.7	4.0	5.8	0.15	0.6[3]	0.67[3]
69/2.5	**W69/2.5SA**	**SSA**	**FL69/2.5**	**FLW69/2.5SA**	**SSA**	3.7	4.0	5.8	0.15	0.43	0.53
BC2.5-8	**WBC2.5-8ZA**	**ZZA**	**FLBC2.5-8**	—	—	3.7	4.3	6.8	0.15	0.57	0.65
60/2.5	**W60/2.5ZA**	**ZZA**	**FL60/2.5**	**FLW60/2.5ZA**	**ZZA**	3.7	4.1	6.8	0.15	0.72	0.83
673	**WA673SA**	**SSA**	**FL673**	**FLWA673SA**	**SSA**	3.6	4.1	5.4	0.08	0.2	0.26
683	**W683ZA**	**ZZA**	**FL683**	**FLW683ZA**	**ZZA**	3.9	4.1	5.8	0.1	0.33	0.38
BC3-8	—	—	**FLBC3-8**	—	—	4.2	4.4	6.8	0.15	0.52	0.6
693	**W693Z**	**ZZ**	**FL693**	**FLW693Z**	**ZZ**	4.2	4.4	6.8	0.15	0.61	0.72
BC3-9	**WBC3-9ZA**	**ZZA**	**FLBC3-9**	**FLAWBC3-9ZA**	**ZZA**	4.2	5.0	7.8	0.15	0.71	0.79
603	**W603Z**	**ZZ**	**FL603**	**FLW603Z**	**ZZ**	4.2	5.0	7.8	0.15	0.92	1
623	**623Z**	**ZZ**	**FL623**	**FL623Z**	**ZZ**	4.2	5.2	8.8	0.15	1.6	1.8
674A	**WA674ASA**	**SSA**	**FL674A**	**FLWA674ASA**	**SSA**	4.6	5.0	6.4	0.08	0.28	0.35
BC4-8	**WBC4-8Z**	**ZZ**	**FLBC4-8**	**FLWBC4-8Z**	**ZZ**	4.8	5.0	6.8	0.08	0.38	0.46
684AX50	**W684AX50Z**	**ZZ**	**FL684AX50**	**FLW684AX50Z**	**ZZ**	5.0	5.2	7.8	0.1	0.67	0.76
BC4-10	**WBC4-10Z**	**ZZ**	**FLBC4-10**	**FLAWBC4-10Z**	**ZZ**	5.2	6.0	8.8	0.15	1	1.1
694	**694Z**	**ZZ**	**FL694**	**FL694Z**	**ZZ**	5.2	6.4	9.8	0.15	1.8	2
604	**604Z**	**ZZ**	**FL604**	**FL604Z**	**ZZ**	5.6	6.6	10.4	0.2	2.1	2.3
624	**624Z**	**ZZ**	**FL624**	**FL624Z**	**ZZ**	5.6	6.2	11.4	0.2	3.2	3.5
634	**634Z**	**ZZ**	—	—	—	6	7.6	14	0.3	5.1	—
675	**WA675Z**	**ZZ**	**FL675**	**FLWA675Z**	**ZZ**	5.6	6.0	7.4	0.08	0.32	0.4
BC5-9	**WBC5-9Z**	**ZZ**	**FLBC5-9**	**FLWBC5-9Z**	**ZZ**	5.2	6.1	7.8	0.15	0.55	0.63
BC5-10	**WBC5-10Z**	**ZZ**	**FLBC5-10**	**FLAWBC5-10Z**	**ZZ**	6.2	6.4	8.8	0.15	0.88	0.97

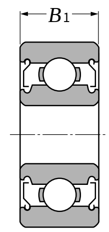
2）This dimension applies to sealed and shielded bearings. 3）Values for double shielded bearings shown.



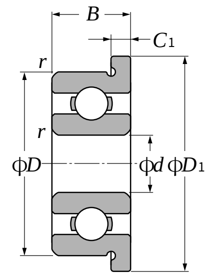
Open type



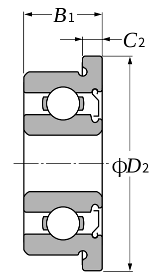
With single shield
(Z)



With double shield
(ZZ)



Open type
with flange
(FL)

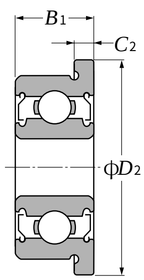


With flanged outer ring
and single shield
(FL···Z)

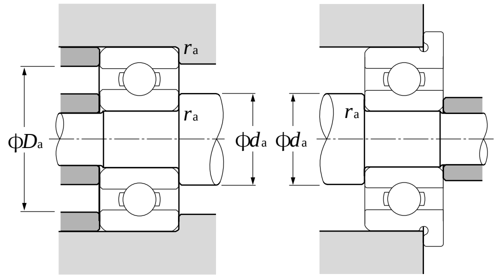
d 5 ～ 9mm

Boundary dimensions (mm)									Basic load ratings dynamic (N)	static (N)	dynamic (kgf)	static (kgf)	Factor	Limiting speeds (min⁻¹)	
d	D	B	B_1	D_1	D_2	C_1	C_2	$r_{s\,min}$ 1)	C_r	C_{or}	C_r	C_{or}	f_o	grease	oil
5	11	4	4	—	12.6	—	0.8	0.15	715	282	73.0	29.0	14.0	43 000	51 000
	11	3	5	12.5	12.5	0.8	1	0.15	715	282	73.0	29.0	14.0	43 000	51 000
	13	4	4	15	15.2	1	1	0.2	1 080	430	110	44.0	13.4	40 000	47 000
	13	5	5	—	15	—	1	0.2	1 080	430	110	44.0	13.4	40 000	47 000
	14	5	5	16	16	1	1	0.2	1 330	505	135	52.0	12.8	39 000	46 000
	16	5	5	18	18	1	1	0.3	1 760	680	179	69.0	12.4	37 000	44 000
	19	6	6	—	—	—	—	0.3	2 340	885	238	90.0	12.1	34 000	40 000
6	10	2.5	3	11.2	11.2	0.6	0.6	0.1	465	196	47.0	20.0	15.2	43 000	51 000
	12	3	4	13.2	13.6	0.6	0.8	0.15	830	365	85.0	37.0	14.5	40 000	47 000
	13	3.5	5	15	15	1.0	1.1	0.15	1 080	440	110	45.0	13.7	39 000	46 000
	15	5	5	17	17	1.2	1.2	0.2	1 350	530	137	54.0	13.3	37 000	44 000
	16	6	6	—	—	—	—	0.2	1 770	695	181	71.0	12.7	36 000	42 000
	17	6	6	19	19	1.2	1.2	0.3	2 190	865	224	88.0	12.3	35 000	42 000
	19	6	6	22	22	1.5	1.5	0.3	2 340	885	238	90.0	12.1	34 000	40 000
7	11	2.5	3	12.2	12.2	0.6	0.6	0.1	555	269	56.0	27.0	15.6	40 000	47 000
	13	3	4	14.2	14.6	0.6	0.8	0.15	825	375	84.0	38.0	14.9	38 000	45 000
	14	3.5	5	16	16	1	1.1	0.15	1 170	505	120	51.0	14.0	37 000	44 000
	17	5	5	19	19	1.2	1.2	0.3	1 610	715	164	73.0	14.0	35 000	41 000
	19	6	6	—	—	—	—	0.3	2 240	910	228	93.0	12.9	34 000	40 000
	22	7	7	—	—	—	—	0.3	3 350	1 400	340	142	12.5	32 000	37 000
8	12	2.5	3.5	13.2	13.6	0.6	0.8	0.1	515	252	52.0	26.0	15.9	38 000	45 000
	14	3.5	4	15.6	15.6	0.8	0.8	0.15	820	385	84.0	39.0	15.2	36 000	43 000
	16	4	5	18	18	1	1.1	0.2	1 610	715	164	73.0	14.0	35 000	41 000
	19	6	6	22	22	1.5	1.5	0.3	1 990	865	202	88.0	13.8	33 000	39 000
	22	7	7	25	25	1.5	1.5	0.3	3 350	1 400	340	142	12.5	32 000	37 000
	24	8	8	—	—	—	—	0.3	4 000	1 590	410	162	11.7	31 000	36 000
9	14	3	4.5	—	—	—	—	0.1	920	465	94.0	48.0	15.5	36 000	42 000
	17	4	5	19	19	1	1.1	0.2	1 720	820	176	83.0	14.4	33 000	39 000
	20	6	6	—	—	—	—	0.3	2 480	1 090	253	111	13.5	32 000	38 000
	24	7	7	—	—	—	—	0.3	3 400	1 450	345	148	12.9	31 000	36 000
	26	8	8	—	—	—	—	0.6	4 550	1 960	465	200	12.4	30 000	35 000

1）Smallest allowable dimension for chamfer dimension r.



With flanged outer ring and double shield (FL···ZZ)



II

Dynamic equivalent radial load
$P_r = XF_r + YF_a$

$\frac{f_0 \cdot F_a}{C_{or}}$	e	$\frac{F_a}{F_r} \leqq e$ X	Y	$\frac{F_a}{F_r} > e$ X	Y
0.172	0.19	1	0	0.56	2.30
0.345	0.22				1.99
0.689	0.26				1.71
1.03	0.28				1.55
1.38	0.30				1.45
2.07	0.34				1.31
3.45	0.38				1.15
5.17	0.42				1.04
6.89	0.44				1.00

Static equivalent radial load
$P_{or} = 0.6F_r + 0.5F_a$
When $P_{or} < F_r$ use $P_{or} = F_r$

Bearing numbers						Abutment and fillet dimensions mm				Mass (approx.) g	
open	with single shield	with double shield	unsealed type with flange	with flanged OR and single shield	with flanged OR and double shield	d_a min	d_a max[2)]	D_a max	r_{as} max	open	unsealed type with flange
—	**WBC5-11Z**	**ZZ**	—	**FLWBC5-11Z**	**ZZ**	6.2	6.8	9.8	0.2	1.8	2
685	**W685Z**	**ZZ**	**FL685**	**FLW685Z**	**ZZ**	6.2	6.8	9.8	0.15	1.1	1.3
695	**695Z**	**ZZ**	**FL695**	**FL695Z**	**ZZ**	6.6	6.9	11.4	0.2	2.4	2.7
—	**WBC5-13Z**	**ZZ**	—	**FLWBC5-13Z**	**ZZ**	6.6	6.9	11.4	0.2	3.4[3)]	3.7[3)]
605	**605Z**	**ZZ**	**FL605**	**FL605Z**	**ZZ**	6.6	7.4	12.4	0.2	3.5	3.9
625	**625Z**	**ZZ**	**FL625**	**FL625Z**	**ZZ**	7	7.6	14	0.3	4.8	5.2
635	**635Z**	**ZZ**	—	—	—	7	9.5	17	0.3	8	—
676A	**WA676AZ**	**ZZ**	**FL676A**	**FLWA676AZ**	**ZZ**	6.6	6.7	9.2	0.1	0.65	0.74
BC6-12	**WBC6-12Z**	**ZZ**	**FLBC6-12**	**FLAWBC6-12Z**	**ZZ**	7.2	7.9	10.8	0.15	1.3	1.4
686	**W686Z**	**ZZ**	**FL686**	**FLW686Z**	**ZZ**	7.0	7.2	11.8	0.15	1.9	2.2
696	**696Z**	**ZZ**	**FL696**	**FL696Z**	**ZZ**	7.6	7.8	13.4	0.2	3.8	4.3
BC6-16A	**BC6-16AZ**	**ZZ**	—	—	—	7.6	8.0	14.4	0.2	5.2	—
606	**606Z**	**ZZ**	**FL606**	**FL606Z**	**ZZ**	8	8.6	15	0.3	6	6.5
626	**626Z**	**ZZ**	**FL626**	**FL626Z**	**ZZ**	8	9.5	17	0.3	8.1	9.2
677	**WA677Z**	**ZZ**	**FL677**	**FLWA677Z**	**ZZ**	7.8	8.1	10.2	0.1	0.67	0.77
BC7-13	**WBC7-13Z**	**ZZ**	**FLBC7-13**	**FLAWBC7-13Z**	**ZZ**	8.2	8.9	11.8	0.15	1.4	1.5
687A	**W687AZ**	**ZZ**	**FL687A**	**FLW687AZ**	**ZZ**	8.2	8.7	12.8	0.15	2.1	2.4
697	**697Z**	**ZZ**	**FL697**	**FL697Z**	**ZZ**	9	10.0	15	0.3	5.2	5.7
607	**607Z**	**ZZ**	—	—	—	9	10.4	17	0.3	8	—
627	**627Z**	**ZZ**	—	—	—	9	12.2	20	0.3	13	—
678A	**W678AZ**	**ZZ**	**FL678A**	**FLAW678AZ**	**ZZ**	8.8	9.1	11.2	0.1	0.75	0.86
BC8-14	**WBC8-14Z**	**ZZ**	**FLBC8-14**	**FLWBC8-14Z**	**ZZ**	9.2	9.5	12.8	0.15	1.8	1.9
688A	**W688AZ**	**ZZ**	**FL688A**	**FLW688AZ**	**ZZ**	9.6	10.0	14.4	0.2	3.1	3.5
698	**698Z**	**ZZ**	**FL698**	**FL698Z**	**ZZ**	10	10.6	17	0.3	7.3	8.4
608	**608Z**	**ZZ**	**FL608**	**FL608Z**	**ZZ**	10	12.2	20	0.3	12	13
628	**628Z**	**ZZ**	—	—	—	10	12.1	22	0.3	17	—
679	**W679Z**	**ZZ**	—	—	—	9.8	10.4	13.2	0.1	1.4	—
689	**W689Z**	**ZZ**	**FL689**	**FLW689Z**	**ZZ**	10.6	10.7	15.4	0.2	3.2	3.6
699	**699Z**	**ZZ**	**—**	**—**	**—**	11	11.6	18	0.3	8.2	—
609	**609Z**	**ZZ**	**—**	**—**	**—**	11	13.1	22	0.3	14	—
629X50	**629X50Z**	**ZZ**	**—**	**—**	**—**	13	13.9	22	0.3	20	—

2) This dimension applies to sealed and shielded bearings. 3) Values for double shielded bearings shown.

1. Classification and Characteristics of Rolling Bearings

1.1 Rolling bearing construction

Most rolling bearings consist of rings with raceway (inner ring and outer ring), rolling elements (either balls or rollers) and cage. The cage separates the rolling elements at regular intervals, holds them in place within the inner and outer raceways, and allows them to rotate freely.

Raceway (inner ring and outer ring) or raceway disc [1)]

The surface on which rolling elements roll is called the "raceway surface". The load placed on the bearing is supported by this contact surface.

Generally the inner ring fits on the axle or shaft and the outer ring on the housing.

Note 1: The raceway of thrust bearing is called "raceway disc," the inner ring is called the "shaft raceway disc" and the outer ring is called the "housing raceway disc."

Rolling elements

Rolling elements classify in two types: balls and rollers. Rollers come in four types: cylindrical, needle, tapered, and spherical.

Balls geometrically contact with the raceway surfaces of the inner and outer rings at "points", while the contact surface of rollers is a "line" contact.

Theoretically, rolling bearings are so constructed as to allow the rolling elements to rotate orbitally while also rotating on their own axes at the same time.

Cages

Cages function to maintain rolling elements at a uniform pitch so load is never applied directly to the cage and to prevent the rolling elements from falling out when handling the bearing. Types of cages differ according to way they are manufactured, and include pressed, machined and formed cages.

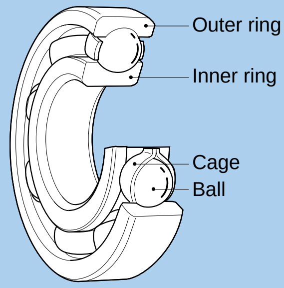
1.2 Classification of rolling bearings

Rolling bearings divide into two main classifications: ball bearings and roller bearings. Ball bearings are classified according to their bearing ring configurations: deep groove type and angular contact type. Roller bearings on the other hand are classified according to the shape of the rollers: cylindrical, needle, tapered and spherical.

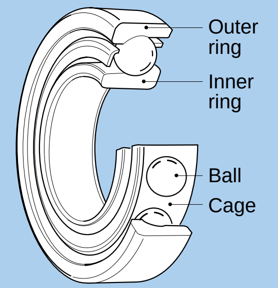
Rolling bearings can be further classified according to the direction in which the load is applied; radial bearings carry radial loads and thrust bearings carry axial loads.

Other classification methods include: 1) number of rolling rows (single, double, or 4-row), 2) separable and non-separable, in which either the inner ring or the outer ring can be detached.

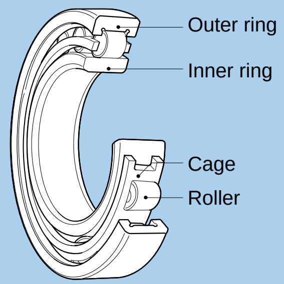
There are also bearings designed for special applications, such as: railway car journal roller bearings, ball screw support bearings, turntable bearings, as well as linear motion bearings (linear ball bearings, linear roller bearings and linear flat roller bearings).Types of rolling bearings are given in **Fig. 1.2**.



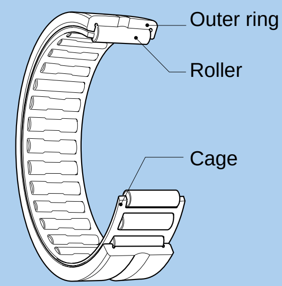
Deep groove ball bearing
Fig A



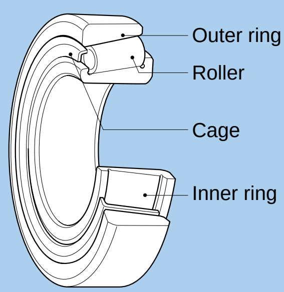
Angular contact ball bearing
Fig.B



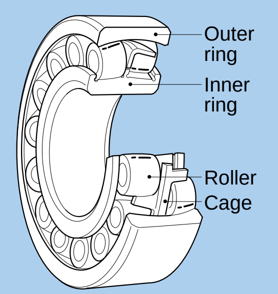
Cylindrical roller bearing
Fig. C



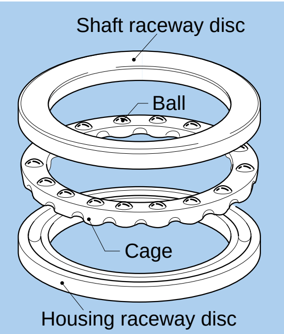
Needle roller bearing
Fig. D



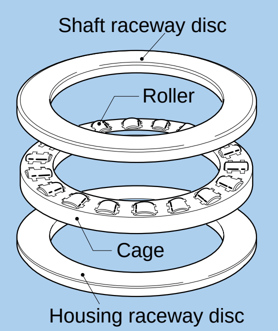
Tapered roller bearing
Fig. E



Spherical roller bearing
Fig. F



Thrust ball bearing
Fig. G



Thrust roller bearing
Fig. H

Fig. 1.1 Rolling bearing

II

II

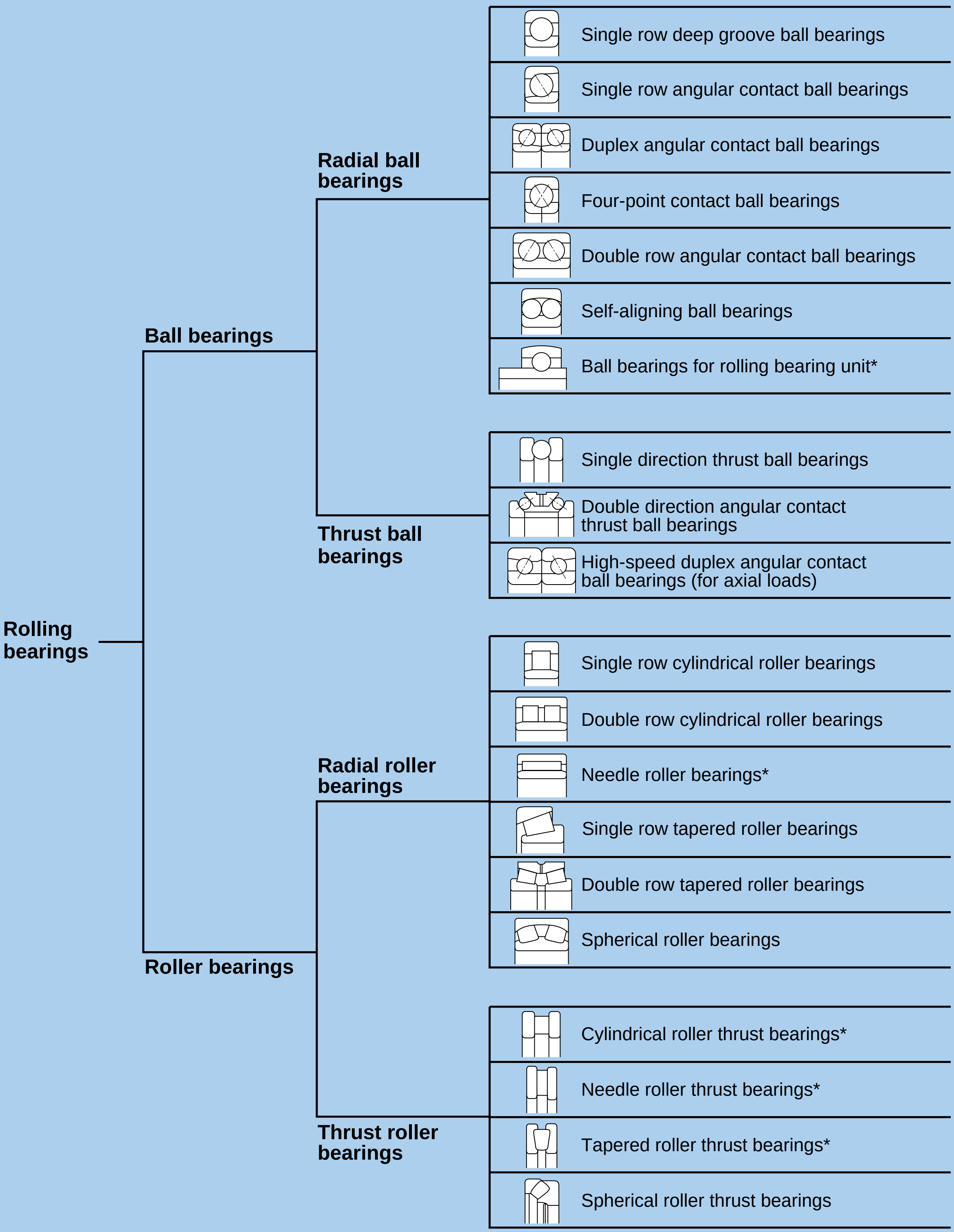
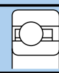
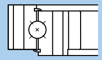
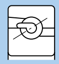
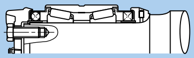
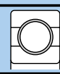
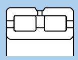
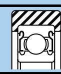
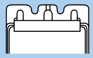
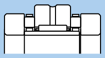
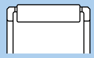
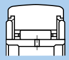
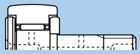


Fig. 1.2 Classification of rolling bearings

II

Special application bearings

-  Ultra thin wall type ball bearings*
-  Turntable bearings*
-  Ball screw support bearings*
-  Railway car journal roller bearings*
-  Ultra-clean vacuum bearings*
-  SL-type cylindrical roller bearings*
-  Rubber molded bearings*
-  Clearance adjusting needle roller bearings*
-  Complex bearings*
-  Connecting rod needle roller bearings with cage*
-  Roller followers*
-  Cam followers*

Linear motion bearings

- Linear ball bearings*
- Linear roller bearings*
- Linear flat roller bearings*

Note: Bearings marked with an asterisk are not contained in this catalog.
For details, see the catalog devoted to the concerned type of bearing.

II

1.3 Characteristics of rolling bearings

1.3.1 Characteristics of rolling bearings

Rolling bearings come in many shapes and varieties, each with its own distinctive features.
However, when compared with sliding bearings, rolling bearings all have the following advantages:

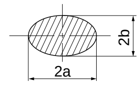
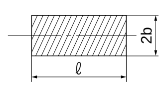
(1) The starting friction coefficient is lower and there is little difference between this and the dynamic friction coefficient.
(2) They are internationally standardized, interchangeable and readily obtainable.
(3) They are easy to lubricate and consume less lubricant.
(4) As a general rule, one bearing can carry both radial and axial loads at the same time.
(5) May be used in either high or low temperature applications.
(6) Bearing rigidity can be improved by preloading.

Construction, classes, and special features of rolling bearings are fully described in the boundary dimensions and bearing numbering system section.

1.3.2 Ball bearings and roller bearings

Table 1.1 gives a comparison of ball bearings and roller bearings.

Table 1.1 Comparison of ball bearings and roller bearings

	Ball bearings	Roller bearings
Contact with raceway	 **Point contact** Contact surface is oval when load is applied.	 **Linear contact** Contact surface is generally rectangular when load is applied.
Characteristics	Because of point contact there is little rolling resistance, ball bearings are suitable for low torque and high-speed applications. They also have superior acoustic characteristics.	Because of linear contact, rotational torque is higher for roller bearings than for ball bearings, but rigidity is also higher.
Load capacity	Load capacity is lower for ball bearings, but radial bearings are capable of bearing loads in both the radial and axial direction.	Load capacity is higher for rolling bearings. Cylindrical roller bearings equipped with a lip can bear slight radial loads. Combining tapered roller bearings in pairs enables the bearings to bear an axial load in both directions.

1.3.3 Radial and thrust bearings

Almost all types of rolling bearings can carry both radial and axial loads at the same time.
Generally, bearings with a contact angle of less than 45° have a much greater radial load capacity and are classed as radial bearings; whereas bearings which have a contact angle over 45° have a greater axial load capacity and are classed as thrust bearings. There are also bearings classed as complex bearings which combine the loading characteristics of both radial and thrust bearings.

1.3.4 Standard bearings and special bearings

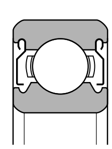
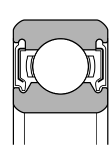
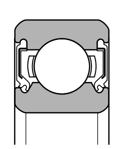
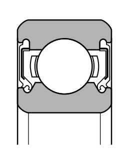
The boundary dimensions and shapes of bearings conforming to international standards are interchangeable and can be obtained easily and economically over the world over. It is therefore better to design mechanical equipment to use standard bearings.

However, depending on the type of machine they are to be used in, and the expected application and function, a non-standard or specially designed bearing may be best to use. Bearings that are adapted to specific applications, and "unit bearings" which are integrated (built-in) into a machine's components, and other specially designed bearings are also available.
The feature of typical standard bearings are as follows:

Deep groove ball bearings

The most common type of bearing, deep groove ball bearings are widely used in a variety of fields. Deep groove ball bearings include shield bearings and sealed bearings with grease make them easier to use.
Deep groove ball bearings also include bearings with a locating snap-ring to facilitate positioning when mounting the outer ring, expansion compensating bearings which absorb dimension variation of the bearing fitting surface due to housing temperature, and TAB bearings that are able to withstand contamination in the lubricating oil.

Table 1.2 Configuration of sealed ball bearings

Type and symbol	Shield	Sealed		
	Non-contact ZZ	Non-contact LLB	Contact LLU	Low torque LLH
Configuration				

Angular contact ball bearings

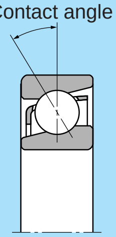
The line that unites point of contact of the inner ring, ball and outer ring runs at a certain angle (contact angle) in the radial direction. Bearings are generally designed with three contact angles.

Angular contact ball bearings can support an axial load, but cannot be used by single bearing because of the contact angle. They must instead be used in pairs or in combinations.

Angular contact ball bearings include double row angular contact ball bearings for which the inner and outer rings are combined as a single unit. The contact angle of double row angular contact ball bearings is 25°.

There are also four-point contact bearings that can support an axial load in both directions by themselves. These bearings however require caution because problems such as excessive temperature rise and wearing could occur depending on the load conditions.

Table 1.3 Contact angle and symbol



Contact angle and contact angle symbol

Contact angle	15 °	30 °	40 °
Contact angle symbol	C	A [1]	B

Note 1: Contact angle symbol has been abbreviated as "A".

Table 1.4 Configuration of double row angular contact ball bearings

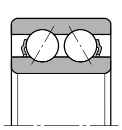
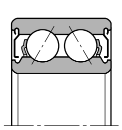
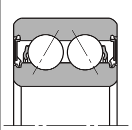
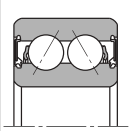
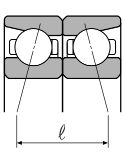
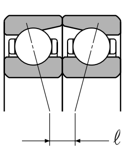
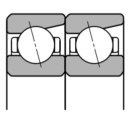
Type and symbol	Open	Shield ZZ	Non-contact sealed LLM	Contact sealed LLD
Configuration				

Table 1.5 Combinations of duplex angular contact ball bearings

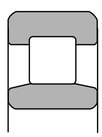
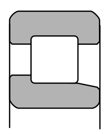
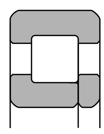
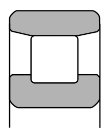
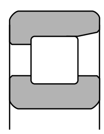
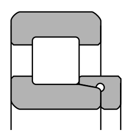
Type and symbol	Back-to-back duplex DB	Face-to-face duplex DF	Tandem duplex DT
Configuration			

Cylindrical roller bearings

Uses rollers for rolling elements, and therefore has a high load capacity. The rollers are guided by the ribs of the inner or outer ring. The inner and outer rings can be separated to facilitate assembly, and both can be fit with shaft or housing tightly. If there is no ribs, either the inner or the outer ring can move freely in the axial direction. Cylindrical roller bearings are therefore ideal to be used as so-called "free side bearings" that absorb shaft expansion. In the case where there is a ribs, the bearing can bear a slight axial load between the end of the rollers and the ribs. Cylindrical roller bearings include the HT type which modifies the shape of roller end face and ribs for increasing axial road capacity. And the E type with a special internal design for enhancing radial load capacity. The E type is standardized for small-diameter sizes. **Table 1.6** shows the basic configuration for cylindrical roller bearings.

In addition to these, there are cylindrical roller bearings with multiple rows of rollers and the SL type of full complement roller bearing without cage.

Table 1.6 Types of cylindrical roller bearings

Type and Symbol	NU type N type	NJ type NF type	NUP type NH type (NJ+HJ)
Drawings	 NU type	 NJ type	 NUP type
	 N type	 NF type	 NH type

II

II

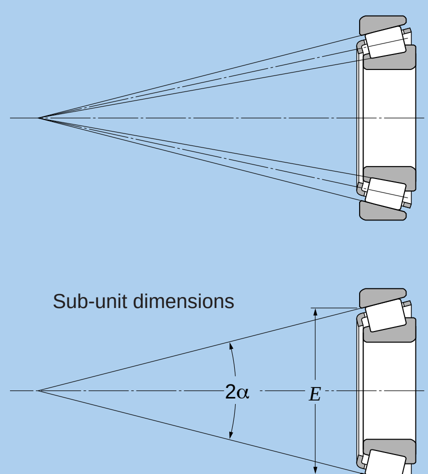
Tapered roller bearings

Tapered roller bearings are designed so the inner/outer ring raceway and apex of the tapered rollers intersect at one point on the bearing centerline. By receiving combined load from inner and outer ring, the rollers are pushed against the inner ring rib and roll guided with rib.

Induced force is produced in the axial direction when a radial load is applied, so must be handled by using a pair of bearings. The inner ring with rollers and outer ring come apart, thus facilitating mounting with clearance or preload. Assembled clearance is however hard to manage and requires special attention. Tapered roller bearings are capable of supporting large loads in both the axial and radial directions.

NTN bearings with 4T-, ET-, T- and U attached to the name conform to ISO and JIS standards for sub-unit dimensions (nominal contact angle, nominal small end diameter of outer ring) and are internationally interchangeable.

NTN also has a line of case hardened steel bearings designed for longer life (ETA-, ET-, etc.). NTN tapered roller bearings also include bearings with two and four rows of tapered rollers for extra-heavy loads.



E : nominal small end diameter of outer ring
α : Nominal contact angle

Fig. 1.3 Tapered roller bearings

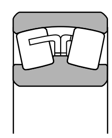
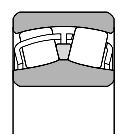
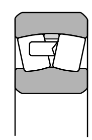
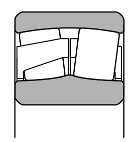
Spherical roller bearings

Equipped with an outer ring with a spherical raceway surface and an inner ring which holds two rows of barrel-shaped rolling elements, NTN spherical roller bearings are able to adjust center alignment to handle inclination of the axle or shaft.

There are variety of bearing types that differ according to internal design.

Spherical roller bearings include as type equipped with an inner ring with a tapered bore. The bearing can easily be mounted on a shaft by means of an adapter or withdrawal sleeve. The bearing is capable of supporting heavy loads, and is therefore often used in industrial machinery. When heavy axial load is applied to the bearing, the load on rollers of another row is disappeared, and can cause problems. Attention must therefore be paid to operating conditions.

Table 1.7 Types of spherical roller bearings

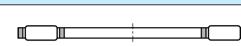
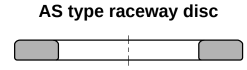
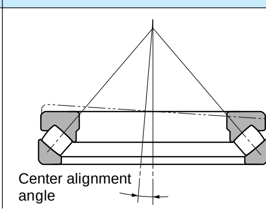
Type	Standard (B type)	C type	213 type	E type
Configuration				

Thrust bearings

There are many types of thrust bearings that differ according to shape of rolling element and application. Allowable rotational speed is generally low and special attention must be paid to lubrication.

In addition to the ones given below, there are various types of thrust bearings for special applications. For details, see the catalog devoted to the concerned type of bearing.

Table 1.8 Types of thrust bearings

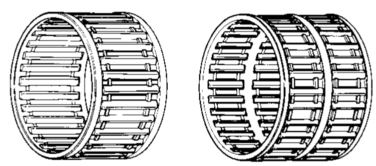
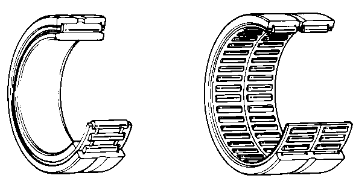
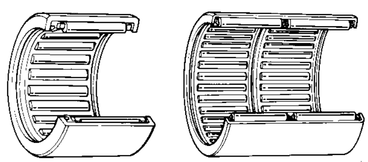
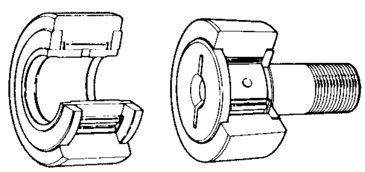
Type	Single direction thrust ball bearing	Needle roller thrust bearing
Configuration		 AXK type  AS type raceway disc  GS/WS type raceway disc
	Cylindrical roller thrust bearing	Spherical roller thrust bearing
		

Needle roller bearings

Needle roller bearings use needle rollers as rolling elements. The needle rollers are a maximum of 5 mm in diameter and are 3 to 10 times as long as they are in diameter. Because the bearings use needle rollers as rolling elements, the cross-section is thin, but they have a high load capacity for their size. Because of the large number of rolling elements, the bearings have high rigidity and are ideally suited to wobbling or pivoting motion.

There is a profusion of types of needle roller bearings, and just a few of the most representative types are covered here. For details, see the catalog devoted to the concerned type of bearing.

Table 1.9 Main types of needle roller bearings

Type	Needle roller bearing with cage
Configuration	
	Solid type needle roller bearings
	
	Shell type needle roller bearings
	
	Roller follower Cam follower
	

Bearing unit

A unit comprised of a ball bearing inserted into various types of housings. The housing can be bolted onto machinery and the inner ring can be easily mounted on the shaft with a set screw.

This means the bearing unit can support rotating equipment without special design to allow for mounting. A variety of standardized housing shapes is available, including pillow and flange types. The outer diameter of the bearing is spherical just like the inner diameter of the housing, so it capable of aligning itself on the shaft.

For lubrication, grease is sealed inside the bearing, and particle generation is prevented by a double seal. For details, see the catalog devoted to the concerned type of bearing.

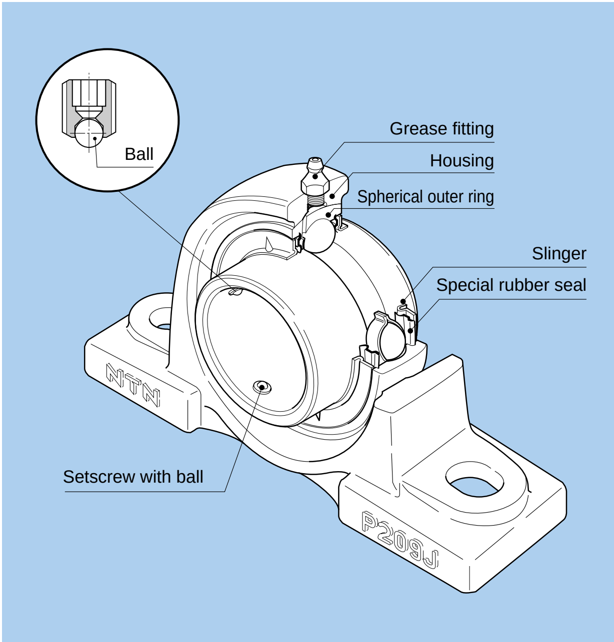


Fig. 1.4 Oil-lubricated bearing unit

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2. Bearing Selection

Rolling element bearings are available in a variety of types, configurations, and sizes. When selecting the correct bearing for your application, it is important to consider several factors, and analyse in various means.

A comparison of the performance characteristics for each bearing type is shown in **Table 2.1**. As a general guideline, the basic procedure for selecting the most appropriate bearing is shown in the following flow chart.

2.1 Bearing selection flow chart

Procedure	Confirm operating conditions and operating environment →	Select bearing type and configuration →	Select bearing dimensions →	Select bearing tolerances →
Confirmation items	●Function and construction of components to house bearings ●Bearing mounting location ●Bearing load (direction and magnitude) ●Rotational speed ●Vibration and shock load ●Bearing temperature (Ambient temperature / temperature rise) ●Operating environment (potential for corrosion, degree of contamination, extent of lubrication)	●Dimensional limitations (refer to page insert ···A-30) ●Bearing load (magnitude, direction, vibration; presence of shock load) (refer to page insert ···A-21) ●Rotational speed (refer to page insert ···A-70) ●Bearing tolerances (refer to page insert ···A-35) ●Rigidity (refer to page insert ···A-67) ●Allowable misalignment of inner/outer rings (refer to page insert ···A-87) ●Friction torque (refer to page insert ···A-71) ●Bearing arrangement (fixed side, floating side) (refer to page insert ···A-15) ●Installation and disassembly requirements (refer to page insert ···A-88) ●Bearing availability and cost	●Design life of components to house bearings (refer to page insert ···A-19) ●Dynamic/static equivalent load conditions (refer to page insert ···A-25) ●Safety factor (refer to page insert ···A-19) ●Allowable speed (refer to page insert ···A-70) ●Allowable axial load (refer to page insert ···A-19, 25) ●Allowable space (refer to page insert ···A-30)	●Shaft runout tolerances (refer to page insert ···A-35) ●Rotational speed (refer to page insert ···A-70) ●Torque fluctuation

Selection of bearing type and configuration

(1) Dimensional limitations

The allowable space for bearings is generally limited. In most cases, shaft diameter (or the bearing bore diameter) has been determined according to the machine's other design specifications. Therefore, bearing's type and dimensions are determined according to bearing bore diameters. For this reason all dimension tables are organized according to standard bore diameters. There is a wide range of standardized bearing types and dimensions: the right one for a particular application can usually be found in these tables.

(2) Bearing load

The characteristics, magnitude, and direction of loads acting upon a bearing are extremely variable. In general, the basic load ratings shown in bearing dimension tables indicate their load capacity. However, in determining the appropriate bearing type, consideration must also be given to whether the acting load is a radial load only or combined radial and axial load, etc. When ball and roller bearings within the same dimension series are considered, the roller bearings have a larger load capacity and are also capable of withstanding greater vibration and shock loads.

(3) Rotational speed

The allowable speed of a bearing will differ depending upon bearing type, size, tolerances, cage type, load, lubricating conditions, and cooling conditions.

The allowable speeds listed in the bearing tables for grease and oil lubrication are for normal tolerance NTN bearings. In general, deep groove ball bearings, angular contact ball bearings, and cylindrical roller bearings are most suitable for high speed applications.

(4) Bearing tolerances

The dimensional accuracy and operating tolerances of bearings are regulated by ISO and JIS standards. For equipment requiring high tolerance shaft runout or high speed operation, bearings with Class 5 tolerance or higher are recommended. Deep groove ball bearings, angular contact ball bearings, and cylindrical roller bearings are recommended for high rotational tolerances.

(5) Rigidity

Elastic deformation occurs along the contact surfaces of a bearing's rolling elements and raceway surfaces under loading. With certain types of equipment it is necessary to reduce this deformation as much as

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Select bearing's internal clearance	Select cage type and material	Select lubricant, lubrication method, sealing method	Select any special bearing specifications	Confirm handling procedures
●Material and shape of shaft and housing (refer to page insert ···A-85) ●Fit (refer to page insert ···A-49) ●Temperature differential between inner/outer rings (refer to page insert ···A-59) ●Allowable misalignment of inner/outer rings (refer to page insert ···A-87) ●Load (magnitude, nature) (refer to page insert ···A-21) ●Amount of preload (refer to page insert ···A-66) ●Rotational speed (refer to page insert ···A-70)	●Rotational speed (refer to page insert ···A-70) ●Noise level ●Vibration and shock load ●Momentary load ●Lubrication type and method (refer to page insert ···A-72)	●Operating temperature (refer to page insert ···A-72) ●Rotational speed (refer to page insert ···A-70) ●Lubrication type and method (refer to page insert ···A-72) ●Sealing method (refer to page insert ···A-80) ●Maintenance and inspection (refer to page insert ···A-94)	●Operating environment (high/low temperature, vacuum, pharmaceutical, etc.) ●Requirement for high reliability	●Installation-related dimensions (refer to page insert ···A-86) ●Installation and disassembly procedures (refer to page insert ···A-88)

possible. Roller bearings exhibit less elastic deformation than ball bearings. Furthermore, in some cases, bearings are given a load in advance (preloaded) to increase their rigidity. This procedure is commonly applied to deep groove ball bearings, angular contact ball bearings, and tapered roller bearings.

(6) Misalignment of inner and outer rings

Shaft flexure, variations in shaft or housing accuracy, and fitting errors. result in a certain degree of misalignment between the bearing's inner and outer rings. In cases where the degree of misalignment is relatively large, self-aligning ball bearings, spherical roller bearings, or bearing units with self-aligning properties are the most appropriate choices. (Refer to **Fig. 2.1**)

(7) Noise and torque levels

Rolling bearings are manufactured and processed according to high precision standards, and therefore generally produce only slight amounts of noise and torque. For applications requiring particularly low-noise or low-torque operation, deep groove ball bearings and cylindrical roller bearings are most appropriate.

(8) Installation and disassembly

Some applications require frequent disassembly and reassembly to enable periodic inspections and repairs. For such applications, bearings with separable inner/outer rings, such as cylindrical roller bearings, needle roller bearings, and tapered roller bearings are most appropriate. Incorporation of adapter sleeves simplifies the installation and disassembly of self-aligning ball bearings and spherical roller bearings with tapered bores.

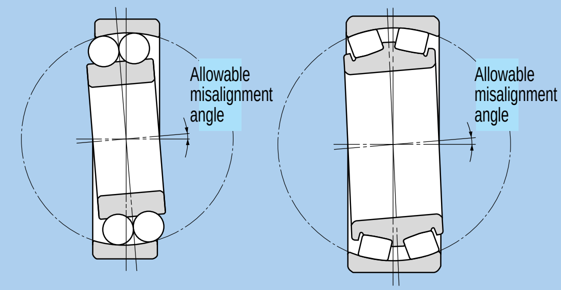

Fig. 2.1

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2.2 Type and characteristics

Table 2.1 shows types and characteristics of rolling bearings.

Table 2.1 Type of rolling bearings and performance comparison

Characteristics \ Bearing types	Deep groove ball bearings	Angular contact ball bearings	Double row angular contact ball bearings	Duplex angular contact ball bearings	Self-aligning ball bearings	Cylindrical roller bearings	Single-flange cylindrical roller bearings	Double-flange cylindrical roller bearings	Double row cylindrical roller bearings	Needle roller bearings
Load Carrying Capacity (Radial load / Axial load)										
High speed❶	☆☆☆☆	☆☆☆☆	☆☆	☆☆☆	☆☆	☆☆☆☆	☆☆☆	☆☆☆	☆☆☆	☆☆☆
High rotating accuracy❶	☆☆☆	☆☆☆	☆☆	☆☆☆		☆☆☆	☆☆	☆	☆☆☆	
Low noise/vibration❶	☆☆☆☆	☆☆☆		☆		☆	☆	☆	☆	☆
Low friction torque❶	☆☆☆☆	☆☆☆		☆☆	☆	☆				
High rigidity❶			☆☆	☆☆		☆☆	☆☆	☆☆	☆☆☆	☆☆
Vibration/shock resistance❶			☆		★	☆☆	☆☆	☆☆	☆☆	☆☆
Allowable misalignment for inner/outer rings❶	☆				☆☆☆	☆				
Stationary in axial direction❷	◎	○	◎	◎ For DB and DF arrangement	◎		○	◎		
Moveable in axial direction❸	○		○	○ For DB arrangement	○	◎			◎	◎
Separable inner/outer rings❹						○	○	○	○	○
Inner ring tapered bore❺					○	○			○	
Remarks		For duplex arrangement				NU, N type	NJ, NF type	NUP, NP, NH type	NNU, NN type	NA type
Reference page	B-5	B-43	B-74	B-43	B-79	B-91	B-91	B-91	B-116	E-2

Tapered roller bearings	Double-row, 4-row tapered roller bearings	Spherical roller bearings	Thrust ball bearings	Double row angular contact thrust ball bearings	Cylindrical roller thrust bearings	Spherical roller thrust bearings	Reference page	Bearing types / Characteristics
								Load Carrying Capacity (Radial load / Axial load)
☆☆☆	☆☆	☆☆	☆	☆☆☆	☆	☆	A-66	High speed❶
☆☆☆	☆		☆	☆☆☆			A-31	High rotating accuracy❶
			☆				—	Low noise/vibration❶
							A-67	Low friction torque❶
☆☆	☆☆☆☆	☆☆☆		☆☆	☆☆☆	☆☆☆	A-54	High rigidity❶
☆☆	☆☆☆	☆☆☆		★	☆☆☆	☆☆☆	A-18	Vibration/shock resistance❶
☆		☆☆☆		★	★	☆☆☆	A-79	Allowable misalignment for inner/outer rings❶
○	◎	◎	○	◎	○	○	A-13	Stationary in axial direction❷
	○	○	○				A-13	Moveable in axial direction❸
○	○		○	○	○	○	—	Separable inner/outer rings❹
		○					A-79	Inner ring tapered bore❺
For duplex arrangement					Including needle roller thrust bearing		—	Remarks
B-133	B-133	B-233	B-269	B-269	E-48	B-269		Reference page

❶ ☆ The number of stars indicates the degree to which that bearing type displays that particular characteristic.
★ Not applicable to that bearing type.
❷ ◎ Indicates dual direction. ○ Indicates single direction axial movement only.
❸ ◎ indicates movement in the axial direction is possible for the raceway surface; ○ indicates movement in the axial direction is possible for the fitting surface of the outer ring or inner ring.
❹ ○ Indicates both inner ring and outer ring are detachable.
❺ ○ Indicates inner ring with tapered bore is possible.

2.3 Selection of bearing arrangement

Shafts or axles are generally supported by a pair of bearings in the axial and radial directions. The bearing which prevents axial movement of the shaft relative to the housing is called the "**fixed side bearing**" and the bearing which allows axial movement relatively is called the "**floating-side bearing**". This allows for expansion and contraction of the shaft due to temperature variation and enables error in bearing mounting clearance to be absorbed.

The **fixed side bearing** is able to support radial and axial loads. A bearing which can fix axial movement in both directions should therefore be selected. A **floating-side bearing** that allows movement in the axial direction while supporting a radial load is desirable. Movement in the axial direction occurs on the raceway surface for bearings with separable inner and outer rings such as cylindrical roller bearings, and occurs on the fitting surface for those which are not separable, such as deep groove ball bearings.

In applications with short distances between bearings, shaft expansion and contraction due to temperature fluctuations is slight, therefore the same type of bearing may be used for both the fixed-side and floating-side bearing. In such cases it is common to use a set of matching bearings, such as angular contact ball bearings, to guide and support the shaft in one axial direction only.

Table 2.2 (1) shows typical bearing arrangements where the bearing type differs on the fixed side and floating side. **Table 2.2 (2)** shows some common bearing arrangements where no distinction is made between the fixed side and floating side. Vertical shaft bearing arrangements are shown in **Table 2.2 (3)**.

Table 2.2 (1) Bearing arrangement (distinction between fixed and floating-side)

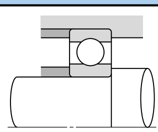
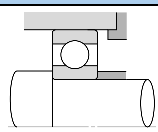
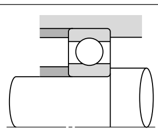
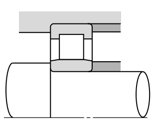
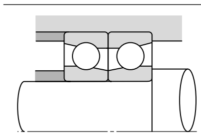
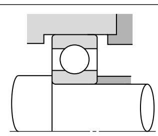
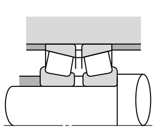
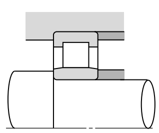
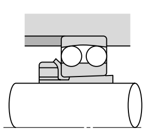
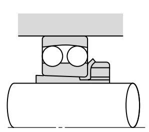
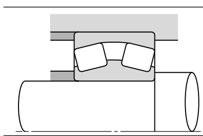
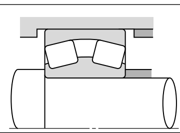
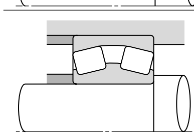
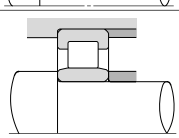
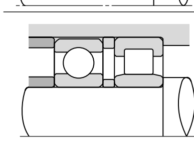
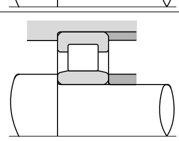
Arrangement		Comment	Application (Reference)
Fixed	Floating		
		1. General arrangement for small machinery. 2. For radial loads, but will also accept axial loads.	Small pumps, auto-mobile transmissions, etc.
		1. Suitable when mounting error and shaft deflection are minimal or used for high rotational speed application. 2. Even with expansion and contraction of shaft, floating side moves smoothly.	Medium-sized electric motors, ventilators, etc.
		1. Radial loading and dual direction of axial loading possible. 2. In place of duplex angular contact ball bearings, double-row angular contact ball bearings are also used.	Worm reduction gear
		1. Heavy loading capable. 2. Shafting rigidity increased by preloading the two back-to-back fixed bearings. 3. Requires high precision shafts and housings, and minimal fitting errors.	Reduction gears for general industrial machinery
		1. Allows for shaft deflection and fitting errors. 2. By using an adaptor on long shafts without screws or shoulders, bearing mounting and dismounting can be facilitated. 3. Self-aligning ball bearings are used for positioning in the axial direction, and not suitable for applications requiring support of axial load.	General industrial machinery
		1. Widely used in general industrial machinery with heavy and shock load demands. 2. Allows for shaft deflection and fitting errors. 3. Accepts radial loads as well as dual direction of axial loads.	Reduction gears for general industrial machinery
		1. Accepts radial loads as well as dual direction axial loads. 2. Suitable when both inner and outer ring require tight fit.	Reduction gears for general industrial machinery
		1. Capable of handling large radial and axial loads at high rotational speeds. 2. Maintains clearance between the bearing's outer diameter and housing inner diameter to prevent deep groove ball bearings from receiving radial loads.	Transmissions for diesel locomotives

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Table 2.2 (2) Bearing arrangement (no distinction between fixed and floating-side)

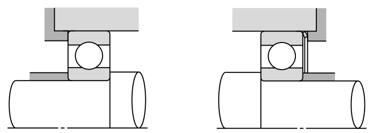
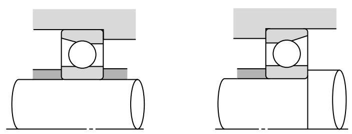
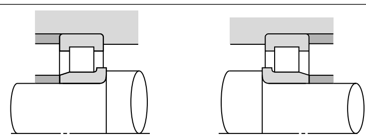
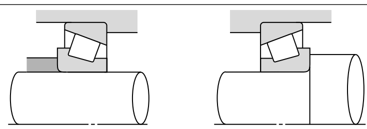
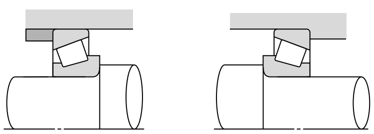
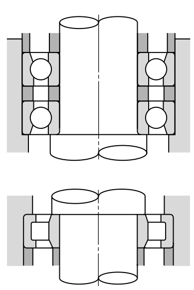
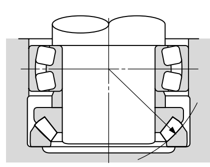
Arrangement	Comment	Application (Reference)
	1. General arrangement for use in small machines. 2. Preload is sometimes applied by placing a spring on the outer ring side surface or inserting a shim. (can be floating-side bearings.)	Small electric motors, small reduction gears, etc.
	1. Back to back arrangement is preferable to face to face arrangement when moment load applied. 2. Able to support axial and radial loads; suitable for high-speed rotation. 3. Rigidity of shaft can be enhanced by providing preload.	Machine tool spindles, etc.
	1. Capable of supporting extra heavy loads and impact loads. 2. Suitable if inner and outer ring tight fit is required. 3. Care must be taken that axial clearance does not become too small during operation.	Construction equipment, mining equipment sheaves, agitators, etc.
 Back to back  Face to face	1. Withstands heavy and shock loads. Wide range application. 2. Shaft rigidity can be enhanced by providing preload, but make sure preload is not excessive. 3. Back-to-back arrangement for moment loads, and face-to-face arrangement to alleviate fitting errors. 4. With face-to-face arrangement, inner ring tight fit is facilitated.	Reduction gears, front and rear axle of automobiles, etc.

Table 2.2 (3) Bearing arrangement (Vertical shaft)

Arrangement	Comment	Application (Reference)
	1. When fixing bearing is a duplex angular contact ball bearing, floating bearing should be a cylindrical roller bearing.	Vertically mounted electric motors, etc.
	1. Most suitable arrangement for very heavy axial loads. 2. Shaft deflection and mounting error can be absorbed by matching the center of the spherical surface with the center of spherical roller thrust bearings.	Crane center shafts, etc.

3. Load Rating and Life

3.1 Bearing life

Even in bearings operating under normal conditions, the surfaces of the raceway and rolling elements are constantly being subjected to repeated compressive stresses which causes flaking of these surfaces to occur. This flaking is due to material fatigue and will eventually cause the bearings to fail. The effective life of a bearing is usually defined in terms of the total number of revolutions a bearing can undergo before flaking of either the raceway surface or the rolling element surfaces occurs.

Other causes of bearing failure are often attributed to problems such as seizing, abrasions, cracking, chipping, scuffing, rust, etc. However, these so called "causes" of bearing failure are usually themselves caused by improper installation, insufficient or improper lubrication, faulty sealing or inaccurate bearing selection. Since the above mentioned "causes" of bearing failure can be avoided by taking the proper precautions, and are not simply caused by material fatigue, they are considered separately from the flaking aspect.

3.2 Basic rating life and basic dynamic load rating

A group of seemingly identical bearings when subjected to identical load and operating conditions will exhibit a wide diversity in their durability.

This "life" disparity can be accounted for by the difference in the fatigue of the bearing material itself. This disparity is considered statistically when calculating bearing life, and the basic rating life is defined as follows.

The basic rating life is based on a 90% statistical model which is expressed as the total number of revolutions 90% of the bearings in an identical group of bearings subjected to identical operating conditions will attain or surpass before flaking due to material fatigue occurs. For bearings operating at fixed constant speeds, the basic rating life (90% reliability) is expressed in the total number of hours of operation.

Basic dynamic load rating expresses a rolling bearing's capacity to support a dynamic load. The basic dynamic load rating is the load under which the basic rating life of the bearing is 1 million revolutions. This is expressed as pure radial load for radial bearings and pure axial load for thrust bearings. These are referred to as "basic dynamic load rating (C_r)" and "basic dynamic axial load rating (C_a)." The basic dynamic load ratings given in the bearing tables of this catalog are for bearings constructed of NTN standard bearing materials, using standard manufacturing techniques.

The relationship between the basic rating life, the basic dynamic load rating and the bearing load is given in formula.

For ball bearings: $$L_{10} = \left(\frac{C}{P}\right)^3 \quad \cdots\cdots\cdots\cdots\cdots (3.1)$$

For roller bearings: $$L_{10} = \left(\frac{C}{P}\right)^{10/3} \quad \cdots\cdots\cdots\cdots (3.2)$$

where,

L_{10} : Basic rating life 10^6 revolutions
C : Basic dynamic load rating, N {kgf}
(C_r: radial bearings, C_a: thrust bearings)
P : Equivalent dynamic load, N {kgf}
(P_r: radial bearings, P_a: thrust bearings)
n : Rotational speed, min^{-1}

The relationship between Rotational speed n and speed factor f_n as well as the relation between the basic rating life L_{10h} and the life factor f_n is shown in **Table 3.1** and **Fig. 3.1.**

Table 3.1 Correlation of bearing basic rating life, life factor, and speed factor

Classification	Ball bearing	Roller bearing
Basic rating life L_{10h} h	$\frac{10^6}{60n}\left(\frac{C}{P}\right)^3 = 500\, f_h^3$	$\frac{10^6}{60n}\left(\frac{C}{P}\right)^{10/3} = 500\, f_h^{10/3}$
Life factor f_h	$f_n \frac{C}{P}$	$f_n \frac{C}{P}$
Speed factor f_n	$\left(\frac{33.3}{n}\right)^{1/3}$	$\left(\frac{33.3}{n}\right)^{3/10}$

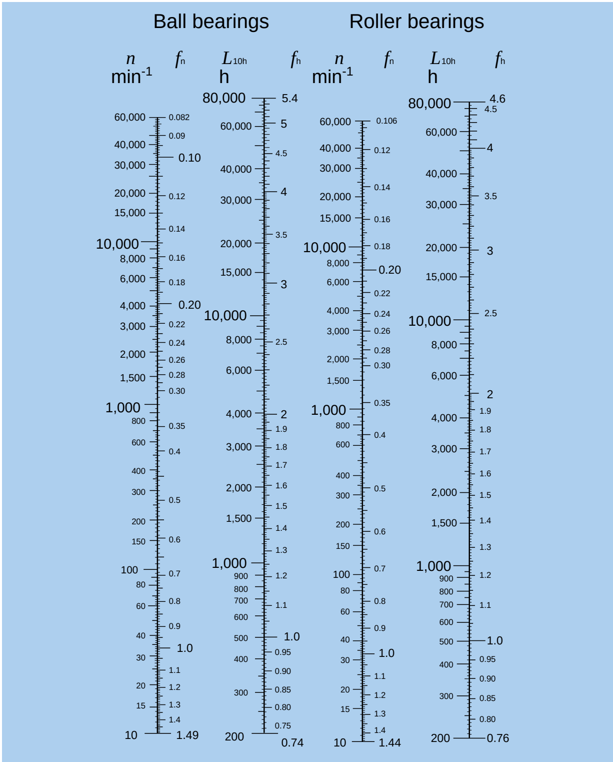


Fig. 3.1 Bearing life rating scale

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When several bearings are incorporated in machines or equipment as complete units, all the bearings in the unit are considered as a whole when computing bearing life (see formula 3.3).

$$L = \frac{1}{\left(\frac{1}{L_1^{e}} + \frac{1}{L_2^{e}} + \cdots \frac{1}{L_n^{e}}\right)^{1/e}} \cdots\cdots\cdots\cdots (3.3)$$

where,

L : Total basic rating life of entire unit, h

L_1 , L_2 $\cdots L_n$: Basic rating life of individual bearings, 1, 2, ⋯n, h

e = 10/9....................For ball bearings

e = 9/8......................For roller bearings

When the load conditions vary at regular intervals, the life can be given by formula (3.4).

$$L_m = \left(\frac{\Phi_1}{L_1} + \frac{\Phi_2}{L_2} + \cdots \frac{\Phi_j}{L_j}\right)^{-1} \cdots\cdots\cdots\cdots (3.4)$$

where,

L_m : Total life of bearing

Φ_j : Frequency of individual load conditions ($\Sigma\Phi_j = 1$)

L_j : Life under individual conditions

If equivalent load P and rotational speed n are operating conditions of the bearing, basic rated dynamic load C that satisfies required life of the bearing is determined using **Table 3.1** and formula (3.5). Bearings that satisfy the required C can be selected from the bearing dimensions table provided in the catalog.

$$C = P\frac{f_h}{f_n} \cdots\cdots\cdots\cdots (3.5)$$

3.3 Adjusted rating life

The basic bearing rating life (90% reliability factor) can be calculated through the formulas mentioned earlier in Section 3.2. However, in some applications a bearing life factor of over 90% reliability may be required. To meet these requirements, bearing life can be lengthened by the use of specially improved bearing materials or manufacturing process. Bearing life is also sometimes affected by operating conditions such as lubrication, temperature and rotational speed.

Basic rating life adjusted to compensate for this is called "adjusted rating life," and is determined using formula (3.6).

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot L_{10} \cdots \quad (3.6)$$

where,

L_{na} : Adjusted rating life in millions of revolutions (10^6)

a_1 : Reliability factor

a_2 : Bearing characteristics factor

a_3 : Operating conditions factor

3.3.1 Reliability factor a_1

The value of reliability factor a_1 is provided in **Table 3.2** for reliability of 90% or greater.

3.3.2 Bearing characteristics factor a_2

Bearing characteristics concerning life vary according to bearing material, quality of material and if using special manufacturing process. In this case, life is adjusted using bearing characteristics factor a_2.

The basic dynamic load ratings listed in the catalog are based on NTN's standard material and process, therefore, the adjustment factor $a_2 = 1$. $a_2 > 1$ may be used for specially enhanced materials and manufacturing methods.If this applies, consult with NTN Engineering.

Dimensions change significantly if bearings made of high carbon chrome bearing steel with conventional heat treatment are used at temperatures in excess of 120˚C for an extended period of time. NTN Engineering therefore offers a bearing for high-temperature applications specially treated to stabilize dimensions at the maximum operating temperature (TS treatment). The treatment however makes the bearing softer and affects life of the bearing. Life is adjusted by multiplying by the values given in **Table 3.3**.

Table 3.2 Reliability factor a_1

Reliability %	L_n	Reliability factor a_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21

Table 3.3 Treatment for stabilizing dimensions

Symbol	Max. operating temperature (C˚)	Bearing characteristics factor a_2
TS2	160	1.00
TS3	200	0.73
TS4	250	0.48

3.3.3 Operating conditions factor a_3

Operating conditions factor a_3 is used to compensate for when lubrication condition worsens due to rise in temperature or rotational speed, lubricant deteriorates, or becomes contaminated with foreign matter.

Generally speaking, when lubricating conditions are satisfactory, the a_3 factor has a value of one; and when lubricating conditions are exceptionally favorable, and all other operating conditions are normal, a_3 can have a value greater than one. a_3 is however less than 1 in the following cases:

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- Dynamic viscosity of lubricating oil is too low for bearing operating temperature
  (13 mm^2/s or less for ball bearings, 20 mm^2/s for roller bearings)
- Rotational speed is particularly low
  (If sum of rotational speed n min^{-1} and rolling element pitch diameter D_{pw} mm is $D_{pw} \cdot n < 10{,}000$)
- Bearing operating temperature is too high
  If bearing operating temperature is too high, the raceway becomes softened, thereby shortening life. Life is adjusted by multiplying by the values given in **Fig. 3.2** as the operating condition factor according to operating temperature. This however does not apply to bearings that have been treated to stabilize dimensions.
- Lubricant contaminated with foreign matter or moisture
  If using special operating condition, consult with NTN Engineering. Even if $a_2 > 1$ is used for specially bearings made of enhanced materials or produced by special manufacturing methods, $a_2 \times a_3 < 1$ is used if lubricating conditions are not favorable.

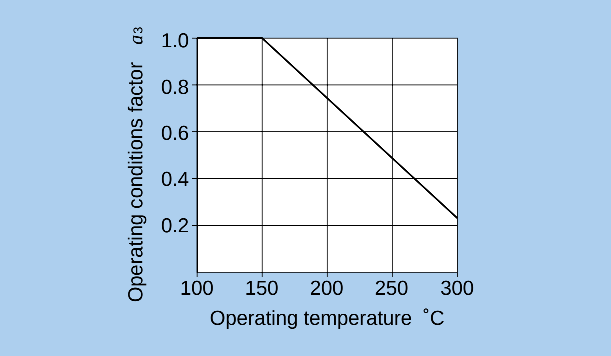

Fig. 3.2 Operating conditions factor according to operating temperature

When a super heavy load is applied, harmful plastic distortion could be produced on the contact surfaces of the rolling elements and raceway. The formulae for determining basic rating life (3.1, 3.2, and 3.6) do not apply if P_r exceeds either C_{or} (Basic static load rating) or 0.5 C_r for radial bearings, or if P_a exceeds 0.5 C_a for thrust bearings.

3.4 Machine applications and requisite life

When selecting a bearing, it is essential that the requisite life of the bearing be established in relation to the operating conditions. The requisite life of the bearing is usually determined by the type of machine in which the bearing will be used, and duration of service and operational reliability requirements. A general guide to these requisite life criteria is shown in **Table 3.4**. When determining bearing size, the fatigue life of the bearing is an important factor; however, besides bearing life, the strength and rigidity of the shaft and housing must also be taken into consideration.

3.5 Basic static load rating

When stationary rolling bearings are subjected to static loads, they suffer from partial permanent deformation of the contact surfaces at the contact point between the rolling elements and the raceway. The amount of deformity increases as the load increases, and if this increase in load exceeds certain limits, the subsequent smooth operation of the bearings is impaired.

It has been found through experience that a permanent deformity of 0.0001 times the diameter of the rolling element, occurring at the most heavily stressed contact point between the raceway and the rolling elements, can be tolerated without any impairment in running efficiency.

Table 3.4 Machine application and requisite life (reference)

Service classification	Machine application and requisite life (reference) L_{10h} $\times 10^3$ h				
	～4	4～12	12～30	30～60	60～
Machines used for short periods or used only occasionally	● Household appliances ● Electric hand tools	● Farm machinery ● Office equipment			
Short period or intermittent use, but with high reliability requirements	● Medical appliances ● Measuring instruments	● Home air-conditioning motor ● Construction equipment ● Elevators ● Cranes	● Crane (sheaves)		
Machines not in constant use, but used for long periods	● Automobiles ● Two-wheeled vehicles	● Small motors ● Buses/trucks ● General gear drives ● Woodworking machines	● Machine spindles ● Industrial motors ● Crushers ● Vibrating screens	● Main gear drives ● Rubber/plastic ● Calender rolls ● Printing machines	
Machines in constant use over 8 hours a day		● Rolling mills ● Escalators ● Conveyors ● Centrifuges	● Railway vehicle axles ● Air conditioners ● Large motors ● Compressor pumps	● Locomotive axles ● Traction motors ● Mine hoists ● Pressed flywheels	● Papermaking machines ● Propulsion equipment for marine vessels
24 hour continuous operation, non-interruptable					● Water supply equipment ● Mine drain pumps/ventilators ● Power generating equipment

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The basic static load rating refers to a fixed static load limit at which a specified amount of permanent deformation occurs. It applies to pure radial loads for radial bearings and to pure axial loads for thrust bearings. The maximum applied load values for contact stress occurring at the rolling element and raceway contact points are given below.

For ball bearings	4,200 MPa {428kgf/mm^2}
For self-aligning ball bearings	4,600 MPa {469kgf/mm^2}
For roller bearings	4,000 MPa {408kgf/mm^2}

Referred to as "basic static radial load rating" for radial bearings and "basic static axial load rating" for thrust bearings, basic static load rating is expressed as C_{or} or C_{oa} respectively and is provided in the bearing dimensions table.

3.6 Allowable static equivalent load

Generally the static equivalent load which can be permitted (See page A-25) is limited by the basic static rating load as stated in **Section 3.5**. However, depending on requirements regarding friction and smooth operation, these limits may be greater or lesser than the basic static rating load.

This is generally determined by taking the safety factor So given in **Table 3.5** and formula (3.7) into account.

$$S_o = C_o / P_o \cdots \quad (3.7)$$

where,

S_o : Safety factor

C_o : Basic static load rating, N {kgf}
(radial bearings: C_{or}, thrust bearings: C_{oa})

P_o : Static equivalent load, N {kgf}
(radial: P_{or}, thrust: C_{oa})

Table 3.5 Minimum safety factor values S_0

Operating conditions	Ball bearings	Roller bearings
High rotational accuracy demand	2	3
Normal rotating accuracy demand (Universal application)	1	1.5
Slight rotational accuracy deterioration permitted (Low speed, heavy loading, etc.)	0.5	1

Note 1: For spherical thrust roller bearings, min. S_0 value=4.
2: For shell needle roller bearings, min. S_0 value=3.
3: When vibration and/or shock loads are present, a load factor based on the shock load needs to be included in the P_0 max value.
4: If a large axial load is applied to deep groove ball bearings or angular ball bearings, the contact oval may exceed the raceway surface. For more information, please contact NTN Engineering.

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4. Bearing Load Calculation

To compute bearing loads, the forces which act on the shaft being supported by the bearing must be determined. Loads which act on the shaft and its related parts include dead load of the rotator, load produced when the machine performs work, and load produced by transmission of dynamic force. These can theoretically be mathematically calculated, but calculation is difficult in many cases.

A method of calculating loads that act upon shafts that convey dynamic force, which is the primary application of bearings, is provided herein.

4.1 Load acting on shafts

4.1.1 Load factor

There are many instances where the actual operational shaft load is much greater than the theoretically calculated load, due to machine vibration and/or shock. This actual shaft load can be found by using formula (4.1).

$$K = f_w \cdot K_c \quad \cdots\cdots (4.1)$$

where,

K : Actual shaft load N {kgf}
f_w : Load factor (**Table 4.1**)
K_c : Theoretically calculated value N {kgf}

Table 4.1 Load factor f_w

Amount of shock	f_w	Application
Very little or no shock	1.0∼1.2	Electric machines, machine tools, measuring instruments.
Light shock	1.2∼1.5	Railway vehicles, automobiles, rolling mills, metal working machines, paper making machines, printing machines, aircraft, textile machines, electrical units, office machines.
Heavy shock	1.5∼3.0	Crushers, agricultural equipment, construction equipment, cranes.

4.1.2 Gear load

The loads operating on gears can be divided into three main types according to the direction in which the load is applied; i.e. tangential (K_t), radial (K_s), and axial (K_a). The magnitude and direction of these loads differ according to the types of gears involved. The load calculation methods given herein are for two general-use gear and shaft arrangements: parallel shaft gears, and cross shaft gears.

(1)Loads acting on parallel shaft gears

The forces acting on spur gears and helical gears are depicted in **Figs. 4.1**, **4.2**, and **4.3**. The load magnitude can be found by using or formulas (4.2), through (4.5).

$$\left.\begin{aligned} K_t &= \frac{19.1\times10^6 \cdot H}{D_p \cdot n} \quad \text{N} \\ &= \frac{1.95\times10^6 \cdot H}{D_p \cdot n} \quad \text{\{kgf\}} \end{aligned}\right\} \cdots\cdots (4.2)$$

$$K_s = K_t \cdot \tan\alpha \quad \text{(Spur gear)} \quad \cdots\cdots (4.3a)$$

$$= K_t \cdot \frac{\tan\alpha}{\cos\beta} \quad \text{(Helical gear)} \quad \cdots\cdots (4.3b)$$

$$K_r = \sqrt{K_t^2 + K_s^2} \quad \cdots\cdots (4.4)$$

$$K_a = K_t \cdot \tan\beta \quad \text{(Helical gear)} \quad \cdots\cdots (4.5)$$

where,

K_t : Tangential gear load (tangential force), N {kgf}
K_s : Radial gear load (separating force), N {kgf}
K_r : Right angle shaft load (resultant force of tangential force and separating force), N {kgf}
K_a : Parallel load on shaft, N {kgf}
H : Transmission force , kW
n : Rotational speed, min^{-1}
D_p : Gear pitch circle diameter, mm
α : Gear pressure angle, deg
β : Gear helix angle, deg

Because the actual gear load also contains vibrations and shock loads as well, the theoretical load obtained by the above formula should also be adjusted by the gear factor f_z as shown in **Table 4.2.**

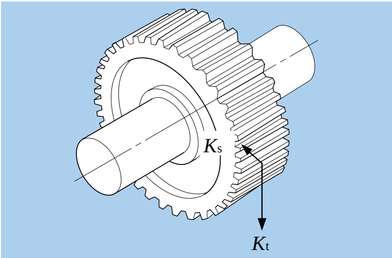

Fig. 4.1 Spur gear loads

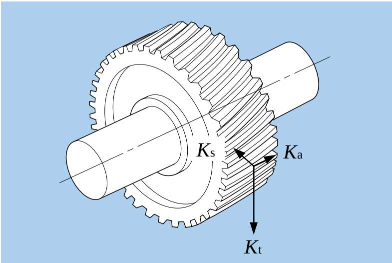

Fig. 4.2 Helical gear loads

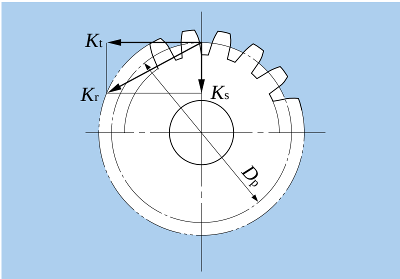

Fig. 4.3 Radial resultant forces

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Table 4.2 Gear factor f_z

Gear type	f_z
Precision ground gears (Pitch and tooth profile errors of less than 0.02 mm)	1.05～1.1
Ordinary machined gears (Pitch and tooth profile errors of less than 0.1 mm)	1.1～1.3

(2)Loads acting on cross shafts

Gear loads acting on straight tooth bevel gears and spiral bevel gears on cross shafts are shown in **Figs. 4.4** and **4.5**. The calculation methods for these gear loads are shown in **Table 4.3**. Herein, to calculate gear loads for straight bevel gears, the helix angle $\beta = 0$.

The symbols and units used in **Table 4.3** are as follows:

K_t : Tangential gear load (tangential force), N {kgf}
K_s : Radial gear load (separating force), N {kgf}
K_a : Parallel shaft load (axial load), N {kgf}
H : Transmission force, kW
n : Rotational speed, min^{-1}
D_{pm} : Mean pitch circle diameter, mm
α : Gear pressure angle, deg
β : Helix angle, deg
δ : Pitch cone angle, deg

Because the two shafts intersect, the relationship of pinion and gear load is as follows:

$$K_{sp}=K_{ag} \cdots\cdots\cdots\cdots \text{(4.6)}$$
$$K_{ap}=K_{sg} \cdots\cdots\cdots\cdots \text{(4.7)}$$

where,
K_{sp}, K_{sg} : Pinion and gear separating force, N {kgf}
K_{ap}, K_{ag} : Pinion and gear axial load, N {kgf}

For spiral bevel gears, the direction of the load varies depending on the direction of the helix angle, the direction of rotation, and which side is the driving side or the driven side. The directions for the separating force (K_s) and axial load (K_a) shown in **Fig. 4.5** are positive directions. The direction of rotation and the helix angle direction are defined as viewed from the large end of the gear. The gear rotation direction in **Fig. 4.5** is assumed to be clockwise (right).

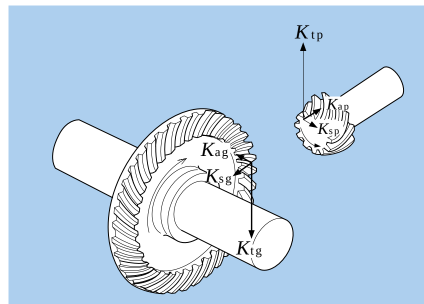

Fig. 4.4 Loads on bevel gears

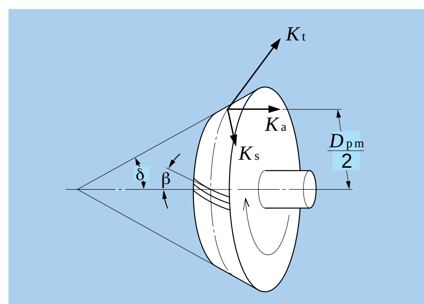

Fig. 4.5 Bevel gear diagram

Table 4.3 Loads acting on bevel gears

Types of load	Rotation direction	Clockwise	Counter clockwise	Clockwise	Counter clockwise
	Helix direction	Right	Left	Left	Right
Tangential load (tangential force) K_t		$K_t=\dfrac{19.1\times10^6\cdot H}{D_{pm}\cdot n}$, $\left\{\dfrac{1.95\times10^6\cdot H}{D_{pm}\cdot n}\right\}$			
Radial load (separation force) K_s	Driving side	$K_s=K_t\left[\tan\alpha\dfrac{\cos\delta}{\cos\beta}+\tan\beta\sin\delta\right]$		$K_s=K_t\left[\tan\alpha\dfrac{\cos\delta}{\cos\beta}-\tan\beta\sin\delta\right]$	
	Driven side	$K_s=K_t\left[\tan\alpha\dfrac{\cos\delta}{\cos\beta}-\tan\beta\sin\delta\right]$		$K_s=K_t\left[\tan\alpha\dfrac{\cos\delta}{\cos\beta}+\tan\beta\sin\delta\right]$	
Parallel load on gear shaft (axial load) K_a	Driving side	$K_a=K_t\left[\tan\alpha\dfrac{\sin\delta}{\cos\beta}-\tan\beta\cos\delta\right]$		$K_a=K_t\left[\tan\alpha\dfrac{\sin\delta}{\cos\beta}+\tan\beta\cos\delta\right]$	
	Driven side	$K_a=K_t\left[\tan\alpha\dfrac{\sin\delta}{\cos\beta}+\tan\beta\cos\delta\right]$		$K_a=K_t\left[\tan\alpha\dfrac{\sin\delta}{\cos\beta}-\tan\beta\cos\delta\right]$	

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4.1.3 Chain / belt shaft load

The tangential loads on sprockets or pulleys when power (load) is transmitted by means of chains or belts can be calculated by formula (4.8).

$$\left.\begin{aligned} K_t &= \frac{19.1\times10^6\cdot H}{D_p\cdot n} \quad \text{N} \\ &= \frac{1.95\times10^6\cdot H}{D_p\cdot n} \quad \{\text{kgf}\} \end{aligned}\right\} \cdots\cdots\cdots\cdots\cdots (4.8)$$

where,

K_t : Sprocket/pulley tangential load, N {kgf}

H : Transmitted force, kW

D_p : Sprocket/pulley pitch diameter, mm

For belt drives, an initial tension is applied to give sufficient constant operating tension on the belt and pulley. Taking this tension into account, the radial loads acting on the pulley are expressed by formula (4.9). For chain drives, the same formula can also be used if vibrations and shock loads are taken into consideration.

$$K_r = f_b \cdot K_t \cdots \quad (4.9)$$

where,

K_r : Sprocket or pulley radial load, N {kgf}

f_b : Chain or belt factor (**Table 4.4**)

Table. 4.4 chain or belt factor f_b

Chain or belt type	f_b
Chain (single)	1.2〜1.5
V-belt	1.5〜2.0
Timing belt	1.1〜1.3
Flat belt (w / tension pulley)	2.5〜3.0
Flat belt	3.0〜4.0

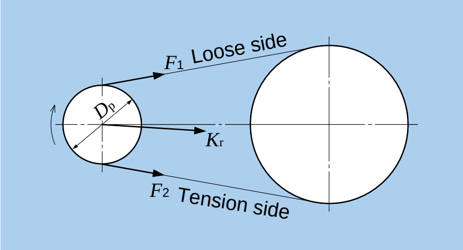


Fig. 4.6 Chain / belt loads

4.2 Bearing load distribution

For shafting, the static tension is considered to be supported by the bearings, and any loads acting on the shafts are distributed to the bearings.

For example, in the gear shaft assembly depicted in **Fig. 4.7**, the applied bearing loads can be found by using formulas (4.10) and (4.11).

This example is a simple case, but in reality, many of the calculations are quite complicated.

$$F_{rA} = \frac{a+b}{b} F_{\text{I}} + \frac{d}{c+d} F_{\text{II}} \cdots\cdots\cdots\cdots\cdots (4.10)$$

$$F_{rB} = -\frac{a}{b} F_{\text{I}} + \frac{c}{c+d} F_{\text{II}} \cdots\cdots\cdots\cdots\cdots (4.11)$$

where,

F_{rA} : Radial load on bearing A, N {kgf}

F_{rB} : Radial load on bearing B, N {kgf}

F_{I}, F_{II} : Radial load on shaft, N {kgf}

If directions of radial load differ, the vector sum of each respective load must be determined.

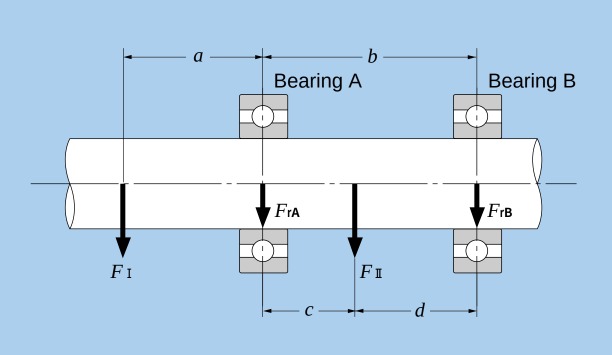


Fig. 4.7

II

4.3 Mean load

The load on bearings used in machines under normal circumstances will, in many cases, fluctuate according to a fixed time period or planned operation schedule. The load on bearings operating under such conditions can be converted to a mean load (F_m), this is a load which gives bearings the same life they would have under constant operating conditions.

(1) Fluctuating stepped load

The mean bearing load, F_m, for stepped loads is calculated from formula (4.12). F_1 , F_2 .... F_n are the loads acting on the bearing; n_1, n_2.......n_n and t_1, t_2....... t_n are the bearing speeds and operating times respectively.

$$F_m = \left[\frac{\Sigma (F_i{}^p \, n_i \, t_i)}{\Sigma (n_i t_i)} \right]^{1/p} \cdots\cdots\cdots\cdots\cdots\cdots\cdots \quad (4.12)$$

where:

$p = 3$ For ball bearings
$p = 10/3$ For roller bearings

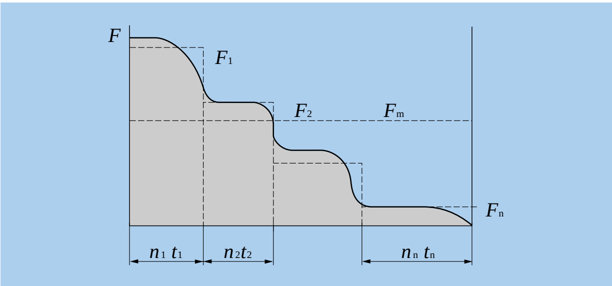


Fig. 4.8 Stepped load

(2) Continuously fluctuating load

Where it is possible to express the function $F(t)$ in terms of load cycle t_o and time t, the mean load is found by using formula (4.13).

$$F_m = \left[\frac{1}{t_o} \int_o^{t_o} F(t)^p \, d_t \right]^{1/p} \cdots\cdots\cdots\cdots\cdots\cdots \quad (4.13)$$

where:

$p = 3$ For ball bearings
$p = 10/3$ For roller bearings

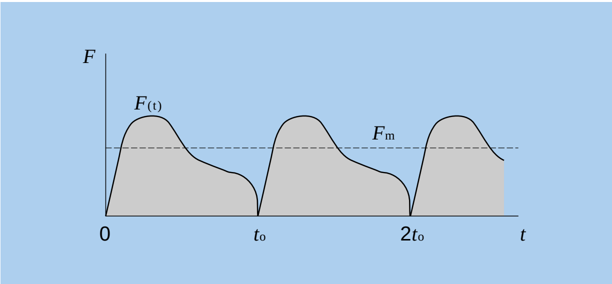


Fig. 4.9 Load that fluctuated as function of time

(3) Linear fluctuating load

The mean load, F_m, can be approximated by formula (4.14).

$$F_m = \frac{F_{min} + 2F_{max}}{3} \cdots \quad (4.14)$$

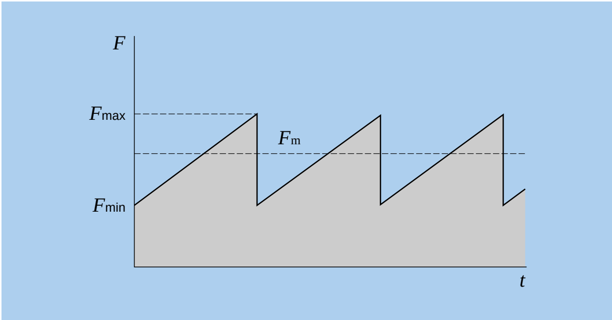


Fig. 4.10 Linear fluctuating load

(4) Sinusoidal fluctuating load

The mean load, F_m, can be approximated by formulas (4.15) and (4.16).

case (a) $F_m = 0.75 F_{max}$ ········· (4.15)
case (b) $F_m = 0.65 F_{max}$ ········· (4.16)

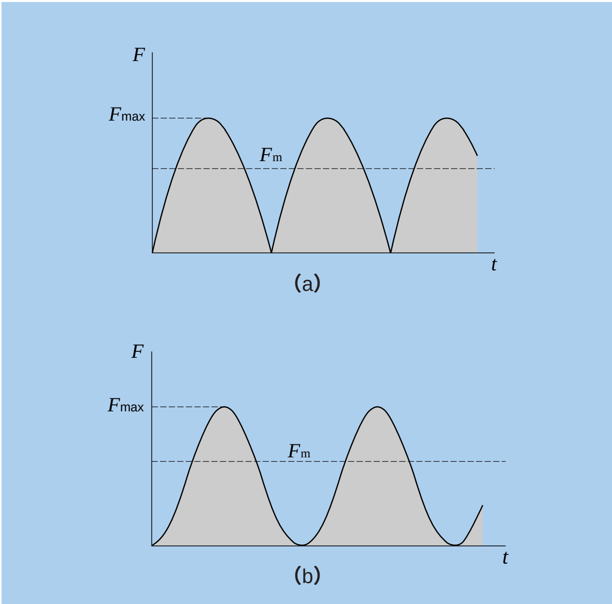


Fig. 4.11 Sinusoidal variable load

II

4.4 Equivalent load

4.4.1 Dynamic equivalent load

When both dynamic radial loads and dynamic axial loads act on a bearing at the same time, the hypothetical load acting on the center of the bearing which gives the bearings the same life as if they had only a radial load or only an axial load is called the dynamic equivalent load.

For radial bearings, this load is expressed as pure radial load and is called the dynamic equivalent radial load. For thrust bearings, it is expressed as pure axial load, and is called the dynamic equivalent axial load.

(1) Dynamic equivalent radial load

The dynamic equivalent radial load is expressed by formula (4.17).

$$P_r = XF_r + YF_a \cdots\cdots\cdots\cdots\cdots\cdots \quad (4.17)$$

where,

P_r : Dynamic equivalent radial load, N {kgf}
F_r : Actual radial load, N {kgf}
F_a : Actual axial load, N {kgf}
X : Radial load factor
Y : Axial load factor

The values for X and Y are listed in the bearing tables.

(2) Dynamic equivalent axial load

As a rule, standard thrust bearings with a contact angle of 90° cannot carry radial loads. However, self-aligning thrust roller bearings can accept some radial load. The dynamic equivalent axial load for these bearings is given in formula (4.18).

$$P_a = F_a + 1.2F_r \cdots\cdots\cdots\cdots\cdots\cdots \quad (4.18)$$

where,

P_a : Dynamic equivalent axial load, N {kgf}
F_a : Actual axial load, N {kgf}
F_r : Actual radial load, N {kgf}

Provided that $F_r / F_a \leqq 0.55$ only.

4.4.2 Static equivalent load

The static equivalent load is a hypothetical load which would cause the same total permanent deformation at the most heavily stressed contact point between the rolling elements and the raceway as under actual load conditions; that is when both static radial loads and static axial loads are simultaneously applied to the bearing.

For radial bearings this hypothetical load refers to pure radial loads, and for thrust bearings it refers to pure centric axial loads. These loads are designated static equivalent radial loads and static equivalent axial loads respectively.

(1) Static equivalent radial load

For radial bearings the static equivalent radial load can be found by using formula (4.19) or (4.20). The greater of the two resultant values is always taken for P_{or}.

$$P_{or} = X_o F_r + Y_o F_a \cdots \quad (4.19)$$

$$P_{or} = F_r \cdots\cdots\cdots\cdots\cdots \quad (4.20)$$

where,

P_{or} : Static equivalent radial load, N {kgf}
F_r : Actual radial load, N {kgf}
F_a : Actual axial load, N {kgf}
X_o : Static radial load factor
Y_o : Static axial load factor

The values for X_o and Y_o are given in the respective bearing tables.

(2) Static equivalent axial load

For spherical thrust roller bearings the static equivalent axial load is expressed by formula (4.21).

$$P_{oa} = F_a + 2.7F_r \cdots \quad (4.21)$$

where,

P_{oa} : Static equivalent axial load, N {kgf}
F_a : Actual axial load, N {kgf}
F_r : Actual radial load, N {kgf}

Provided that $F_r / F_a \leqq 0.55$ only.

4.4.3 Load calculation for angular contact ball bearings and tapered roller bearings

For angular contact ball bearings and tapered roller bearings the pressure cone apex (load center) is located as shown in **Fig. 4.12**, and their values are listed in the bearing tables.

When radial loads act on these types of bearings the component force is induced in the axial direction. For this reason, these bearings are used in pairs. For load calculation this component force must be taken into consideration and is expressed by formula (4.22).

$$F_a = \frac{0.5F_r}{Y} \cdots\cdots\cdots\cdots\cdots\cdots\cdots \quad (4.22)$$

where,

F_a: Axial component force, N {kgf}
F_r: Radial load, N {kgf}
Y: Axial load factor

The dynamic equivalent radial loads for these bearing pairs are given in **Table 4.5**.

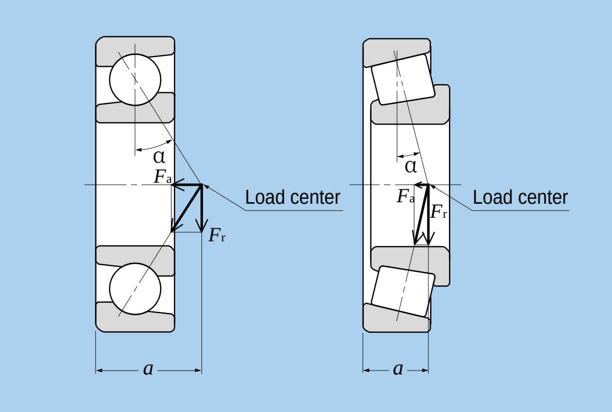
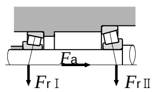
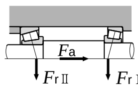
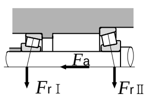
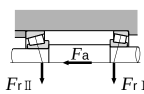


Fig. 4.12 Pressure cone apex and axial component force

II

Table 4.5 Bearing arrangement and dynamic equivalent load

Bearing arrangement	Load condition	Axial load	Dynamic equivalent radial load
Rear Brg I Brg II 	$\frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} \leqq \frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} + F_a$	$F_{a\,\mathrm{I}} = \frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} + F_a$	$P_{r\,\mathrm{I}} = XF_{r\,\mathrm{I}} + Y_{\mathrm{I}}\left(\frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} + F_a\right)$
		———	$P_{r\,\mathrm{II}} = F_{r\,\mathrm{II}}$
Front Brg II Brg I 	$\frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} > \frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} + F_a$	———	$P_{r\,\mathrm{I}} = F_{r\,\mathrm{I}}$
		$F_{a\,\mathrm{II}} = \frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} - F_a$	$P_{r\,\mathrm{II}} = XF_{r\,\mathrm{II}} + Y_{\mathrm{II}}\left(\frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} - F_a\right)$
Rear Brg I Brg II 	$\frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} \leqq \frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} + F_a$	———	$P_{r\,\mathrm{I}} = F_{r\,\mathrm{I}}$
		$F_{a\,\mathrm{II}} = \frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} + F_a$	$P_{r\,\mathrm{II}} = XF_{r\,\mathrm{II}} + Y_{\mathrm{II}}\left(\frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} + F_a\right)$
Front Brg II Brg I 	$\frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} > \frac{0.5F_{r\,\mathrm{I}}}{Y_{\mathrm{I}}} + F_a$	$F_{a\,\mathrm{I}} = \frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} - F_a$	$P_{r\,\mathrm{I}} = XF_{r\,\mathrm{I}} + Y_{\mathrm{I}}\left(\frac{0.5F_{r\,\mathrm{II}}}{Y_{\mathrm{II}}} - F_a\right)$
		———	$P_{r\,\mathrm{II}} = F_{r\,\mathrm{II}}$

Note 1: Applies when preload is zero.
2: Radial forces in the opposite direction to the arrow in the above illustration are also regarded as positive.

4.5 Bearing rating life and load calculation examples

In the examples given in this section, for the purpose of calculation, all hypothetical load factors as well as all calculated load factors may be presumed to be included in the resultant load values.

(Example 1)
What is the rating life in hours of operation (L_{10h}) for deep groove ball bearing **6208** operating at rotational speed n = 650 min^{-1}, with a radial load F_r of 3.2 kN {326 kgf} ?

From formula (4.17) the dynamic equivalent radial load:

$$P_r = F_r = 3.2\text{kN} \quad \{326\text{kgf}\}$$

Basic dynamic load rating C_r for bearing 6208 given on page B-12 is 29.1 kN {2970 kgf}, ball bearing speed factor fn relative to rotational speed n = 650 min^{-1} from **Fig. 3.1** is f_n = 0.37. Thus life factor f_h from formula (3.5) is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{29.1}{3.2} = 3.36$$

Therefore, with f_h = 3.36 from **Fig. 3.1** the rated life, L_{10h}, is approximately 19,000 hours.

(Example 2)
What is the life rating L_{10h} for the same bearing and conditions as in **Example 1**, but with an additional axial load F_a of 1.8 kN {184 kgf} ?

To find the dynamic equivalent radial load value for P_r, the radial load factor X and axial load factor Y are used. Basic static load rating C_{or} for bearing 6208 given on page B-12 is 17.8 kN {1820 kgf} and f_o is 14.0. Therefore:

$$\frac{f_o \cdot F_a}{C_{or}} = \frac{14 \times 1.8}{17.8} = 1.42$$

Calculating by the proportional interpolation method given on page B-13, e = 0.30.
For the operating radial load and axial load:

$$\frac{F_a}{F_r} = \frac{1.8}{3.2} = 0.56 > e = 0.30$$

From page B-13 X = 0.56 and Y = 1.44, and from formula (4.17) the equivalent radial load, P_r, is:

$$P_r = XF_r + YF_a = 0.56 \times 3.2 + 1.43 \times 1.8$$
$$= 4.38 \quad \text{kN} \quad \{447\text{kgf}\}$$

From Fig. 3.1 and formula (3.1) the life factor, f_h, is:

$$f_h = f_n \frac{C_r}{P_r} = 0.37 \times \frac{29.1}{4.38} = 2.46$$

Therefore, with life factor f_h = 2.46, from **Fig. 3.1** the rated life, L_{10h}, is approximately 7,500 hours.

(Example 3)
Determine the optimum model number for a cylindrical roller bearing operating at the rotational speed n = 450 min^{-1}, with a radial load F_r of 200 kN {20,400kgf}, and which must have a life (L_{10h}) of over 20,000 hours.

From **Fig. 3.1** the life factor f_h = 3.02 (L_{10h} at 20,000), and the speed factor f_n = 0.46 (n = 450 min^{-1}). To find the required basic dynamic load rating, C_r, formula (3.1) is used.

$$C_r = \frac{f_h}{f_n} P_r = \frac{3.02}{0.46} \times 200$$
$$= 1{,}313\text{kN} \quad \{134{,}000\text{kgf}\}$$

From page B-106, the smallest bearing that fulfills all the requirements is **NU2336** (C_r = 1,380 kN {141,000kgf}).

II

II

(Example 4)
The spur gear shown in **Fig. 4.13** (pitch diameter D_p = 150 mm, pressure angle α = 20°) is supported by a pair of tapered roller bearings, 4T-32206 (C_r = 54.5 kN {5,600 kgf}) and 4T-32205 (C_r = 42 kN {4300 kfg}). Find rating life for each bearing when gear transfer power H = 150 kW and rotational speed n = 2,000 min^{-1}.

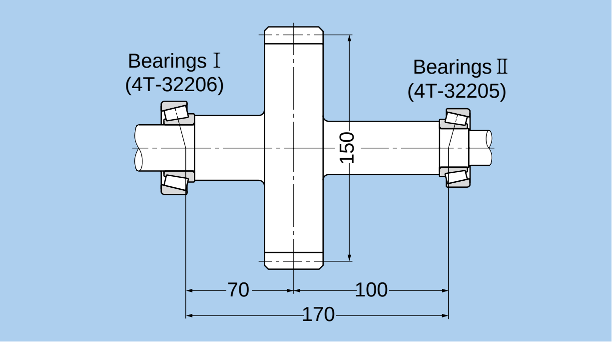


Fig. 4.13 Spur gear diagram

The gear load from formulas (4.2), (4.3a) and (4.4) is:

$$K_t = \frac{19.1\times10^6 \cdot H}{D_p \cdot n} = \frac{19,100\times150}{150\times2,000}$$
$$= 9.55\text{kN} \quad \{974\text{kgf}\}$$
$$K_s = K_t \cdot \tan\alpha = 9.55\times\tan20^\circ$$
$$= 3.48\text{kN} \quad \{355\text{kgf}\}$$
$$K_r = \sqrt{K_t^2 + K_s^2} = \sqrt{9.55^2 + 3.48^2}$$
$$= 10.16\text{kN} \quad \{1,040\text{kgf}\}$$

The radial loads for bearings I and II are:

$$F_{r\text{I}} = \frac{100}{170} K_r = \frac{100}{170} \times 10.16 = 5.98\text{kN} \quad \{610\text{kgf}\}$$

$$F_{r\text{II}} = \frac{70}{170} K_r = \frac{70}{170} \times 10.16 = 4.18\text{kN} \quad \{426\text{kgf}\}$$

$$\frac{0.5F_{r\text{I}}}{Y_\text{I}} = 1.87 > \frac{0.5F_{r\text{II}}}{Y_\text{II}} = 1.25$$

From **Table 4.5**, equivalent radial load:

$$P_{r\text{I}} = F_{r\text{I}} = 5.98\text{kN} \quad \{610\text{kgf}\}$$

$$P_{r\text{II}} = XF_{r\text{II}} + Y_\text{II}\frac{0.5F_{r\text{I}}}{Y_\text{I}}$$
$$= 0.4\times4.18 + 1.67\times1.87$$
$$= 4.79\text{kN} \quad \{489\text{kgf}\}$$

From formula (3.5) and **Fig. 3.1** the life factor, f_h, for each bearing is:

$$f_{h\text{I}} = f_n\frac{C_{r\text{I}}}{P_{r\text{I}}} = 0.293\times54.5/5.98 = 2.67$$

$$f_{h\text{II}} = f_n\frac{C_{r\text{II}}}{P_{r\text{II}}} = 0.293\times42.0/4.79 = 2.57$$

Therefore: a_2 = 1.4 (4T-tapered roller bearings shown in **B-144**)

$$L_{h1} = 13,200\times a_2$$
$$= 13,200\times1.4$$
$$= 18,480 \text{ hour}$$
$$L_{h2} = 11,600\times a_2$$
$$= 11,600\times1.4$$
$$= 16,240 \text{ hour}$$

The combined bearing life, L_h, from formula (3.3) is:

$$L_h = \frac{1}{\left[\frac{1}{L_{h1}{}^e} + \frac{1}{L_{h2}{}^e}\right]^{1/e}}$$

$$= \frac{1}{\left[\frac{1}{18,480^{9/8}} + \frac{1}{16,240^{9/8}}\right]^{8/9}}$$

$$= 9,330 \text{ hour}$$

(Example 5)
Find the mean load for spherical roller bearing **23932** (L_a = 320 kN {33,000 kgf}) when operated under the fluctuating conditions shown in **Table 4.6**.

Table 4.6

Condition No. i	Operating time ϕ_i %	Radial load F_{ri} kN { kgf }	Axial load F_{ai} kN { kgf }	Revolution n_i min^{-1}
1	5	10 { 1020 }	2 { 204 }	1200
2	10	12 { 1220 }	4 { 408 }	1000
3	60	20 { 2040 }	6 { 612 }	800
4	15	25 { 2550 }	7 { 714 }	600
5	10	30 { 3060 }	10 { 1020 }	400

The equivalent radial load, P_r, for each operating condition is found by using formula (4.17) and shown in **Table 4.7**. Because all the values for F_{ri} and F_{ai} from the bearing tables are greater than $F_a / F_r > e$ = 0.18, X = 0.67, Y_2 = 5.50.

$$P_{ri} = XF_{ri} + Y_2 F_{ai} = 0.67F_{ri} + 5.50F_{ai}$$

From formula (4.12) the mean load, F_m, is:

$$F_m = \left[\frac{\Sigma\left(P_{ri}^{10/3} \cdot n_i \cdot \phi_i\right)}{\Sigma\left(n_i \cdot \phi_i\right)}\right]^{3/10} = 48.1\text{kN} \quad \{4{,}906\text{kgf}\}$$

Table 4.7

Condition No. i	Equivalent radial load. P_{ri} kN { kgf }
1	17.7 { 1805 }
2	30.0 { 3060 }
3	46.4 { 4733 }
4	55.3 { 5641 }
5	75.1 { 7660 }

(Example 6)
Find the threshold values for rating life time and allowable axial load when cylindrical roller bearing NUP312 is used under the following conditions:
Provided that intermittent axial load and oil lubricant.

Radial load F_r=10kN {1,020kgf}

Rotational speed n =2,000 min^{-1}

Radial load is:

$$P_r = F_r = 10\text{kN} \quad \{1{,}020\text{kgf}\}$$

The speed factor of cylindrical roller bearing, f_n, at n= 2,000 min^{-1}, from **Table 3.1**

$$f_n = \left(\frac{33.3}{2{,}000}\right)^{3/10} = 0.293$$

The life factor, f_h, from formula (3.4)

$$f_h = 0.293 \times \frac{124}{10} = 3.63$$

Therefore the basic rated life, L_{10h} , from **Table 3.1**

$$L_{10h} = 500 \times 3.63^{10/3} \quad 37{,}000$$

And next, allowable axial load of cylindrical roller bearing is shown in page B-93.

In formula (1) on page B-93, based on NUP312 from Table 4 on page B-93, k = 0.065.

$$d_p = (60+130)/2 = 95\text{mm}, \quad n = 2{,}000 \text{ min}^{-1}$$

Take into consideration that intermittent axial load.

$$d_p \cdot n \times 10^4 = 19 \times 10^4$$

In **Fig. 1** on page B-93, $d_p \cdot n$ = 19×10^4. In the case of intermittent axial load, allowable surface pressure at the lip P_t = 40 MPa.

Therefore the allowable axial load, P_t, following

$$P_z = 0.065 \times 60^2 \times 40 = 9{,}360\text{N} \quad \{954\text{kgf}\}$$

Based on **Table 4** of page B-93, it is within the limits of $F_{a\,max}$ < 0.4×10,000 = 4,000 N. Therefore P_t < 4,000 N {408 kgf}.

II

5. Boundary Dimensions and Bearing Number Codes

5.1 Boundary dimensions

A rolling bearing's major dimensions, known as "boundary dimensions," are shown in **Figs. 5.1** - **5.3**. To facilitate international bearing interchangeability and economical bearing production, bearing boundary dimensions have been standardized by the International Standards Organization (ISO). In Japan, rolling bearing boundary dimensions are regulated by Japanese Industrial Standards (JIS B 1512).

Those boundary dimensions which have been standardized include: bearing bore diameter, outside diameter, width/height, and chamfer dimensions - all important dimensions when considering the compatibility of shafts, bearings, and housings. However, as a general rule, bearing internal construction dimensions are not covered by these dimensions.

For metric series rolling bearings there are 90 standardized bore diameters (d) ranging in size from 0.6mm - 2,500mm.

Outer diameter dimensions (D) for radial bearings with standardized bore diameter dimensions are covered in the "diameter series;" their corresponding width dimensions (B) are covered in the "width series." For thrust bearings there is no width series; instead, these dimensions are covered in the "height series." The combination of all these series is known as the "dimension series." All series numbers are shown in **Table 5.1**.

Although many rolling bearing dimensions are standardized, and have been listed here for purposes of

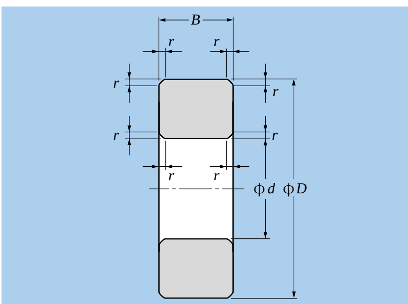


Fig. 5.1 Radial bearings
(excluding tapered roller bearings)

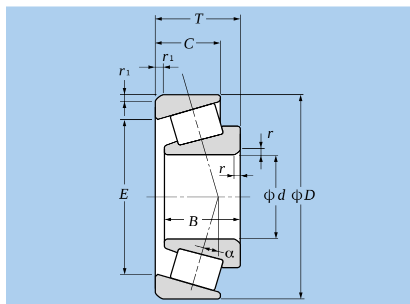


Fig. 5.2 Tapered roller bearings

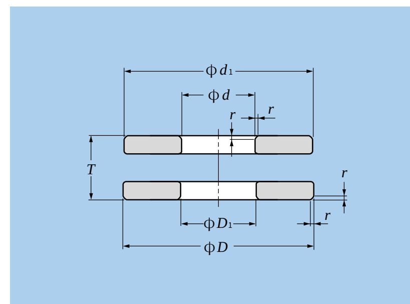


Fig. 5.3 Single direction thrust bearings

Table 5.1 Dimension series numbers

		Dimension series			
		Diameter series (outer diameter dimensions)	Width series (width dimensions)	Height series (height dimensions)	Reference diagram
Radial bearings (excluding tapered roller bearings)	number	7, 8, 9, 0, 1, 2, 3, 4	8, 0, 1, 2, 3, 4, 5, 6	——	**Diagram 5.4**
	dimensions	small ◄──► large	small ◄──► large		
Tapered roller bearings	number	9, 0, 1, 2, 3	0, 1, 2, 3	——	**Diagram 5.5**
	dimensions	small ◄──► large	small ◄──► large		
Thrust bearings	number	0, 1, 2, 3, 4	——	7, 9, 1, 2	**Diagram 5.6**
	dimensions	small ◄──► large		small ◄► large	

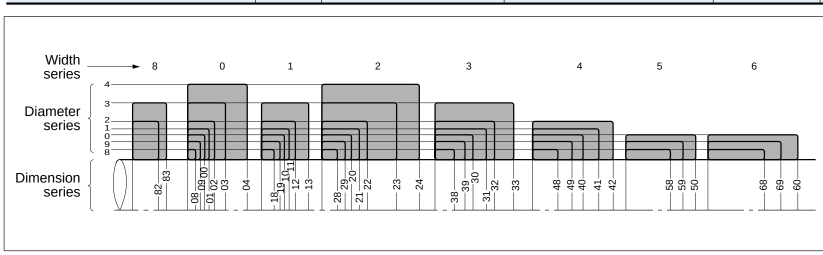


Fig. 5.4 Dimension series for radial bearings
(excluding tapered roller bearings; diameter series 7 has been omitted)

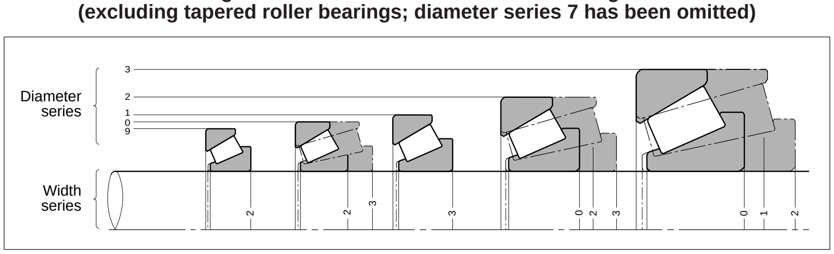


Fig. 5.5 Dimension series for tapered roller bearings

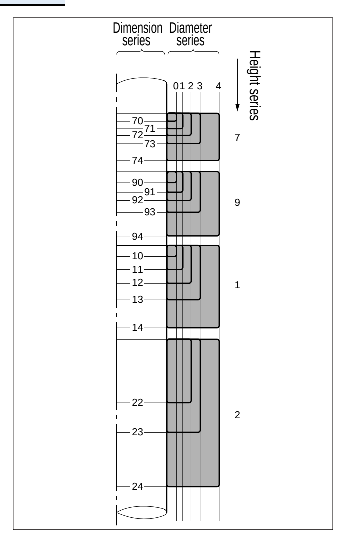


Fig. 5.6 Dimension series for thrust bearings
(excluding diameter series 5)

future standardization, there are many standard bearing dimensions which are not presently manufactured.

Boundary dimensions for radial bearings (excluding tapered roller bearings) are shown in the attached tables.

5.2 Bearing numbers

Rolling bearing part numbers indicate bearing type, dimensions, tolerances, internal construction, and other related specifications. Bearing numbers are comprised of a "basic number" followed by "supplementary codes." The makeup and order of bearing numbers is shown in **Table 5.2**.

The basic number indicates general information about a bearing, such as its fundamental type, boundary dimensions, series number, bore diameter code and contact angle. The supplementary codes derive from prefixes and suffixes which indicate a bearing's tolerances, internal clearances, and related specifications.

(Bearing number examples)

6205ZZC3／2A

- 6 — **Deep groove ball bearing**
- 2 — Diameter series 2
- 05 — Nominal bore diameter 25mm
- ZZ — Shielded (both)
- C3 — Radial internal clearance C3
- ／2A — Shell Alvania 2 grease

23034BD1

- 2 — **Spherical roller bearing**
- 3 — Width series 3
- 0 — Dimension series 0
- 34 — Nominal bore diameter 170mm
- B — Type B
- D1 — Lubrication hole/lubrication groove (when outer diameter is less than 320mm)

7012CDB／GMP4

- 7 — **Angular contact ball bearing**
- 0 — Dimension series 0
- 12 — Nominal bore diameter 60mm
- C — Contact angle 15°
- DB — Back-to-back duplex arrangement
- ／GM — Medium preload
- P4 — Tolerances JIS Class 4

240／750BK30

- 2 — **Spherical roller bearing**
- 4 — Width series 4
- 0 — Dimension series 0
- ／750 — Bore diameter 750mm
- B — Type B
- K30 — Bore diameter: tapered inner ring bore, standard taper ratio 1:30

NU320G1C3

- NU — **Cylindrical roller bearing NU type**
- 3 — Dimension series 3
- 20 — Nominal bore diameter 100mm
- G1 — High strength machined brass rivetless cage with square holes
- C3 — Radial internal clearance C3

51120L1P5

- 5 — **Thrust ball bearing**
- 1 — Height series 1
- 1 — Diameter series 1
- 20 — Nominal bore diameter 100mm
- L1 — High strength, machined brass cage
- P5 — Tolerances JIS Class 5

4T－30208

- 4T－ — Spec. 4T
- 3 — **Tapered roller bearing**
- 0 — Width series 0
- 2 — Diameter series 2
- 08 — Nominal bore diameter 40mm

Table 5.2 Bearing number composition and arrangement

Supplementary prefix code — Special application/material/heat treatment code

Code	Description
4T:	4T tapered roller bearings
ET:	ET tapered roller bearings
ETA:	ET+special heat treatment
E:	Bearing using case hardened steel
EA:	Bearing made of nitride-treated case hardened steel
TA:	Bearing made of nitride-treated bearing steel (SUJ3)
TM:	Bearing made of special heat-treated bearing steel (SUJ3)
F:	Stainless steel bearings
N:	High speed steel bearings
M:	Plated bearings
5S:	Ceramic rolling element bearings
HL:	HL roller bearings
ECO:	ECO-Top tapered roller bearings
LH:	Bearing made of bearing steel that provides long life at high temperatures (STJ2), which is treated to stabilize dimensions at temperatures up to 250°C
TS3:	Dimension stabilized bearing for high temperature use (to 200°C)
TS4:	Dimension stabilized bearing for high temperature use (to 250°C)

Basic number — Bearing series

Bearing series code	Dimension series code: Width/height series❶	Dimension series code: Diameter series
Deep groove ball bearings (type code 6)		
68	(1)	8
69	(1)	9
60	(1)	0
62	(0)	2
63	(0)	3
Angular contact ball bearings (type code 7)		
78	(1)	8
79	(1)	9
70	(1)	0
72	(0)	2
73	(0)	3
Self-aligning ball bearings (type code 1,2)		
12	(0)	2
13	(0)	3
22	(2)	2
23	(2)	3
Cylindrical roller bearings (type code NU, N, NF, NNU, NN, etc.)		
NU10	1	0
NU2	(0)	2
NU22	2	2
NU3	(0)	3
NU23	2	3
NU4	(0)	4
NNU49	4	9
NN30	3	0
Tapered roller bearings (type code 3)		
329X	2	9
320X	2	0
302	0	2
322	2	2
303	0	3
303D	0	3
313X	1	3
323	2	3
Spherical roller bearings (type code 2)		
239	3	9
230	3	0
240	4	0
231	3	1
241	4	1
222	2	2
232	3	2
213	1	3
223	2	3
Single direction thrust ball bearings (type code 5)		
511	1	1
512	1	2
513	1	3
514	1	4
Cylindrical roller thrust bearings (type code 8)		
811	1	1
812	1	2
893	9	3
Spherical thrust roller bearings (type code 2)		
292	9	2
293	9	3
294	9	4

Basic number — Bore diameter code

Code	bore diameter mm
/0.6	0.6
/1.5	1.5
/2.5	2.5
1	1
⋮	⋮
9	9
00	10
01	12
02	15
03	17
/22	22
/28	28
/32	32
04	20
05	25
06	30
⋮	⋮
88	440
92	460
96	480
/500	500
/530	530
/560	560
	⋮
/2,360	2,360
/2,500	2,500

Basic number — Contact angle code

Code❶	Contact angle
Angular contact ball bearings	
(A)	Standard contact angle 30°
B	Standard contact angle 40°
C	Standard contact angle 15°
Tapered roller bearings	
(B)	Contact angle over 10° to/including 17°
C	Contact angle over 17° to/including 24°
D	Contact angle over 24° to/including 32°

❶ Codes in ( ) are not shown in nominal numbers.

Note: Please consult NTN Engineering concerning bearing series codes, and supplementary prefix/suffix codes not listed in the above table.

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Supplementary suffix codes							
Internal modifications code	cage code	Seal / Shield code	External configuration code	Duplex arrangement code	Internal clearance /preload code[1]	Tolerance code	Lubrication code
U: Internationally interchangeable tapered roller bearings	L1: High strength, machined brass cage	LLB: Synthetic rubber seal (non-contact type)	K: Tapered inner ring bore, standard taper ratio 1:12	DB: Back-to-back arrangement	C2: Internal clearance less than normal	P6: JIS Class 6	/2A: Shell Alvania 2 grease
R: Non-internationally interchangeable tapered roller bearings	F1: Machined carbon steel cage	LLU: Synthetic rubber seal (contact type)	K30: Tapered inner ring bore, standard taper ratio 1:30	DF: Face-to-face arrangement	(CN): Normal clearance	P5: JIS Class 5	/3A: Shell Alvania 3 grease
ST: Low torque tapered roller bearings	G1: High strength machined brass rivetless cage with square holes,	LLH: Synthetic rubber seal (low-torque type)	N: With snap ring groove	DT: Tandem arrangement	C3: Internal clearance greater than normal	P4: JIS Class 4	/8A: Shell Alvania EP2 grease
HT: High axial load use cylindrical roller bearings	G2: Pin type cage	ZZ: Steel shield	NR: With snap ring	D2: Two matched, paired bearings	C4: Internal clearance greater than C3	P2: JIS Class 2	/5K: MULTEMP SRL
	J: Pressed steel cage		D: With oil hole	G: Flush ground	C5: Internal clearance greater than C4	2: ABMA Class 2	/LX11: Barierta JFE552
	T2: Plastic mold cage		D1: Lubrication hole/lubrication groove	$+\alpha$: Spacer (α = spacer's standard width dimensions)	CM: Radial internal clearance for electric motor use	3: ABMA Class 3	/LP03: Thermosetting grease (grease for poly-lube bearings)
					/GL: Light preload	0: ABMA Class 0	
					/GN: Normal preload	00: ABMA Class 00	
					/GM: Medium preload		
					/GH: Heavy preload		

6. Bearing Tolerances

6.1 Dimensional accuracy and running accuracy

Bearing “tolerances” or dimensional accuracy and running accuracy, are regulated by ISO and JIS B 1514 standards (rolling bearing tolerances). For dimensional accuracy, these standards prescribe the tolerances necessary when installing bearings on shafts or in housings. Running accuracy is defined as the allowable limits for bearing runout during operation.

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Dimensional accuracy

Dimensional accuracy constitutes the acceptable values for bore diameter, outer diameter, assembled bearing width, and bore diameter uniformity as seen in chamfer dimensions, allowable inner ring tapered bore deviation and shape error. Also included are, average bore diameter variation, outer diameter variation, average outer diameter unevenness, as well as raceway width and height variation (for thrust bearings).

Running accuracy

Running accuracy constitutes the acceptable values for inner and outer ring radial runout and axial runout, inner ring side runout, and outer ring outer diameter runout.

Allowable rolling bearing tolerances have been established according to precision classes. Bearing precision is stipulated as JIS class 6, class 5, class 4, or class 2, with precision rising from ordinary precision indicated by class 0.

Table 6.1 indicates which standards and precision classes are applicable to the major bearing types. **Table 6.2** shows a relative comparison between JIS B 1514 precision class standards and other standards. For greater detail on allowable limitations and values, refer to **Tables 6.3** - **6.9**. Allowable values for chamfer dimensions are shown in **Table 6.10**, and allowable limitations and values for radial bearing inner ring tapered bores are shown in **Table 6.11**.

Table 6.1 Bearing types and applicable tolerance

Bearing type		Applicable standard	Tolerance class					Tolerance table
Deep groove ball bearings		JIS B 1514 (ISO492)	class 0	class 6	class 5	class 4	class 2	Table 6.3
Angular contact ball bearings			class 0	class 6	class 5	class 4	class 2	
Self-aligning ball bearings			class 0	—	—	—	—	
Cylindrical roller bearigns			class 0	class 6	class 5	class 4	class 2	
Needle roller bearings			class 0	class 6	class 5	class 4	—	
Spherical roller bearings			class 0	—	—	—	—	
Tapered roller bearings	metric	JIS B 1514	class 0,6X	class 6	class 5	class 5	—	**Table 6.4**
	Inch	ANSI/ABMA Std.19	class 4	class 2	class 3	class 0	class 00	**Table 6.5**
	J series	ANSI/ABMA Std.19.1	class K	class N	class C	class B	class A	**Table 6.6**
Thrust ball bearings		JIS B 1514 (ISO199)	class 0	class 6	class 5	class 4	—	**Table 6.7**
Spherical roller thrust bearings			class 0	—	—	—	—	**Table 6.8**
Double direction angular contact thrust ball bearings		NTN standard	—	—	class 5	class 4	—	**Table 6.9**

Table 6.2 Comparison of tolerance classifications of national standards

Standard	Applicable standerd	Tolerance Class					Bearing Types
Japanese industrial standard (JIS)	JIS B 1514	Class 0,6X	Class 6	Class 5	Class 4	Class 2	All type
International Organization for Standardization (ISO)	ISO 492	Normal class Class 6X	Class 6	Class 5	Class 4	Class 2	Radial bearings
	ISO 199	Normal Class	Class 6	Class 5	Class 4	—	Thrust ball bearings
	ISO 578	Class 4	—	Class 3	Class 0	Class 00	Tapered roller bearings (Inch series)
	ISO 1224	—	—	Class 5A	Class 4A	—	Precision instrument bearings
Deutsches Institut fur Normung(DIN)	DIN 620	P0	P6	P5	P4	P2	All type
American National Standards Institute (ANSI)	ANSI/ABMA Std.20 ❶	ABEC-1 RBEC-1	ABEC-3 RBEC-3	ABEC-5 RBEC-5	ABEC-7	ABEC-9	Radial bearings (Except tapered roller bearings)
American Bearing Manufacturer's Association (ABMA)	ANSI/ABMA Std.19.1	Class K	Class N	Class C	Class B	Class A	Tapered roller bearings (Metric series)
	ANSI/ABMA Std.19	Class 4	Class 2	Class 3	Class 0	Class 00	Tapered roller bearings (Inch series)

❶ "ABEC" is applied for ball bearings and "RBEC" for roller bearings.

Notes 1: JIS B 1514, ISO 492 and 199, and DIN 620 have the same specification level.

2: The tolerance and allowance of JIS B 1514 are a little different from those of ABMA standards.

Table 6.3 Tolerance of radial bearings (Except tapered roller bearings)
Table 6.3 (1) Inner rings

Nominal bore diameter *d* mm		Dimensional tolerance of mean bore diameter within plane Δ_{dmp}										Bore diameter variation V_{dp}														
												diameter series 9					diameter series 0.1					diameter series 2.3.4				
		class 0		class 6		class 5		class 4 [1]		class 2 [1]		class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	max					max					max				
0.6 [4]	2.5	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
2.5	10	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
10	18	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
18	30	0	-10	0	-8	0	-6	0	-5	0	-2.5	13	10	6	5	2.5	10	8	5	4	2.5	8	6	5	4	2.5
30	50	0	-12	0	-10	0	-8	0	-6	0	-2.5	15	13	8	6	2.5	12	10	6	5	2.5	9	8	6	5	2.5
50	80	0	-15	0	-12	0	-9	0	-7	0	-4	19	15	9	7	4	19	15	7	5	4	11	9	7	5	4
80	120	0	-20	0	-15	0	-10	0	-8	0	-5	25	19	10	8	5	25	19	8	6	5	15	11	8	6	5
120	150	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
150	180	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
180	250	0	-30	0	-22	0	-15	0	-12	0	-8	38	28	15	12	8	38	28	12	9	8	23	17	12	9	8
250	315	0	-35	0	-25	0	-18	—	—	—	—	44	31	18	—	—	44	31	14	—	—	26	19	14	—	—
315	400	0	-40	0	-30	0	-23	—	—	—	—	50	38	23	—	—	50	38	18	—	—	30	23	18	—	—
400	500	0	-45	0	-35	—	—	—	—	—	—	56	44	—	—	—	56	44	—	—	—	34	26	—	—	—
500	630	0	-50	0	-40	—	—	—	—	—	—	63	50	—	—	—	63	50	—	—	—	38	30	—	—	—
630	800	0	-75	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
800	1 000	0	-100	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 000	1 250	0	-125	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 250	1 600	0	-160	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 600	2 000	0	-200	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

❶ The dimensional difference Δ_{ds} of bore diameter to applied for class 4 and 2 is the same as the tolerance of dimentional difference Δ_{dmp} of average bore diameter. However, the dimensional difference is applied to diameter series 0, 1, 2, 3 and 4 against Class 4, and to all the diameter series against Class 2.

Table 6.3 (2) Outer rings

Nominal outside diameter *D* mm		Dimensional tolerance of mean outside diameter within plane Δ_{Dmp}										Outside diameter variation [6] V_{Dp} open type														
												diameter series 9					diameter series 0.1					diameter series 2.3.4				
		class 0		class 6		class 5		class 4 [5]		class 2 [5]		class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2
over	incl.	high	low	high	low	high	low	high	low	high	low	max					max					max				
2.5 [8]	6	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
6	18	0	-8	0	-7	0	-5	0	-4	0	-2.5	10	9	5	4	2.5	8	7	4	3	2.5	6	5	4	3	2.5
18	30	0	-9	0	-8	0	-6	0	-5	0	-4	12	10	6	5	4	9	8	5	4	4	7	6	5	4	4
30	50	0	-11	0	-9	0	-7	0	-6	0	-4	14	11	7	6	4	11	9	5	5	4	8	7	5	5	4
50	80	0	-13	0	-11	0	-9	0	-7	0	-4	16	14	9	7	4	13	11	7	5	4	10	8	7	5	4
80	120	0	-15	0	-13	0	-10	0	-8	0	-5	19	16	10	8	5	19	16	8	6	5	11	10	8	6	5
120	150	0	-18	0	-15	0	-11	0	-9	0	-5	23	19	11	9	5	23	19	8	7	5	14	11	8	7	5
150	180	0	-25	0	-18	0	-13	0	-10	0	-7	31	23	13	10	7	31	23	10	8	7	19	14	10	8	7
180	250	0	-30	0	-20	0	-15	0	-11	0	-8	38	25	15	11	8	38	25	11	8	8	23	15	11	8	8
250	315	0	-35	0	-25	0	-18	0	-13	0	-8	44	31	18	13	8	44	31	14	10	8	26	19	14	10	8
315	400	0	-40	0	-28	0	-20	0	-15	0	-10	50	35	20	15	10	50	35	15	11	10	30	21	15	11	10
400	500	0	-45	0	-33	0	-23	—	—	—	—	56	41	23	—	—	56	41	17	—	—	34	25	17	—	—
500	630	0	-50	0	-38	0	-28	—	—	—	—	63	48	28	—	—	63	48	21	—	—	38	29	21	—	—
630	800	0	-75	0	-45	0	-35	—	—	—	—	94	56	35	—	—	94	56	26	—	—	55	34	26	—	—
800	1 000	0	-100	0	-60	—	—	—	—	—	—	125	75	—	—	—	125	75	—	—	—	75	45	—	—	—
1 000	1 250	0	-125	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 250	1 600	0	-160	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1 600	2 000	0	-200	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
2 000	2 500	0	-250	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

❺ The dimensional difference Δ_{Ds} of outer diameter to be applied for classes 4 and 2 is the same as the tolerance of dimensional difference Δ_{Dmp} of average outer diameter. However, the dimensional difference is applied to diameter series 0, 1, 2, 3 and 4 against Class 4, and also to all the diameter series against Class 2.

II

Unit μm

Mean bore diameter variation V_{dmp}					Inner ring radial runout K_{ia}					Side runout with bore S_d			Inner ring axial runout S_{ia}[2]			Inner ring width deviation Δ_{Bs}										Inner ring width variation V_{Bs}				
																normal						modified[3]								
																class 0,6		class 5,4		class 2		class 0,6		class 5,4						
class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2	high	low	high	low	high	low	high	low	high	low	class 0	class 6	class 5	class 4	class 2
max					max					max			max													max				
6	5	3	2	1.5	10	5	4	2.5	1.5	7	3	1.5	7	3	1.5	0	-40	0	-40	0	-40	—	—	0	-250	12	12	5	2.5	1.5
6	5	3	2	1.5	10	6	4	2.5	1.5	7	3	1.5	7	3	1.5	0	-120	0	-40	0	-40	0	-250	0	-250	15	15	5	2.5	1.5
6	5	3	2	1.5	10	7	4	2.5	1.5	7	3	1.5	7	3	1.5	0	-120	0	-80	0	-80	0	-250	0	-250	20	20	5	2.5	1.5
8	6	3	2.5	1.5	13	8	4	3	2.5	8	4	1.5	8	4	2.5	0	-120	0	-120	0	-120	0	-250	0	-250	20	20	5	2.5	1.5
9	8	4	3	1.5	15	10	5	4	2.5	8	4	1.5	8	4	2.5	0	-120	0	-120	0	-120	0	-250	0	-250	20	20	5	3	1.5
11	9	5	3.5	2	20	10	5	4	2.5	8	5	1.5	8	5	2.5	0	-150	0	-150	0	-150	0	-380	0	-250	25	25	6	4	1.5
15	11	5	4	2.5	25	13	6	5	2.5	9	5	2.5	9	5	2.5	0	-200	0	-200	0	-200	0	-380	0	-380	25	25	7	4	2.5
19	14	7	5	3.5	30	18	8	6	2.5	10	6	2.5	10	7	2.5	0	-250	0	-250	0	-250	0	-500	0	-380	30	30	8	5	2.5
19	14	7	5	3.5	30	18	8	6	5	10	6	4	10	7	5	0	-250	0	-250	0	-250	0	-500	0	-380	30	30	8	5	4
23	17	8	6	4	40	20	10	8	5	11	7	5	13	8	5	0	-300	0	-300	0	-300	0	-500	0	-500	30	30	10	6	5
26	19	9	—	—	50	25	13	—	—	13	—	—	15	—	—	0	-350	0	—	—	—	0	-500	0	—	35	35	13	—	—
30	23	12	—	—	60	30	15	—	—	15	—	—	20	—	—	0	-400	0	—	—	—	0	-630	0	—	40	40	15	—	—
34	26	—	—	—	65	35	—	—	—	—	—	—	—	—	—	0	-450	—	—	—	—	—	—	—	—	50	45	—	—	—
38	30	—	—	—	70	40	—	—	—	—	—	—	—	—	—	0	-500	—	—	—	—	—	—	—	—	60	50	—	—	—
55	—	—	—	—	80	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	70	—	—	—	—
75	—	—	—	—	90	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	80	—	—	—	—
94	—	—	—	—	100	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	100	—	—	—	—
120	—	—	—	—	120	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	120	—	—	—	—
150	—	—	—	—	140	—	—	—	—	—	—	—	—	—	—	0	—	—	—	—	—	—	—	—	—	140	—	—	—	—

❷ Applies to ball bearings such as deep groove ball bearings and angular ball bearings.
❸ To be applied for individual raceway rings manufactured for combined bearing use.
❹ Nominal bore diameter of bearings of 0.6 mm is included in this dimensional division.

Unit μm

Outside diameter variation V_{DP}[6] Sealed/shield bearings diameter series		Mean bore diameter variation V_{Dmp}					Outer ring radial runout K_{ea}					Outside surface inclination S_D			Outside ring axial runout S_{ea}[7]			Outer ring width deviation Δ_{Cs}	Outer ring width variation V_{Cs}			
2,3,4 class 0	0,1,2,3,4 class 6	class 0	class 6	class 5	class 4	class 2	class 0	class 6	class 5	class 4	class 2	class 5	class 4	class 2	class 5	class 4	class 2		class 0,6	class 5	class 4	class 2
max		max					max					max			max			all type	max			
10	9	6	5	3	2	1.5	15	8	5	3	1.5	8	4	1.5	8	5	1.5	Depends on tolerance of Δ_{Bs} in relation to d of same bearing	Depends on tolerance of Δ_{Bs} in relation to d of same bearing	5	2.5	1.5
10	9	6	5	3	2	1.5	15	8	5	3	1.5	8	4	1.5	8	5	1.5			5	2.5	1.5
12	10	7	6	3	2.5	2	15	9	6	4	2.5	8	4	1.5	8	5	2.5			5	2.5	1.5
16	13	8	7	4	3	2	20	10	7	5	2.5	8	4	1.5	8	5	2.5			5	2.5	1.5
20	16	10	8	5	3.5	2	25	13	8	5	4	8	4	1.5	10	5	4			6	3	1.5
26	20	11	10	5	4	2.5	35	18	10	6	5	9	5	2.5	11	6	5			8	4	2.5
30	25	14	11	6	5	2.5	40	20	11	7	5	10	5	2.5	13	7	5			8	5	2.5
38	30	19	14	7	5	3.5	45	23	13	8	5	10	5	2.5	14	8	5			8	5	2.5
—	—	23	15	8	6	4	50	25	15	10	7	11	7	4	15	10	7			10	7	4
—	—	26	19	9	7	4	60	30	18	11	7	13	8	5	18	10	7			11	7	5
—	—	30	21	10	8	5	70	35	20	13	8	13	10	7	20	13	8			13	8	7
—	—	34	25	12	—	—	80	40	23	—	—	15	—	—	23	—	—			15	—	—
—	—	38	29	14	—	—	100	50	25	—	—	18	—	—	25	—	—			18	—	—
—	—	55	34	18	—	—	120	60	30	—	—	20	—	—	30	—	—			20	—	—
—	—	75	45	—	—	—	140	75	—	—	—	—	—	—	—	—	—			—	—	—
—	—	—	—	—	—	—	160	—	—	—	—	—	—	—	—	—	—			—	—	—
—	—	—	—	—	—	—	190	—	—	—	—	—	—	—	—	—	—			—	—	—
—	—	—	—	—	—	—	220	—	—	—	—	—	—	—	—	—	—			—	—	—
—	—	—	—	—	—	—	250	—	—	—	—	—	—	—	—	—	—			—	—	—

❻ To be applied in case snap rings are not installed on the bearings.
❼ Applies to ball bearings such as deep groove ball bearings and angular ball bearings.
❽ Nominal outer diameter of bearings of 2.5 mm is included in this dimensional division.

Table 6.4 Tolerance of tapered roller bearings (Metric series)

Table 6.4 (1) Inner rings

Nominal bore diameter d mm		Dimensional tolerance of mean bore diameter within plane Δ_{dmp}						Bore diameter variation V_{dp}				Mean bore diameter variation V_{dmp}				Inner ring radial runout K_{ia}				Side runout with bore S_d	
		class 0,6X		class 5,6		class 4 ❶		class 0,6X	class 6	class 5	class 4	class 0,6X	class 6	class 5	class 4	class 0,6X	class 6	class 5	class 4	class 5	class 4
over	incl.	high	low	high	low	high	low	max				max				max				max	
10	18	0	-12	0	-7	0	-5	12	7	5	4	9	5	5	4	15	7	5	3	7	3
18	30	0	-12	0	-8	0	-6	12	8	6	5	9	6	5	4	18	8	5	3	8	4
30	50	0	-12	0	-10	0	-8	12	10	8	6	9	8	5	5	20	10	6	4	8	4
50	80	0	-15	0	-12	0	-9	15	12	9	7	11	9	6	5	25	10	7	4	8	5
80	120	0	-20	0	-15	0	-10	20	15	11	8	15	11	8	5	30	13	8	5	9	5
120	180	0	-25	0	-18	0	-13	25	18	14	10	19	14	9	7	35	18	11	6	10	6
180	250	0	-30	0	-22	0	-15	30	22	17	11	23	16	11	8	50	20	13	8	11	7
250	315	0	-35	—	—	—	—	35	—	—	—	26	—	—	—	60	—	—	—	—	—
315	400	0	-40	—	—	—	—	40	—	—	—	30	—	—	—	70	—	—	—	—	—
400	500	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
500	630	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
630	800	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
800	1,000	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—

❶ The dimensional difference Δ_{ds} of bore diameter to be applied for class 4 is the same as the tolerance of dimensional difference Δ_{dmp} of average bore diameter.

Table 6.4 (2) Outer rings

Nominal outside diameter D mm		Dimensional tolerance of mean outside diameter within plane Δ_{Dmp}						Outside diameter variation V_{Dp}				Mean bore diameter variation V_{Dmp}				Outer ring radial runout K_{ea}				Outside surface inclination S_D ❷	
		class 0,6X		class 5,6		class 4 ❸		class 0,6X	class 6	class 5	class 4	class 0,6X	class 6	class 5	class 4	class 0,6X	class 6	class 5	class 4	class 5	class 4
over	incl.	high	low	high	low	high	low	max				max				max				max	
18	30	0	-12	0	-8	0	-6	12	8	6	5	9	6	5	4	18	9	6	4	8	4
30	50	0	-14	0	-9	0	-7	14	9	7	5	11	7	5	5	20	10	7	5	8	4
50	80	0	-16	0	-11	0	-9	16	11	8	7	12	8	6	5	25	13	8	5	8	4
80	120	0	-18	0	-13	0	-10	18	13	10	8	14	10	7	5	35	18	10	6	9	5
120	150	0	-20	0	-15	0	-11	20	15	11	8	15	11	8	6	40	20	11	7	10	5
150	180	0	-25	0	-18	0	-13	25	18	14	10	19	14	9	7	45	23	13	8	10	5
180	250	0	-30	0	-20	0	-15	30	20	15	11	23	15	10	8	50	25	15	10	11	7
250	315	0	-35	0	-25	0	-18	35	25	19	14	26	19	13	9	60	30	18	11	13	8
315	400	0	-40	0	-28	0	-20	40	28	22	15	30	21	14	10	70	35	20	13	13	10
400	500	0	-45	—	—	—	—	45	—	—	—	34	—	—	—	80	—	—	—	—	—
500	630	0	-50	—	—	—	—	50	—	—	—	38	—	—	—	100	—	—	—	—	—

❷ Does not apply to bearings with flange.
❸ The dimensional difference Δ_{Ds} of outside diameter to be applied for class 4 is the same as the tolerance of dimensional difference Δ_{Dmp} of average outside diameter.

Unit μm

Inner ring axial runout S_{ia}	Inner ring width deviation Δ_{Bs}						Assembly width deviation of single-row tapered roller bearings Δ_{Ts}						Combination width deviation of double row bearings Δ_{B1s}, Δ_{C1s}		Combination width deviation of 4-row bearings Δ_{B2s}, Δ_{C2s}	
class 4	class 0,6		class 6X		class 4,5		class 0,6		class 6X		class 4,5		class 0,6,5		class 0,6,5	
max	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
3	0	-120	0	-50	0	-200	+200	0	+100	0	+200	-200	—	—	—	—
4	0	-120	0	-50	0	-200	+200	0	+100	0	+200	-200	—	—	—	—
4	0	-120	0	-50	0	-240	+200	0	+100	0	+200	-200	+240	-240	—	—
4	0	-150	0	-50	0	-300	+200	0	+100	0	+200	-200	+300	-300	—	—
5	0	-200	0	-50	0	-400	+200	-200	+100	0	+200	-200	+400	-400	+500	-500
7	0	-250	0	-50	0	-500	+350	-250	+150	0	+350	-250	+500	-500	+600	-600
8	0	-300	0	-50	0	-600	+350	-250	+150	0	+350	-250	+600	-600	+750	-750
—	0	-350	0	-50	—	—	+350	-250	+200	0	—	—	+700	-700	+900	-900
—	0	-400	0	-50	—	—	+400	-400	+200	0	—	—	+800	-800	+1 000	-1 000
—	—	—	—	—	—	—	—	—	—	—	—	—	+900	-900	+1 200	-1 200
—	—	—	—	—	—	—	—	—	—	—	—	—	+1 000	-1 000	+1 200	-1 200
—	—	—	—	—	—	—	—	—	—	—	—	—	+1 500	-1 500	+1 500	-1 500
—	—	—	—	—	—	—	—	—	—	—	—	—	+1 500	-1 500	+1 500	-1 500

Unit μm

Outer ring axial runout S_{ea}	Outer ring width deviation Δ_{Cs}			
class 4	class 0,6,5,4		class 6X ❹	
max	sup.	inf.	sup.	inf.
5	Depends on tolerance of Δ_{Bs} in relation to d of same bearing		0	-100
5			0	-100
5			0	-100
6			0	-100
7			0	-100
8			0	-100
10			0	-100
10			0	-100
13			0	-100
—			0	-100
—			0	-100

❹ Applies to bearing where d is greater than 10 mm but is less than or equal to 400 mm.

Table 6.4 (3) Effective width of outer and inner rings with roller Unit μm

Nominal bore diameter d mm		Effective width deviation of roller and inner ring assembly of tapererd roller bearing Δ_{T1s}				Tapered roller bearing outer ring effective width deviation Δ_{T2s}			
		class 0		class 6X		class 0		class 6X	
over	incl.	high	low	high	low	high	low	high	low
10	18	+100	0	+50	0	+100	0	+50	0
18	30	+100	0	+50	0	+100	0	+50	0
30	50	+100	0	+50	0	+100	0	+50	0
50	80	+100	0	+50	0	+100	0	+50	0
80	120	+100	-100	+50	0	+100	-100	+50	0
120	180	+150	-150	+50	0	+200	-100	+100	0
180	250	+150	-150	+50	0	+200	-100	+100	0
250	315	+150	-150	+100	0	+200	-100	+100	0
315	400	+200	-200	+100	0	+200	-200	+100	0

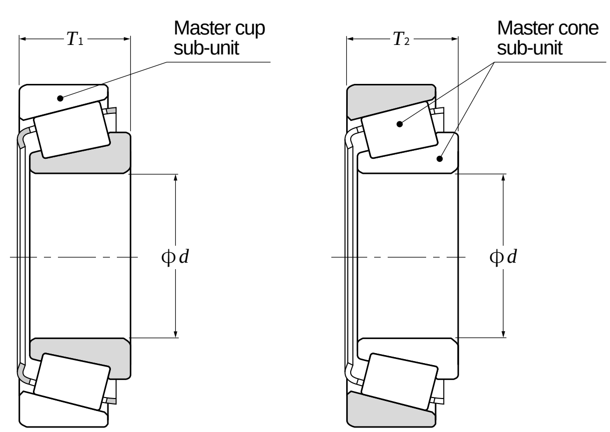


Table 6.5 Tolerance of tapered roller bearings (Inch series)

Table 6.5 (1) Inner rings

Unit μm

Nominal bore diameter d mm (inch)		Single bore diameter deviation Δ_{ds}									
		Class 4		Class 2		Class 3		Class 0		Class 00	
over	incl.	high	low	high	low	high	low	high	low	high	low
—	76.2 (3)	+13	0	+13	0	+13	0	+13	0	+8	0
76.2 (3)	266.7 (10.5)	+25	0	+25	0	+13	0	+13	0	+8	0
266.7 (10.5)	304.8 (12)	+25	0	+25	0	+13	0	+13	0	—	—
304.8 (12)	609.6 (24)	+51	0	+51	0	+25	0	—	—	—	—
609.6 (24)	914.4 (36)	+76	0	—	—	+38	0	—	—	—	—
914.4 (36)	1 219.2 (48)	+102	0	—	—	+51	0	—	—	—	—
1 219.2 (48)	—	+127	0	—	—	+76	0	—	—	—	—

Table 6.5 (2) Outer rings

Unit μm

Nominal outside diameter D mm (inch)		Single outside diameter deviation Δ_{Ds}									
		Class 4		Class 2		Class 3		Class 0		Class 00	
over	incl.	high	low	high	low	high	low	high	low	high	low
—	266.7 (10.5)	+25	0	+25	0	+13	0	+13	0	+8	0
266.7 (10.5)	304.8 (12)	+25	0	+25	0	+13	0	+13	0	—	—
304.8 (12)	609.6 (24)	+51	0	+51	0	+25	0	—	—	—	—
609.6 (24)	914.4 (36)	+76	0	+76	0	+38	0	—	—	—	—
914.4 (36)	1 219.2 (48)	+102	0	—	—	+51	0	—	—	—	—
1 219.2 (48)	—	+127	0	—	—	+76	0	—	—	—	—

Table 6.5 (3) Single-row tapered roller bearing assembly width, combination width of 4-row bearings, effective width of inner ring with rollers, effective width of outer ring

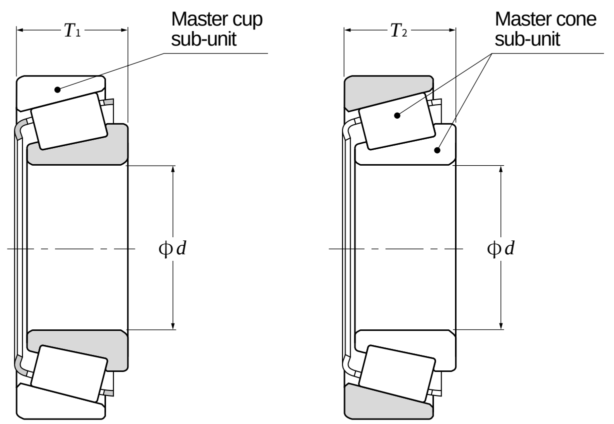
Nominal bore diameter d mm (inch)		Nominal outside diameter D mm (inch)		Overall width deviation of assembled single row tapered roller bearing Δ_{Ts}								Overall width deviation of assembled 4-row tapered roller bearings Δ_{B2s}, Δ_{C2s}	
				Class 4		Class 2		Class 3		Class 0,00		Class 4,2,3,0	
over	incl.	over	incl.	high	low	high	low	high	low	high	low	high	low
—	101.6 (4)	—	508.0 (20)	+203	0	+203	0	+203	-203	+203	-203	+1 524	-1 524
101.6 (4)	304.8 (12)			+356	-254	+203	0	+203	-203	+203	-203	+1 524	-1 524
304.8 (12)	609.6 (24)			+381	-381	+381	-381	+203	-203	—	—	+1 524	-1 524
304.8 (12)	609.6 (36)	508.0 (20)	—	+381	-381	+381	-381	+381	-381	—	—	+1 524	-1 524
609.6 (24)	—			+381	-381	—	—	+381	-381	—	—	+1 524	-1 524

Table 6.5 (4) Radial deflection of inner and outer rings

Unit μm

Nominal outside diameter D mm (inch)		Inner ring radial runout K_{ia} Outer ring radial runout K_{ea}				
		Class 4	Class 2	Class 3	Class 0	Class 00
over	incl.	max				
—	304.8 (14)	51	38	8	4	2
304.8 (14)	609.6 (24)	51	38	18	—	—
609.6 (24)	914.4 (36)	76	51	51	—	—
914.4 (36)	—	76	—	76	—	—

II



Unit μm

Effective width deviation of roller and inner ring assembly of tapered roller bearing Δ_{T1s}						Tapered roller bearing outer ring effective width deviation Δ_{T2s}					
Class 4		Class 2		Class 3		Class 4		Class 2		Class 3	
high	low	high	low	high	low	high	low	high	low	high	low
+102	0	+102	0	+102	-102	+102	0	+102	0	+102	-102
+152	-152	+102	0	+102	-102	+203	-102	+102	0	+102	-102
—	—	+178	-178 ❶	+102	-102 ❶	—	—	+203	-203 ❶	+102	-102 ❶
—	—	—	—	—	—	—	—	—	—	—	—
—	—	—	—	—	—	—	—	—	—	—	—

❶ To be applied for nominal bore diameters d of 406.400 mm (16 inch) or less.

Table 6.6 Tolerance of tapered roller bearings of J series (Metric series)

Table 6.6 (1) Inner rings

Nominal bore diameter d mm		Mean bore diameter deviation Δ_{dmp}								Bore diameter variation V_{dp}				Mean bore diameter variation V_{dmp}			
		Class K		Class N		Class C		Class B		Class K	Class N	Class C	Class B	Class K	Class N	Class C	Class B
over	incl.	high	low	high	low	high	low	high	low	max				max			
10	18	0	-12	0	-12	0	-7	0	-5	12	12	4	3	9	9	5	4
18	30	0	-12	0	-12	0	-8	0	-6	12	12	4	3	9	9	5	4
30	50	0	-12	0	-12	0	-10	0	-8	12	12	4	3	9	9	5	5
50	80	0	-15	0	-15	0	-12	0	-9	15	15	5	3	11	11	5	5
80	120	0	-20	0	-20	0	-15	0	-10	20	20	5	3	15	15	5	5
120	180	0	-25	0	-25	0	-18	0	-13	25	25	5	3	19	19	5	7
180	250	0	-30	0	-30	0	-22	0	-15	30	30	6	4	23	23	5	8

Note: Please consult NTN Engineering for Class A bearings.

Table 6.6 (2) Outer rings

Nominal outside diameter D mm		Mean outside diameter deviation Δ_{Dmp}								Outside diameter variation V_{Dp}				Mean outside diameter variation V_{Dmp}				outer ring axial runout S_{ea}
		Class K		Class N		Class C		Class B		Class K	Class N	Class C	Class B	Class K	Class N	Class C	Class B	Class B
over	incl.	high	low	high	low	high	low	high	low	max				max				max
18	30	0	-12	0	-12	0	-8	0	-6	12	12	4	3	9	9	5	4	3
30	50	0	-14	0	-14	0	-9	0	-7	14	14	4	3	11	11	5	5	3
50	80	0	-16	0	-16	0	-11	0	-9	16	16	4	3	12	12	6	5	4
80	120	0	-18	0	-18	0	-13	0	-10	18	18	5	3	14	14	7	5	4
120	150	0	-20	0	-20	0	-15	0	-11	20	20	5	3	15	15	8	6	4
150	180	0	-25	0	-25	0	-18	0	-13	25	25	5	3	19	19	9	7	5
180	250	0	-30	0	-30	0	-20	0	-15	30	30	6	4	23	23	10	8	6
250	315	0	-35	0	-35	0	-25	0	-18	35	35	8	5	26	26	13	9	6
315	400	0	-40	0	-40	0	-28	0	-20	40	40	10	5	30	30	14	10	6

Note: Please consult NTN Engineering for Class A bearings.

Table 6.6 (3) Effective width of inner and outer rings

Unit μm

Nominal bore diameter d mm		Effective width deviation of roller and inner ring assembly of tapered roller bearing Δ_{T1s}								Tapered roller bearing outer ring effective width deviation Δ_{T2s}							
		Class K		Class N		Class C		Class B		Class K		Class N		Class C		Class B	
over	incl.	high	low	high	low	high	low	high	low	high	low	high	low	high	low	high	low
10	80	+100	0	+50	0	+100	-100	*	*	+100	0	+50	0	+100	-100	*	*
80	120	+100	-100	+50	0	+100	-100	*	*	+100	-100	+50	0	+100	-100	*	*
120	180	+150	-150	+50	0	+100	-100	*	*	+200	-100	+100	0	+100	-150	*	*
180	250	+150	-150	+50	0	+100	-150	*	*	+200	-100	+100	0	+100	-150	*	*

Note 1: "*" mark are to be manufactured only for combined bearings.
2: Please consult NTN Engineering for Class A bearings.

II

Unit μm

Inner ring axial runout S_{ia}	Overall width deviation of assembled tapered roller bearing Δ_{Ts}							
Class B	Class K		Class N		Class C		Class B	
max	sup	inf	sup	inf	sup	inf	sup	inf
3	+200	0	+100	0	+200	-200	+200	-200
4	+200	0	+100	0	+200	-200	+200	-200
4	+200	0	+100	0	+200	-200	+200	-200
4	+200	0	+100	0	+200	-200	+200	-200
5	+200	-200	+100	0	+200	-200	+200	-200
7	+350	-250	+150	0	+350	-250	+200	-250
8	+350	-250	+150	0	+350	-300	+200	-300

Table 6.6 (4) Radial runout of inner and outer rings

Unit μm

Nominal outside diameter D mm		Inner ring radial runout K_{ia} and Outer ring radial runout K_{ea}			
		Class K	Class N	Class C	Class B
over	incl.	max			
18	30	18	18	5	3
30	50	20	20	6	3
50	80	25	25	6	4
80	120	35	35	6	4
120	150	40	40	7	4
150	180	45	45	8	4
180	250	50	50	10	5
250	315	60	60	11	5
315	400	70	70	13	5

Note: Please consult NTN Engineering for Class A bearings.

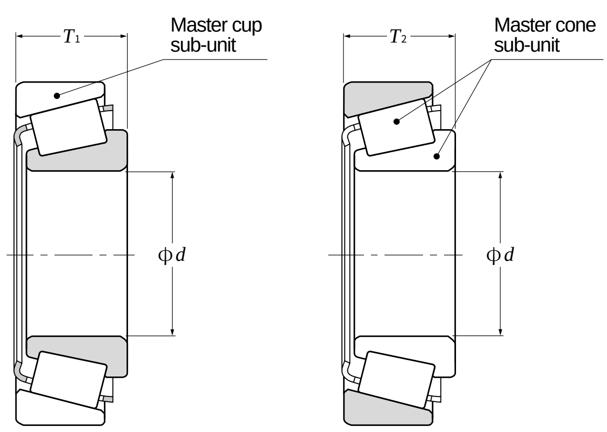


Table 6.7 Tolerance of thrust ball bearings

Table 6.7 (1) Shaft raceway disc

Unit μm

Nominal bore diameter d mm		Mean bore diameter deviation Δ_{dmp}				Bore diameter variation V_{dp}		Raceway thickness variation S_i			
		Class 0,6,5		Class 4		Class 0,6,5	Class 4	Class 0	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	max		max			
—	18	0	-8	0	-7	6	5	10	5	3	2
18	30	0	-10	0	-8	8	6	10	5	3	2
30	50	0	-12	0	-10	9	8	10	6	3	2
50	80	0	-15	0	-12	11	9	10	7	4	3
80	120	0	-20	0	-15	15	11	15	8	4	3
120	180	0	-25	0	-18	19	14	15	9	5	4
180	250	0	-30	0	-22	23	17	20	10	5	4
250	315	0	-35	0	-25	26	19	25	13	7	5
315	400	0	-40	0	-30	30	23	30	15	7	5
400	500	0	-45	0	-35	34	26	30	18	9	6
500	630	0	-50	0	-40	38	30	35	21	11	7

II

Table 6.7 (2) Housing raceway disc

Unit μm

Nominal outside diameter D mm		Mean outside diameter deviation Δ_{Dmp}				Outside diameter variation V_{Dp}		Raceway thickness variation S_e			
		Class 0,6,5		Class 4		Class 0,6,5	Class 4	Class 0	Class 6	Class 5	Class 4
over	incl.	high	low	high	low	max		max			
10	18	0	-11	0	-7	8	5	According to the tolerance of S_1 against "d" of the same bearings			
18	30	0	-13	0	-8	10	6				
30	50	0	-16	0	-9	12	7				
50	80	0	-19	0	-11	14	8				
80	120	0	-22	0	-13	17	10				
120	180	0	-25	0	-15	19	11				
180	250	0	-30	0	-20	23	15				
250	315	0	-35	0	-25	26	19				
315	400	0	-40	0	-28	30	21				
400	500	0	-45	0	-33	34	25				
500	630	0	-50	0	-38	38	29				
630	800	0	-75	0	-45	55	34				

Table 6.7 (3) Bearing height

Unit μm

Nominal bore diameter d mm		Single direction Bearing height ❶ deviation Δ_{Ts}	
over	incl.	high	low
—	30	0	-75
30	50	0	-100
50	80	0	-125
80	120	0	-150
120	180	0	-175
180	250	0	-200
250	315	0	-225
315	400	0	-300
400	500	0	-350
500	630	0	-400

❶ This standard is applied for flat back face bearing of class 0.

II

Table 6.8 Tolerance of spherical thrust roller bearing

Table 6.8 (1) Shaft raceway disc

Unit μm

Nominal bore diameter d mm		Mean bore diameter deviation Δ_{dmp}		Bore diameter variation V_{dp}	Side runout with bore S_d	Bearing height deviation Δ_{Ts}	
over	incl.	high	low	max	max	high	low
50	80	0	-15	11	25	+150	-150
80	120	0	-20	15	25	+200	-200
120	180	0	-25	19	30	+250	-250
180	250	0	-30	23	30	+300	-300
250	315	0	-35	26	35	+350	-350
315	400	0	-40	30	40	+400	-400
400	500	0	-45	34	45	+450	-450

Table 6.8 (2) Housing raceway disc

Unit μm

Nominal outside diameter D mm		Single plane mean outside diameter deviation Δ_{Dmp}	
over	incl.	high	low
120	180	0	-25
180	250	0	-30
250	315	0	-35
315	400	0	-40
400	500	0	-45
500	630	0	-50
630	800	0	-75
800	1,000	0	-100

Table 6.9 Tolerance of double direction type angular contact thrust ball bearings

Table 6.9 (1) Inner rings and bearing height

Unit μm

Nominal bore diameter d mm		Mean bore diameter deviation Δ_{dmp} Bore diameter deviation Δ_{ds}				Side runout with bore S_d		Inner ring axial runout S_{ia}		Inner ring width variation V_{Bs}		Bearing height deviation Δ_{Ts}	
		Class 5		Class 4		Class 5	Class 4	Class 5	Class 4	Class 5	Class 4	Class 5, Class 4	
over	incl.	high	low	high	low	max		max		max		high	low
18	30	0	-6	0	-5	8	4	5	3	5	2.5	0	-300
30	50	0	-8	0	-6	8	4	5	3	5	3	0	-400
50	80	0	-9	0	-7	8	5	6	5	6	4	0	-500
80	120	0	-10	0	-8	9	5	6	5	7	4	0	-600
120	180	0	-13	0	-10	10	6	8	6	8	5	0	-700
180	250	0	-15	0	-12	11	7	8	6	10	6	0	-800
250	315	0	-18	0	-15	13	8	10	8	13	7	0	-900
315	400	0	-23	0	-18	15	9	13	10	15	9	0	-1,000

Table 6.9 (2) Outer rings

Unit μm

Nominal outside diameter D mm		Mean outside diameter deviation Δ_{Dmp} Outside diameter deviation Δ_{Ds}		Outside surface inclination S_D		Outer ring axial runout S_{ea}		Outer ring width variation V_{Cs}	
		Class 5	Class 4	Class 5	Class 4	Class 5	Class 4	Class 5	Class 4
over	incl.	high	low	max		max		max	
30	50	-30	-40	8	4	According to tolerance of S_{ia} against "d" of the same bearings		5	2.5
50	80	-40	-50	8	4			6	3
80	120	-50	-60	9	5			8	4
120	150	-60	-75	10	5			8	5
150	180	-60	-75	10	5			8	5
180	250	-75	-90	11	7			10	7
250	315	-90	-105	13	8			11	7
315	400	-110	-125	13	10			13	8
400	500	-120	-140	15	13			15	10

6.2 Chamfer measurements and tolerance or allowable values of tapered bore

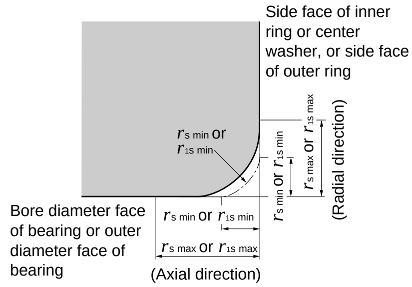


Table 6.10 Allowable critical-value of bearing chamfer

Table 6.10 (1) Radial bearing (Except tapered roller bearing)

Unit mm

$r_{s\,min}$ ❶ or $r_{1s\,min}$	Nominal bore diameter d over	incl.	$r_{s\,max}$ or $r_{1s\,max}$ Radial direction	Axial direction
0.05	—	—	0.1	0.2
0.08	—	—	0.16	0.3
0.1	—	—	0.2	0.4
0.15	—	—	0.3	0.6
0.2	—	—	0.5	0.8
0.3	—	40	0.6	1
	40	—	0.8	1
0.6	—	40	1	2
	40	—	1.3	2
1	—	50	1.5	3
	50	—	1.9	3
1.1	—	120	2	3.5
	120	—	2.5	4
1.5	—	120	2.3	4
	120	—	3	5
2	—	80	3	4.5
	80	220	3.5	5
	220	—	3.8	6
2.1	—	280	4	6.5
	280	—	4.5	7
2.5	—	100	3.8	6
	100	280	4.5	6
	280	—	5	7
3	—	280	5	8
	280	—	5.5	8
4	—	—	6.5	9
5	—	—	8	10
6	—	—	10	13
7.5	—	—	12.5	17
9.5	—	—	15	19
12	—	—	18	24
15	—	—	21	30
19	—	—	25	38

❶ These are the allowable minimum dimensions of the chamfer dimension "r" or "r_1" and are described in the dimensional table.

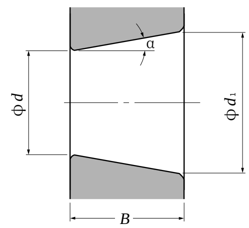
Table 6.10 (2) Tapered roller bearings of metric series

Unit mm

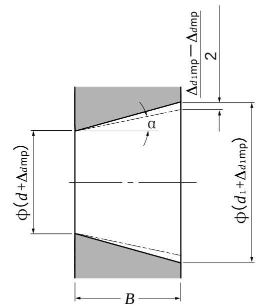
$r_{s\,min}$ ❷ or $r_{1s\,min}$	Nominal bore ❸ diameter of bearing "d" or nominal outside diameter "D" over	incl.	$r_{s\,max}$ or $r_{1s\,max}$ Radial direction	Axial direction
0.3	—	40	0.7	1.4
	40	—	0.9	1.6
0.6	—	40	1.1	1.7
	40	—	1.3	2
1	—	50	1.6	2.5
	50	—	1.9	3
1.5	—	120	2.3	3
	120	250	2.8	3.5
	250	—	3.5	4
2	—	120	2.8	4
	120	250	3.5	4.5
	250	—	4	5
2.5	—	120	3.5	5
	120	250	4	5.5
	250	—	4.5	6
3	—	120	4	5.5
	120	250	4.5	6.5
	250	400	5	7
	400	—	5.5	7.5
4	—	120	5	7
	120	250	5.5	7.5
	250	400	6	8
	400	—	6.5	8.5
5	—	180	6.5	8
	180	—	7.5	9
6	—	180	7.5	10
	180	—	9	11

❷ These are the allowable minimum dimensions of the chamfer dimension "r" or "r_1" and are described in the dimensional table.
❸ Inner rings shall be in accordance with the division of "d" and outer rings with that of "D".

Note: This standard will be applied to the bearings whose dimensional series (refer to the dimensional table) are specified in the standard of ISO 355 or JIS B 1512. For further information concerning bearings outside of these standards or tapered roller bearings using US customary unit, please contact NTN Engineering.



Theoretical tapered bore



Tapered bore having dimensional difference of the average bore diameter within the flat surface

Table 6.10 (3) Thrust bearings

Unit mm

$r_{s\,min}$ or $r_{1\,min}$ ❹	$r_{s\,max}$ or $r_{1s\,max}$ Radial and axial direcition
0.05	0.1
0.08	0.16
0.1	0.2
0.15	0.3
0.2	0.5
0.3	0.8
0.6	1.5
1	2.2
1.1	2.7
1.5	3.5
2	4
2.1	4.5
3	5.5
4	6.5
5	8
6	10
7.5	12.5
9.5	15
12	18
15	21
19	25

❹ These are the allowable minimum dimensions of the chamfer dimension "r" or "r_1" and are described in the dimensional table.

Table 6.11 (1) Tolerance of and tolerance values for tapered bore of radial bearings
Standard taper ratio 1:12 tapared hole (class 0) Unit μm

d mm		Δ_{dmp}		$\Delta_{d1mp} - \Delta_{dmp}$		V_{dp} ❶ ❷
over	incl.	high	low	high	low	max
	10	+ 22	0	+ 15	0	9
10	18	+ 27	0	+ 18	0	11
18	30	+ 33	0	+ 21	0	13
30	50	+ 39	0	+ 25	0	16
50	80	+ 46	0	+ 30	0	19
80	120	+ 54	0	+ 35	0	22
120	180	+ 63	0	+ 40	0	40
180	250	+ 72	0	+ 46	0	46
250	315	+ 81	0	+ 52	0	52
315	400	+ 89	0	+ 57	0	57
400	500	+ 97	0	+ 63	0	63
500	630	+110	0	+ 70	0	70
630	800	+125	0	+ 80	0	—
800	1,000	+140	0	+ 90	0	—
1,000	1,250	+165	0	+105	0	—
1,250	1,600	+195	0	+125	0	—

Table 6.11 (2) Tolerance of and tolerance values for tapered bore of radial bearings
Standard taper ratio 1:30 tapered bore (class 0) Units μm

d mm		Δ_{dmp}		$\Delta_{d1mp} - \Delta_{dmp}$		V_{dp} ❶ ❷
over	incl.	high	low	high	low	max
50	80	+15	0	+30	0	19
80	120	+20	0	+35	0	22
120	180	+25	0	+40	0	40
180	250	+30	0	+46	0	46
250	315	+35	0	+52	0	52
315	400	+40	0	+57	0	57
400	500	+45	0	+63	0	63
500	630	+50	0	+70	0	70

❶ Applies to all radial flat planes of inner ring tapered bore.
❷ Does not apply to diameter series 7 and 8.

Note: Quantifiers

For a standard taper ratio of 1:12 $d_1 = d + \frac{1}{12} B$

For a standard taper ratio of 1:30 $d_1 = d + \frac{1}{30} B$

Δ_{dmp} : Dimensional difference of the average bore diameter within the flat surface at the theoretical small end of the tapered bore.

Δ_{d1mp} : Dimensional difference of the average bore diameter within the flat surface at the theoretical large end of the tapered bore.

V_{dp} : Unevenness of the bore diameter with the flat surface

B : Nominal width of inner ring

α : Half of the tapered bore's nominal taper angle

For a standard taper ratio of 1:12 $\alpha = 2°23'9.4''$

For a standard taper ratio of 1:30 $\alpha = 0°57'7.4''$

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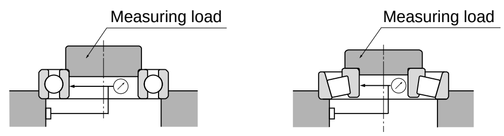
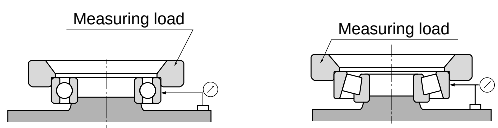
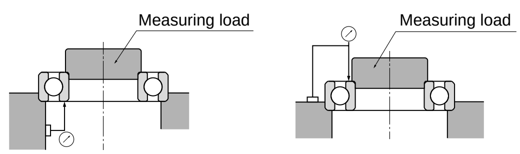
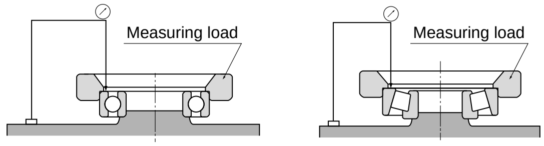
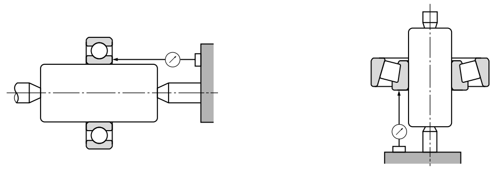
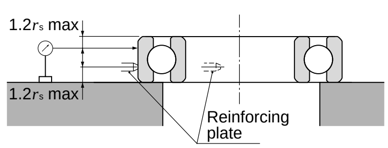
6.3 Bearing tolerance measurement methods

For reference, measurement methods for rolling bearing tolerances are in JIS B 1515.

Table 6.12 shows some of the major methods of measuring rotation tolerances.

Table 6.12 Rotation tolerance measurement methods

Characteristic tolerance	Measurement method	
Inner ring radial runout (K_{ia})		Radial runout of the inner ring is the difference between the maximum and minimum reading of the measuring device when the inner ring is turned one revolution.
Outer ring radial runout (K_{ea})		Radial runout of the outer ring is the difference between the maximum and minimum reading of the measuring device when the outer ring is turned one revolution.
Inner ring axial runout (S_{ia})		Axial runout of the inner ring is the difference between the maximum and minimum reading of the measuring device when the inner ring is turned one revolution.
Outer ring axial runout (S_{ea})		Axial runout of the outer ring is the difference between the maximum and minimum reading of the measuring device when the outer ring is turned one revolution.
Inner ring side runout with bore (S_d)		Inner ring side runout with bore is the difference between the maximum and minimum reading of the measuring device when the inner ring is turned one revolution together with the tapered mandrel.
Outer ring outside surface inclination (S_D)		Outer ring outside surface inclination is the difference between the maximum and minimum reading of the measuring device when the outside ring is turned one revolution along the reinforcing plate.

7 Bearing Fits

7.1 Fitting

For rolling bearings, inner and outer rings are fixed on the shaft or in the housing so that relative movement does not occur between fitting surfaces during operation or under load. This relative movement between the fitting surfaces of the bearing and the shaft or housing can occur in a radial direction, an axial direction, or in the direction of rotation. Types of fitting include tight, transition and loose fitting, which may be selected depending on whether or not there is interference.

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The most effective way to fix the fitting surfaces between a bearing's raceway and shaft or housing is to apply a "**tight fit**." The advantage of this tight fit for thin walled bearings is that it provides uniform load support over the entire ring circumference without any loss of load carrying capacity. However, with a tight fit, ease of installation and disassembly is lost; and when using a non-separable bearing as the floating-side bearing, axial displacement is not possible. For this reason, a tight fit cannot be recommended in all cases.

7.2 The necessity of a proper fit

In some cases, improper fit may lead to damage and shorten bearing life, therefore it is necessary to make a careful investigation in selecting a proper fit. Some of the bearing failure caused by improper fit are listed below.

- Raceway cracking, early flaking and displacement of raceway
- Raceway and shaft or housing abrasion caused by creeping and fretting corrosion
- Seizing caused by negative internal clearances
- Increased noise and deteriorated rotational accuracy due to raceway groove deformation

Please refer to insert pages A-96 〜 A-99 for information concerning diagnosis of these conditions.

7.3 Fit selection

Selection of a proper fit is dependent upon thorough analysis of bearing operating conditions, including consideration of:

- Shaft and housing material, wall thickness, finished surface accuracy, etc.
- Machinery operating conditions (nature and magnitude of load, rotational speed, temperature, etc.)

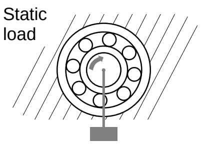
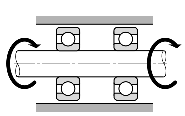
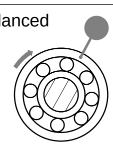
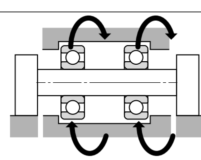
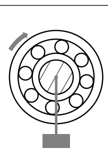
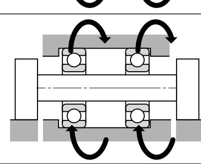
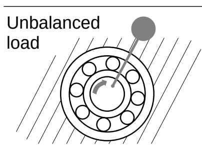
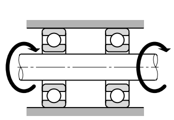
7.3.1 "Tight fit" or "Loose fit"

(1) For raceways under rotating loads, a tight fit is necessary. (Refer to **Table 7.1**) "Raceways under rotating loads" refers to raceways receiving loads rotating relative to their radial direction. For raceways under static loads, on the other hand, a loose fit is sufficient.

(Example) Rotating inner ring load = the direction of the radial load on the inner ring is rotating relatively

(2) For non-separable bearings, such as deep groove ball bearings, it is generally recommended that either the inner ring or outer ring be given a loose fit.

Table 7.1 Radial load and bearing fit

Illustration	Bearing rotation	Ring load	Fit
Static load 	 Inner ring: Rotating Outer ring: Stationary	Rotating inner ring load	Inner ring : Tight fit
Unbalanced load 	 Inner ring: Stationary Outer ring: Rotating	Static outer ring load	Outer ring : Loose fit
Static load 	 Inner ring: Stationary Outer ring: Rotating	Static inner ring load	Inner ring : Loose fit
Unbalanced load 	 Inner ring: Rotating Outer ring: Stationary	Rotating outer ring load	Outer ring : Tight fit

7.3.2 Recommended Fits

Bearing fit is governed by the selection tolerances for bearing shaft diameters and housing bore diameters.

Widely used fits for 0 Class tolerance bearings and various shaft and housing bore diameter tolerances are shown in **Table 7.1**.

Generally-used, standard fits for most types of bearings and operating conditions are shown in **Tables 7.2** - **7.7**.

Table 7.2: Fits for radial bearings
Table 7.3: Fits for thrust bearings
Table 7.4: Fits for electric motor bearings
Table 7.6: Fits for inch series tapered roller bearings (ANSI Class 4)
Table 7.7: Fits for inch series tapered roller bearings (ANSI Class 3 and 0)
Table 7.5. shows fits and their numerical values.

For special fits or applications, please consult NTN Engineering.

7.3.3 Interference minimum and maximum values

The following points should be considered when it is necessary to calculate the interference for an application:

- In calculating the minimum required amount of interference keep in mind that:
  1) interference is reduced by radial loads
  2) interference is reduced by differences between bearing temperature and ambient temperature
  3) interference is reduced by variation of fitting surfaces
- The upper limit value should not exceed 1/1000 of the shaft diameter.

Required interference calculations are shown below.

(1) Radial loads and required interference

Interference of the inner ring and shaft decreases when a radial load is applied to the bearing. The interference required to secure effective interference is expressed by formulae (7.1) and (7.2).

$F_r \leqq 0.3\, C_{or}$

$$\left.\begin{aligned} \Delta_{dF} &= 0.08\,(d \cdot F_r / B)^{1/2} && \text{N} \\ &= 0.25\,(d \cdot F_r / B)^{1/2} && \{\text{kgf}\} \end{aligned}\right\} \cdots\cdots\cdots (7.1)$$

$F_r > 0.3\, C_{or}$

$$\left.\begin{aligned} \Delta_{dF} &= 0.02\,(F_r / B) && \text{N} \\ &= 0.2\,(F_r / B) && \{\text{kgf}\} \end{aligned}\right\} \cdots\cdots\cdots (7.2)$$

Where,

Δ_{dF} : Required effective interference according to radial load μm
d : Bearing bore diameter mm
B : Inner ring width mm
F_r : Radial load N {kgf}
C_{or} : Basic static load rating N {kgf}

(2) Temperature difference and required interference

Interference between inner rings and steel shafts is reduced as a result of temperature increases (difference between bearing temperature and ambient temperature, ΔT) caused by bearing rotation. Calculation of the minimum required amount of interference in such cases is shown in formula (7.3).

$$\Delta_{dT} = 0.0015 \cdot d \cdot \Delta T \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots (7.3)$$

Δ_{dT} : Required effective interference for temperature difference μm
ΔT : Difference between bearing temperature and ambient temperature ˚C
d : Bearing bore diameter mm

(3) Fitting surface variation and required interference

Interference decreases because the fitting surface is smoothened by fitting (surface roughness is reduced). The amount the interference decreases depends on roughness of the fitting surface. It is generally necessary to anticipate the following decrease in interference.

For ground shafts: 1.0〜2.5 μm
For lathed shafts: 5.0〜7.0 μm

(4) Maximum interference

When bearing rings are installed with an interference fit, tension or compression stress may occur along their raceways. If interference is too great, this may cause damage to the rings and reduce bearing life. You should try to obtain the previously described upper limit.

7.3.4 Other details

(1) Tight interference fits are recommended for,
- Operating conditions with large vibration or shock loads
- Applications using hollow shafts or housings with thin walls
- Applications using housings made of light alloys or plastic

(2) Small interference fits are preferable for,
- Applications requiring high running accuracy
- Applications using small sized bearings or thin walled bearings

(3) Consideration must also be given to the fact that fit selection will effect internal bearing clearance selection. (refer to page insert A-58)

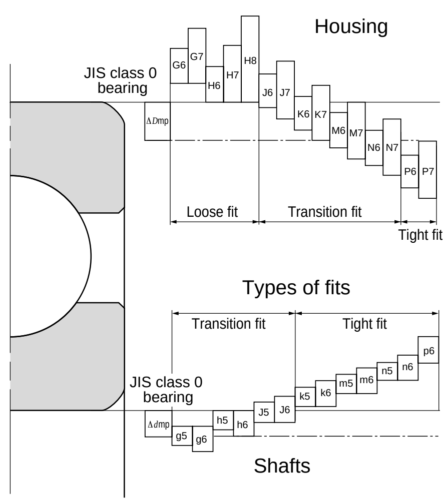


Fig 7.1 State of fitting

II

(4) A particular type of fit is recommended for SL type cylindrical roller bearings.(See page C-44.)

II

Table 7.2 General standards for radial bearing fits (JIS Class 0, 6X, 6)

Table 7.2 (1) Tolerance class of shafts commonly used for radial bearings (classes 0, 6X and 6)

Conditions		Ball bearings		Cylindrical roller bearing Tapered roller bearing		Spherical roller bearing		Shaft tolerance class	Remarks
		Shaft diameter (mm)							
		Over	Under	Over	Under	Over	Under		
Cylindrical bore bearing (Classes 0, 6X and 6)									
Inner ring rotational load or load of undetermined direction	Light load ❶ or fluctuating load	— 18 100 —	18 100 200 —	— — 40 140	— 40 140 200	— — — —	— — — —	h5 js6 k6 m6	When greater accuracy is required js5, k5, and m5 may be substituted for js6, k6, and m6.
	Ordinary ❶ load	— 18 100 140 200 — —	18 100 140 200 280 — —	— — 40 100 140 200 —	— 40 100 140 200 400 —	— — 40 65 100 140 280	— 40 65 100 140 280 500	js5 k5 m5 m6 n6 p6 r6	Alteration of inner clearances to accommodate fit is not a consideration with single-row angular contact bearings and tapered roller bearings. Therefore, k5 and m5 may be substituted for k6 and m6.
	Heavy load ❶ or impact load	— — —	— — —	50 140 200	140 200 —	50 100 140	100 140 200	n6 p6 r6	Use bearings with larger internal clearances than CN clearance bearings.
Inner ring static load	Inner ring must move easily over shaft	Overall shaft diameter						g6	When greater accuracy is required use g5. For large bearings, f6 will suffice for to facilitate movement.
	Inner does not have to move easily over shaft	Overall shaft diameter						h6	When greater accuracy is required use h5.
Center axial load		Overall shaft diameter						js6	Generally, shaft and inner rings are not fixed using interferance.
Tapered bore bearing (class 0) (with adapter or withdrawal sleeve)									
Overall load		Overall shaft diameter						h9/IT5 ❷	h10/IT7 ❷ will suffice for power transmitting shafts.

Table 7.2 (2) Fit with shaft (fits for tapered bore bearings (Class 0) with adapter assembly/withdrawal sleeve)

All loads	All bearing types	All shaft diameters	Tolerance class	h9 / IT5 ❷	General applications
				h10/ IT7 ❷	Transmission shafts, etc.

❶ Standards for light loads, normal loads, and heavy loads

- Light loads: equivalent radial load ≦ 0.06 C_r
- Normal loads: 0.06 C_r < equivalent radial load ≦ 0.12 C_r
- Heavy loads: 0.12 C_r < equivalent radial load

❷ IT5 and IT7 show shaft roundness tolerances, cylindricity tolerances, and related values.

Note: All values and fits listed in the above tables are for solid steel shafts.

II

Table 7.2 (3) Tolerance class of housing bore commonly used for radial bearings (classes 0, 6X and 6)

Conditions				Toleration class of housing bore	Remarks
Housing	**Types of load**		**Outer ring axial ❷ direction movement**		
Single housing or divided housing	Outer ring static load	All types of loads	Able to move.	H7	G7 will suffice for large bearings or bearings with large temperature differential between the outer ring and housing.
		Light load ❶ or ordinary load ❶	Able to move.	H8	—
		Shaft and inner ring become hot.	Able to move easily.	G7	F7 will suffice for large bearings or bearings with large temperature differential between the outer ring and housing.
Single housing		Requires precision rotation with light or ordinary loads.	As a rule, cannot move.	K6	Primarily applies to roller bearings.
			Able to move.	JS6	Primarily applies to ball bearings.
		Requires quiet operation.	Able to move.	H6	—
	Indeterminate load	Light or ordinary load	Able to move.	JS7	If precision is required, JS6 and K6 are used in place of JS7 and K7.
		Ordinary load or heavy load ❶	As a rule, cannot move.	K7	
		Large impact load	Cannot move.	M7	—
	Outer ring rotational load	Light or fluctuating load	Cannot move.	M7	—
		Ordinary or heavy load	Cannot move.	N7	Primarily applies to ball bearings.
		Heavy load or large impact load with thin housing	Cannot move.	P7	Primarily applies to roller bearings.

❶ Standards for light loads, normal loads, and heavy loads

Light loads: equivalent radial load $\leqq 0.06\ C_r$
Normal loads: $0.06\ C_r <$ equivalent radial load $\leqq 0.12\ C_r$
Heavy loads: $0.12\ C_r <$ equivalent radial load

❷ Indicates whether or not outer ring axial displacement is possible with non-separable type bearings.

Note 1: All values and fits listed in the above tables are for cast iron or steel housings.
2: If only center axial load is applied to the bearing, select a tolerance class that provides clearance for the outer ring in the axial direction.

Table 7.3 Standard fits for thrust bearings (JIS Class 0 and 6)

Table 7.3 (1) Shaft fits

Bearing type	Load conditions		Fit	Shaft diameter mm over incl.	Tolerance class
All thrust bearings	Centered axial load only		Transition fit	All sizes	js6 or h6
Spherical roller thrust bearings	Combined load	Inner ring static load	Transition fit	All sizes	js6
		Inner ring rotating load or Indeterminate load	Transition fit	— ～ 200	k6 or js6
				200 ～ 400	m6 or k6
			Tight fit	400 ～	n6 or m6

Table 7.3 (2) Housing fits

Bearing type	Load conditions		Fit	Tolerance class	Remarks
All thrust bearings	Centered axial load only		Loose fit	Select a tolerance class that will provide clearance between outer ring and housing.	
				H8	Greater accuracy required with thrust ball bearings
Spherical roller thrust bearings	Combined load	Outer ring static load		H7	——
		Indeterminate load or outer ring rotating load	Transition fit	K7	Normal operating conditions
				M7	For relatively large radial loads

Note: All values and fits listed in the above tables are for cast iron or steel housings.

Table 7.4 Fits for electric motor bearings

Bearing type	Shaft fits		Housing fits	
	Shaft diameter mm over incl.	Tolerance class	Housing bore diameter	Tolerance class
Deep groove ball bearings	～ 18 18 ～100 100 ～160	j5 k5 m5	All sizes	H6 or J6
Cylindrical roller bearings	～ 40 40 ～160 160 ～200	k5 m5 n6	All sizes	H6 or J6

Table 7.5 Numeric value table of fitting for radial bearing of 0 class

Table 7.5 (1) Fitting against shaft

Nominal bore diameter of bearing d mm over	incl.	Mean bore diameter deviation ❶ Δ_{dmp} high	low	g5 bearing / shaft	g6 bearing / shaft	h5 bearing / shaft	h6 bearing / shaft	j5 bearing / shaft	js5 bearing / shaft	j6 bearing / shaft
3	6	0	-8	4T～ 9L	4T～12L	8T～ 5L	8T～ 8L	11T～ 2L	10.5T～ 2.5L	14T～ 2L
6	10	0	-8	3T～11L	3T～14L	8T～ 6L	8T～ 9L	12T～ 2L	11T ～ 3L	15T～ 2L
10	18	0	-8	2T～14L	2T～17L	8T～ 8L	8T～11L	13T～ 3L	12T ～ 4L	16T～ 3L
18	30	0	-10	3T～16L	3T～20L	10T～ 9L	10T～13L	15T～ 4L	14.5T～ 4.5L	19T～ 4L
30	50	0	-12	3T～20L	3T～25L	12T～11L	12T～16L	18T～ 5L	17.5T～ 5.5L	23T～ 5L
50	80	0	-15	5T～23L	5T～29L	15T～13L	15T～19L	21T～ 7L	21.5T～ 6.5L	27T～ 7L
80	120	0	-20	8T～27L	8T～34L	20T～15L	20T～22L	26T～ 9L	27.5T～ 7.5L	33T～ 9L
120 140 160	140 160 180	0	-25	11T～32L	11T～39L	25T～18L	25T～25L	32T～11L	34T ～ 9L	39T～11L
180 200 225	200 225 250	0	-30	15T～35L	15T～44L	30T～20L	30T～29L	37T～13L	40T ～10L	46T～13L
250 280	280 315	0	-35	18T～40L	18T～49L	35T～23L	35T～32L	42T～16L	46.5T～11.5L	51T～16L
315 355	355 400	0	-40	22T～43L	22T～54L	40T～25L	40T～36L	47T～18L	52.5T～12.5L	58T～18L
400 450	450 500	0	-45	25T～47L	25T～60L	45T～27L	45T～40L	52T～20L	58.5T～13.5L	65T～20L

❶ Above table is not applicable to tapered roller bearings whose bore diameter d is 30mm or less.

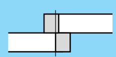
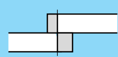
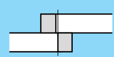
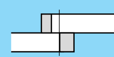
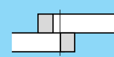
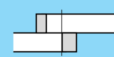
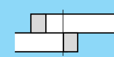
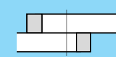
Table 7.5 (2) Fitting against housing

Nominal outside diameter of bearing D mm over	incl.	Mean outside diameter deviation ❷ Δ_{Dmp} high	low	G7 housing / bearing	H6 housing / bearing	H7 housing / bearing	J6 housing / bearing	J7 housing / bearing	Js7 housing / bearing	K6 housing / bearing
6	10	0	-8	5L～ 28L	0～17L	0～ 23L	4T～13L	7T～16L	7.5T～15.5L	7T～10L
10	18	0	-8	6L～ 32L	0～19L	0～ 26L	5T～14L	8T～18L	9T ～17L	9T～10L
18	30	0	-9	7L～ 37L	0～22L	0～ 30L	5T～17L	9T～21L	10.5T～19.5L	11T～11L
30	50	0	-11	9L～ 45L	0～27L	0～ 36L	6T～21L	11T～25L	12.5T～23.5L	13T～14L
50	80	0	-13	10L～ 53L	0～32L	0～ 43L	6T～26L	12T～31L	15T ～28L	15T～17L
80	120	0	-15	12L～ 62L	0～37L	0～ 50L	6T～31L	13T～37L	17.5T～32.5L	18T～19L
120	150	0	-18	14L～ 72L	0～43L	0～ 58L	7T～36L	14T～44L	20T ～38L	21T～22L
150	180	0	-25	14L～ 79L	0～50L	0～ 65L	7T～43L	14T～51L	20T ～45L	21T～29L
180	250	0	-30	15L～ 91L	0～59L	0～ 76L	7T～52L	16T～60L	23T ～53L	24T～35L
250	315	0	-35	17L～104L	0～67L	0～ 87L	7T～60L	16T～71L	26T ～61L	27T～40L
315	400	0	-40	18L～115L	0～76L	0～ 97L	7T～69L	18T～79L	28.5T～68.5L	29T～47L
400	500	0	-45	20L～128L	0～85L	0～108L	7T～78L	20T～88L	31.5T～76.5L	32T～53L

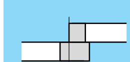
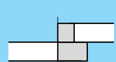
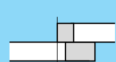
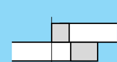
❷ Above table is not applicable to tapered roller bearings whose outside diameter D is 150mm or less.
Note: Fitting symbol "L" indicates clearance and "T" indicates interference.

II

Unit µm

js6	k5	k6	m5	m6	n6	p6	r6	Nominal bore diameter of bearing *d* mm	
bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft	bearing shaft		
								over	incl.
12T～4L	14T～1T	17T～1T	17T～4T	20T～4T	24T～8T	28T～12T	– –	3	6
12.5T～4.5L	15T～1T	18T～1T	20T～6T	23T～6T	27T～10T	32T～15T	– –	6	10
13.5T～5.5L	17T～1T	20T～1T	23T～7T	26T～7T	31T～12T	37T～18T	– –	10	18
16.5T～6.5L	21T～2T	25T～2T	27T～8T	31T～8T	38T～15T	45T～22T	– –	18	30
20T～8L	25T～2T	30T～2T	32T～9T	37T～9T	45T～17T	54T～26T	– –	30	50
24.5T～9.5L	30T～2T	36T～2T	39T～11T	45T～11T	54T～20T	66T～32T	– –	50	80
31T～11L	38T～3T	45T～2T	48T～13T	55T～13T	65T～23T	79T～37T	– –	80	120
37.5T～12.5L	46T～3T	53T～3T	58T～15T	65T～15T	77T～27T	93T～43T	113T～63T	120	140
							115T～65T	140	160
							118T～68T	160	180
44.5T～14.5L	54T～4T	63T～4T	67T～17T	76T～17T	90T～31T	109T～50T	136T～77T	180	200
							139T～80T	200	225
							143T～84T	225	250
51T～16L	62T～4T	71T～4T	78T～20T	87T～20T	101T～34T	123T～56T	161T～94T	250	280
							165T～98T	280	315
58T～18L	69T～4T	80T～4T	86T～21T	97T～21T	113T～37T	138T～62T	184T～108T	315	355
							190T～114T	355	400
65T～20L	77T～5T	90T～4T	95T～23T	108T～23T	125T～40T	153T～68T	211T～126T	400	450
							217T～132T	450	500

Unit µm

K7	M7	N7	P7	Nominal outside diameter of bearing *D* mm	
housing bearing	housing bearing	housing bearing	housing bearing		
				over	incl.
10T～13L	15T～8L	19T～4L	24T～1T	6	10
12T～14L	18T～8L	23T～3L	29T～3T	10	18
15T～15L	21T～9L	28T～2L	35T～5T	18	30
18T～18L	25T～11L	33T～3L	42T～6T	30	50
21T～22L	30T～13L	39T～4L	51T～8T	50	80
25T～25L	35T～15L	45T～5L	59T～9T	80	120
28T～30L	40T～18L	52T～6L	68T～10T	120	150
28T～37L	40T～25L	52T～13L	68T～3T	150	180
33T～43L	46T～30L	60T～16L	79T～3T	180	250
36T～51L	52T～35L	66T～21L	88T～1T	250	315
40T～57L	57T～40L	73T～24L	98T～1T	315	400
45T～63L	63T～45L	80T～28L	108T～0	400	500

II

Table 7.6 General fitting standards for tapered roller bearings using US customary unit (ANSI class 4)

Table 7.6 (1) Fit with shaft

Unit μm

Operating conditions		Nominal bearing bore diameter d mm over		incl.	Bore diameter tolerance Δ_{ds} high	low	Shaft diameter tolerance high	low	Fitting❶ max		min	Remark
Inner ring rotational load	Ordinary load		~	76.2	+13	0	+ 38	+ 25	38T	~	12T	Applicable when slight impact load is applied as well.
		76.2	~	304.8	+25	0	+ 64	+ 38	64T	~	13T	
		304.8	~	609.6	+51	0	+127	+ 76	127T	~	25T	
		609.6	~	914.4	+76	0	+190	+114	190T	~	38T	
	Heavy load Impact load		~	76.2	+13	0	+ 64	+ 38	38T	~	12T	
		76.2	~	304.8	+25	0	0.5 μm mean interference per 1 mm of inner ring bore diameter. Minimum interference is 25 μm. Tolerance for the shaft is adjusted to match tolerance of bearing bore diameter.					
		304.8	~	609.6	+51	0						
		609.6	~	914.4	+76	0						
Outer ring rotational load	Inner ring does not have to move easily over shaft with ordinary load.		~	76.2	+13	0	+ 13	0	13T	~	13L	Not applicable when impact load is applied.
		76.2	~	304.8	+25	0	+ 25	0	25T	~	25L	
		304.8	~	609.6	+51	0	+ 51	0	51T	~	51L	
		609.6	~	914.4	+76	0	+ 76	0	76T	~	76L	
	Inner ring must move easily over shaft with ordinary load.		~	76.2	+13	0	0	- 13	0	~	13L	
		76.2	~	304.8	+25	0	0	- 25	0	~	50L	
		304.8	~	609.6	+51	0	0	- 51	0	~	102L	
		609.6	~	914.4	+76	0	0	- 76	0	~	152L	

Table 7.6 (2) Fit with housing

Unit μm

Operating conditions		Nominal bearing outer diameter D mm over		incl.	Outer diameter tolerance Δ_{Ds} high	low	Housing bore diameter tolerance high	low	Fitting❶ max		min	Types of fit
Inner ring rotational load	When used on floating- or fixed side		~	76.2	+25	0	+ 76	+ 51	26L	~	76L	loose fit
		76.2	~	127.0	+25	0	+ 76	+ 51	26L	~	76L	
		127.0	~	304.8	+25	0	+ 76	+ 51	26L	~	76L	
		304.8	~	609.6	+51	0	+152	+102	51L	~	152L	
		609.6	~	914.4	+76	0	+229	+152	76L	~	229L	
	When outer ring is adjusted in axial direction		~	76.2	+25	0	+ 25	0	25T	~	25L	transition fit
		76.2	~	127.0	+25	0	+ 25	0	25T	~	25L	
		127.0	~	304.8	+25	0	+ 51	0	25T	~	51L	
		304.8	~	609.6	+51	0	+ 76	+ 26	25T	~	76L	
		609.6	~	914.4	+76	0	+127	+ 51	25T	~	127L	
	When outer ring is not adjusted in axial direction		~	76.2	+25	0	- 13	- 38	63T	~	13T	tight fit
		76.2	~	127.0	+25	0	- 25	- 51	76T	~	25T	
		127.0	~	304.8	+25	0	- 25	- 51	76T	~	25T	
		304.8	~	609.6	+51	0	- 25	- 76	127T	~	25T	
		609.6	~	914.4	+76	0	- 25	-102	178T	~	25T	
Outer ring rotational load	When outer ring is not adjusted in axial direction		~	76.2	+25	0	- 13	- 38	63T	~	13T	
		76.2	~	127.0	+25	0	- 25	- 51	76T	~	25T	
		127.0	~	304.8	+25	0	- 25	- 51	76T	~	25T	
		304.8	~	609.6	+51	0	- 25	- 76	127T	~	25T	
		609.6	~	914.4	+76	0	- 25	-102	178T	~	25T	

❶ Fitting symbol "L" indicates clearance and "T" indicates interference.

II

Table 7.7 General fitting standards for tapered roller bearings using US customary unit (ANSI classes 3 and 0)

Table 7. (1) Fit with shaft

Unit µm

Operating conditions		Nominal bearing bore diameter d mm over	incl.	Bore diameter tolerance Δ_{ds} high	low	Shaft diameter tolerance high	low	Fitting ❶ max	min
Inner ring rotational load	Precision machine tool spindles		304.8	+13	0	+ 30	+ 18	30T	5T
		304.8	609.6	+25	0	+ 64	+ 38	64T	13T
		609.6	914.4	+38	0	+102	+ 64	102T	26T
	Heavy load Impact load High-speed rotation		76.2	+13	0	Minimum interference is 0.25 µm per inner ring bore diameter.			
		76.2	304.8	+13	0				
		304.8	609.6	+25	0				
		609.6	914.4	+38	0				
Outer ring rotational load	Precision machine tool spindles		304.8	+13	0	+ 13	0	30T	5T
		304.8	609.6	+25	0	+ 25	0	64T	13T
		609.6	914.4	+38	0	+102	0	102T	26T

Note: For class 0, bearing bore diameter d applies to 241.3 mm or less.

Table 7.7 (2) Fit with housing

Unit µm

Operating conditions		Nominal bearing outer diameter D mm over	incl.	Outer diameter tolerance Δ_{Ds} high	low	Housing bore diameter tolerance high	low	Fitting ❶ max	min	Type of fit
Inner ring rotational load	When used for floating-side		152.4	+13	0	+ 38	+ 25	12L	38L	loose fit
		152.4	304.8	+13	0	+ 38	+ 25	12L	38L	
		304.8	609.6	+25	0	+ 64	+ 38	13L	64L	
		609.6	914.4	+38	0	+ 89	+ 51	13L	89L	
	When used for fixed side		152.4	+13	0	+ 25	+ 13	0	25L	
		152.4	304.8	+13	0	+ 25	+ 13	0	25L	
		304.8	609.6	+25	0	+ 51	+ 25	0	51L	
		609.6	914.4	+38	0	+ 76	+ 38	0	76L	
	When outer ring is adjusted in axial direction		152.4	+13	0	+ 13	0	13T	13L	transition fit
		152.4	304.8	+13	0	+ 13	0	13T	13L	
		304.8	609.6	+13	0	+ 25	0	25T	25L	
		609.6	914.4	+38	0	+ 38	0	38T	38L	
	When outer ring is not adjusted in axial direction		152.4	+13	0	0	- 13	26T	0	tight fit
		152.4	304.8	+13	0	0	- 25	38T	0	
		304.8	609.6	+25	0	0	- 25	50T	0	
		609.6	914.4	+38	0	0	- 38	76T	0	
Outer ring rotational load	Ordinary load When outer ring is not adjusted in axial direction		152.4	+13	0	- 13	- 25	38T	13T	
		152.4	304.8	+13	0	- 13	- 38	51T	13T	
		304.8	609.6	+25	0	- 13	- 38	63T	13T	
		609.6	914.4	+38	0	- 13	- 51	89T	13T	

❶ Fitting symbol "L" indicates clearance and "T" indicates interference.
Note: For class 0, bearing outer diameter D applies to 304.8 mm or less.

8. Bearing Internal Clearance and Preload

8.1 Bearing internal clearance

Bearing internal clearance is the amount of internal free movement before mounting.

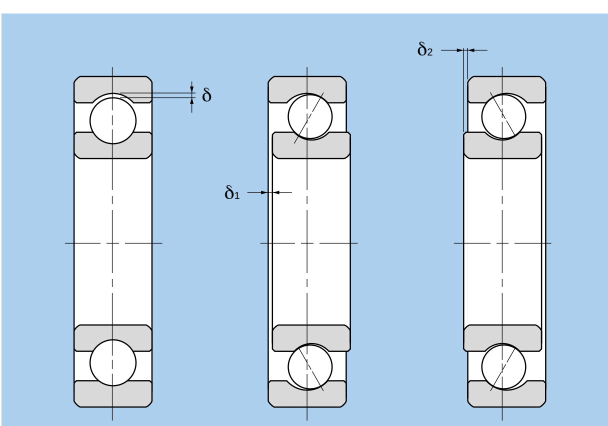
As shown in **Fig. 8.1**, when either the inner ring or the outer ring is fixed and the other ring is free to move, displacement can take place in either an axial or radial direction. This amount of displacement (radially or axially) is termed the internal clearance and, depending on the direction, is called the radial internal clearance or the axial internal clearance.

When the internal clearance of a bearing is measured, a slight measurement load is applied to the raceway so the internal clearance may be measured accurately. However, at this time, a slight amount of elastic deformation of the bearing occurs under the measurement load, and the clearance measurement value (measured clearance) is slightly larger than the true clearance. This difference between the true bearing clearance and the increased amount due to the elastic deformation must be compensated for. These compensation values are given in **Table 8.1**. For roller bearings the amount of elastic deformation can be ignored.

The internal clearance values for each bearing class are shown in **Tables 8.3** through **8.11**.

8.2 Internal clearance selection

The internal clearance of a bearing under operating conditions (effective clearance) is usually smaller than the same bearing's initial clearance before being installed and operated. This is due to several factors including bearing fit, the difference in temperature between the inner and outer rings, etc. As a bearing's operating clearance has an effect on bearing life, heat generation, vibration, noise, etc.; care must be taken in selecting the most suitable operating clearance.



Radial clearance = δ Axial clearance = $\delta_1 + \delta_2$

Fig. 8.1 Internal clearance

8.2.1 Criteria for selecting bearing internal clearance

A bearing's life is theoretically maximum when operating clearance is slightly negative at steady operation. In reality it is however difficult to constantly maintain this optimal condition. If the negative clearance becomes enlarged by fluctuating operating conditions, heat will be produced and life will decrease dramatically. Under ordinary circumstances you should therefore select an initial internal clearance where the operating clearance is slightly larger than zero.

For ordinary operating conditions, use fitting for ordinary loads. If rotational speed and operating temperature are ordinary, selecting normal clearance enables you to obtain the proper operating clearance. **Table 8.2** gives examples applying internal clearances other than CN (normal) clearance.

8.2.2 Calculation of operating clearance

Operating clearance of a bearing can be calculated from initial bearing internal clearance and decrease in internal clearance due to interference and decrease in internal clearance due to difference in temperature of the inner and outer rings.

$$\delta_{eff} = \delta_o - (\delta_f + \delta_t) \quad \cdots\cdots\cdots\cdots (8.1)$$

where,

δ_{eff} : Effective internal clearance, mm
δ_o : Bearing internal clearance, mm
δ_f : Reduced amount of clearance due to

Table 8.1 Adjustment of radial internal clearance based on measured load (deep groove ball bearing) Unit μm

Nominal Bore Diameter *d* mm over	incl.	Measuring Load N	{kgf}	Internal clearance adjustmer C2	CN	C3	C4	C5
10 ❶	18	24.5	{2.5}	3~4	4	4	4	4
18	50	49	{5}	4~5	5	6	6	6
50	200	147	{15}	6~8	8	9	9	9

❶ This diameter is included in the group.

Table 8.2 Examples of applications where bearing clearances other than CN (normal) clearance are used

Operating conditions	Applications	Selected clearance
With heavy or shock load, clearance is large.	Railway vehicle axles	C3
	Vibration screens	C3, C4
With indeterminate load, both inner and outer rings are tight-fitted.	Railway vehicle traction motors	C4
	Tractors and final speed regulators	C4
Shaft or inner ring is heated.	Paper making machines and driers	C3, C4
	Rolling mill table rollers	C3
Reduction of noise and vibration when rotating.	Micromotors	C2, CM
Adjustment of clearance to minimize shaft runout.	Main spindles of lathes (Double-row cylindrical roller bearings)	C9NA, C0NA
Loose fitting for both inner and outer rings.	Compressor roll neck	C2

II

interference, mm

δ_t : Reduced amount of clearance due to temperature differential of inner and outer rings, mm

(1) Reduced clearance due to interference

When bearings are installed with interference fits on shafts and in housings, the inner ring will expand and the outer ring will contract; **thus reducing the bearings' internal clearance.** The amount of expansion or contraction varies depending on the shape of the bearing, the shape of the shaft or housing, dimensions of the respective parts, and the type of materials used. The differential can range from approximately **70% to 90% of the effective interference.**

$$\delta_f = (0.70 \sim 0.90)\, \Delta_{deff} \quad \cdots\cdots (8.2)$$

where,

δ_f : Reduced amount of clearance due to interference, mm

Δ_{deff} : Effective interference, mm

(2) Reduced internal clearance due to inner/outer ring temperature difference.

During operation, normally the outer ring will range from 5 to 10˚C cooler than the inner ring or rotating parts. However, if the cooling effect of the housing is large, the shaft is connected to a heat source, or a heated substance is conducted through the hollow shaft; the temperature difference between the two rings can be even greater. **The amount of internal clearance is thus further reduced by the differential expansion of the two rings.**

$$\delta_t = \alpha \cdot \Delta T \cdot D_o \quad \cdots\cdots (8.3)$$

where,

δ_t : Amount of reduced clearance due to heat differential, mm

α : Bearing material expansion coefficient 12.5×10^{-6}/°C

ΔT : Inner/outer ring temperature differential, ℃

D_o : Outer ring raceway diameter, mm

Outer ring raceway diameter, D_o, values can be approximated by using formula (8.4) or (8.5).

For ball bearings and spherical roller bearings,

$$D_o = 0.20\,(d + 4.0D) \quad \cdots\cdots (8.4)$$

For roller bearings (except Spherical roller bearing),

$$D_o = 0.25\,(d + 3.0D) \quad \cdots\cdots (8.5)$$

where,

d : Bearing bore diameter, mm

D : Bearing outside diameter, mm

Table 8.3 Radial internal clearance of deep groove ball bearings

Unit μm

Nominal bore diameter d mm		C2		CN		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
—	2.5	0	6	4	11	10	20	—	—	—	—
2.5	6	0	7	2	13	8	23	—	—	—	—
6	10	0	7	2	13	8	23	14	29	20	37
10	18	0	9	3	18	11	25	18	33	25	45
18	24	0	10	5	20	13	28	20	36	28	48
24	30	1	11	5	20	13	28	23	41	30	53
30	40	1	11	6	20	15	33	28	46	40	64
40	50	1	11	6	23	18	36	30	51	45	73
50	65	1	15	8	28	23	43	38	61	55	90
65	80	1	15	10	30	25	51	46	71	65	105
80	100	1	18	12	36	30	58	53	84	75	120
100	120	2	20	15	41	36	66	61	97	90	140
120	140	2	23	18	48	41	81	71	114	105	160
140	160	2	23	18	53	46	91	81	130	120	180
160	180	2	25	20	61	53	102	91	147	135	200
180	200	2	30	25	71	63	117	107	163	150	230
200	225	2	35	25	85	75	140	125	195	175	265
225	250	2	40	30	95	85	160	145	225	205	300
250	280	2	45	35	105	90	170	155	245	225	340
280	315	2	55	40	115	100	190	175	270	245	370
315	355	3	60	45	125	110	210	195	300	275	410
355	400	3	70	55	145	130	240	225	340	315	460
400	450	3	80	60	170	150	270	250	380	350	510
450	500	3	90	70	190	170	300	280	420	390	570
500	560	10	100	80	210	190	330	310	470	440	630
560	630	10	110	90	230	210	360	340	520	490	690

II

Table 8.4 Radial internal clearance of self-aligning ball bearings

Nominal bore diameter *d* mm		Bearing with cylindrical bore									
		C2		CN		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
2.5	6	1	8	5	15	10	20	15	25	21	33
6	10	2	9	6	17	12	25	19	33	27	42
10	14	2	10	6	19	13	26	21	35	30	48
14	18	3	12	8	21	15	28	23	37	32	50
18	24	4	14	10	23	17	30	25	39	34	52
24	30	5	16	11	24	19	35	29	46	40	58
30	40	6	18	13	29	23	40	34	53	46	66
40	50	6	19	14	31	25	44	37	57	50	71
50	65	7	21	16	36	30	50	45	69	62	88
65	80	8	24	18	40	35	60	54	83	76	108
80	100	9	27	22	48	42	70	64	96	89	124
100	120	10	31	25	56	50	83	75	114	105	145
120	140	10	38	30	68	60	100	90	135	125	175
140	160	15	44	35	80	70	120	110	161	150	210

Table 8.5 (1) Radial internal clearance for duplex angular contact ball bearings

Unit μm

Nominal bore diameter *d* mm		C1		C2		CN		C3		C4	
over	incl.	min	max	min	max	min	max	min	max	min	max
—	10	3	8	6	12	8	15	15	22	22	30
10	18	3	8	6	12	8	15	15	24	30	40
18	30	3	10	6	12	10	20	20	32	40	55
30	50	3	10	8	14	14	25	25	40	55	75
50	80	3	11	11	17	17	32	32	50	75	95
80	100	3	13	13	22	22	40	40	60	95	120
100	120	3	15	15	30	30	50	50	75	110	140
120	150	3	16	16	33	35	55	55	80	130	170
150	180	3	18	18	35	35	60	60	90	150	200
180	200	3	20	20	40	40	65	65	100	180	240

Note: The clearance group in the table is applied only to contact angles in the table below.

Contact angle symbol	Nominal contact angle	Applicable clearance group❷
C	15°	C1, C2
A❶	30°	C2, CN, C3
B	40°	CN, C3, C4

❶ Not indicated for bearing number.
❷ For information concerning clearance other than applicable clearance, please contact NTN Engineering.

Table 8.5 (2) Radial internal clearance of self-aligning ball bearings

Unit μm

Nominal bore diameter *d* mm		C2		CN		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
10 only		0	10	5	15	10	21	16	28	24	36
10	18	1	11	6	16	12	23	19	31	28	40
18	24	1	11	6	16	13	24	21	33	31	43
24	30	1	13	6	19	13	26	21	35	31	45
30	40	2	15	7	22	15	30	24	39	35	50
40	50	2	15	9	24	17	32	28	45	40	57
50	65	0	15	7	24	16	33	28	48	41	61
65	80	1	17	11	31	21	42	34	56	50	74
80	100	3	20	13	36	25	49	40	65	58	67

Table 8.6 Radial internal clearance of bearings for electric motor

Unit μm

Nominal bore diameter *d* mm		Radial internal clearance CM			
		Deep groove ball bearings		Cylindrical roller bearings	
over	incl.	min	max	min	max
10 (incl.)	18	4	11	—	—
18	24	5	12	—	—
24	30	5	12	15	30
30	40	9	17	15	30
40	50	9	17	20	35
50	65	12	22	25	40
65	80	12	22	30	45
80	100	18	30	35	55
100	120	18	30	35	60
120	140	24	38	40	65
140	160	24	38	50	80
160	180	—	—	60	90
180	200	—	—	65	100

Note 1: Suffix CM is added to bearing numbers.
Example: 6205ZZCM
2: Clearance not interchangeable for cylindrical roller bearings.

II

Unit μm

Bearing with tapered bore										Nominal bore diameter d mm	
C2		CN		C3		C4		C5			
min	max	min	max	min	max	min	max	min	max	over	incl.
—	—	—	—	—	—	—	—	—	—	2.5	6
—	—	—	—	—	—	—	—	—	—	6	10
—	—	—	—	—	—	—	—	—	—	10	14
—	—	—	—	—	—	—	—	—	—	14	18
7	17	13	26	20	33	28	42	37	55	18	24
9	20	15	28	23	39	33	50	44	62	24	30
12	24	19	35	29	46	40	59	52	72	30	40
14	27	22	39	33	52	45	65	58	79	40	50
18	32	27	47	41	61	56	80	73	99	50	65
23	39	35	57	50	75	69	98	91	123	65	80
29	47	42	68	62	90	84	116	109	144	80	100
35	56	50	81	75	108	100	139	130	170	100	120
40	68	60	98	90	130	120	165	155	205	120	140
45	74	65	110	100	150	140	191	180	240	140	160

Table 8.7 Interchangeable radial internal clearance for cylindrical roller bearing (cylindrical bore)

Unit μm

Nominal bore diameter d mm		C2		CN		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
—	10	0	25	20	45	35	60	50	75	—	—
10	24	0	25	20	45	35	60	50	75	65	90
24	30	0	25	20	45	35	60	50	75	70	95
30	40	5	30	25	50	45	70	60	85	80	105
40	50	5	35	30	60	50	80	70	100	95	125
50	65	10	40	40	70	60	90	80	110	110	140
65	80	10	45	40	75	65	100	90	125	130	165
80	100	15	50	50	85	75	110	105	140	155	190
100	120	15	55	50	90	85	125	125	165	180	220
120	140	15	60	60	105	100	145	145	190	200	245
140	160	20	70	70	120	115	165	165	215	225	275
160	180	25	75	75	125	120	170	170	220	250	300
180	200	35	90	90	145	140	195	195	250	275	330
200	225	45	105	105	165	160	220	220	280	305	365
225	250	45	110	110	175	170	235	235	300	330	395
250	280	55	125	125	195	190	260	260	330	370	440
280	315	55	130	130	205	200	275	275	350	410	485
315	355	65	145	145	225	225	305	305	385	455	535
355	400	100	190	190	280	280	370	370	460	510	600
400	450	110	210	210	310	310	410	410	510	565	665
450	500	110	220	220	330	330	440	440	550	625	735

II

Table 8.8 Non-interchangeable radial internal clearance for cylindrical roller bearing

Nominal bore diameter *d* mm		Bearing with cylindrical bore											
		C1NA		C2NA		NA❶		C3NA		C4NA		C5NA	
over	incl.	min	max	min	max	min	max	min	max	min	max	min	max
—	10	5	10	10	20	20	30	35	45	45	55	—	—
10	18	5	10	10	20	20	30	35	45	45	55	65	75
18	24	5	10	10	20	20	30	35	45	45	55	65	75
24	30	5	10	10	25	25	35	40	50	50	60	70	80
30	40	5	12	12	25	25	40	45	55	55	70	80	95
40	50	5	15	15	30	30	45	50	65	65	80	95	110
50	65	5	15	15	35	35	50	55	75	75	90	110	130
65	80	10	20	20	40	40	60	70	90	90	110	130	150
80	100	10	25	25	45	45	70	80	105	105	125	155	180
100	120	10	25	25	50	50	80	95	120	120	145	180	205
120	140	15	30	30	60	60	90	105	135	135	160	200	230
140	160	15	35	35	65	65	100	115	150	150	180	225	260
160	180	15	35	35	75	75	110	125	165	165	200	250	285
180	200	20	40	40	80	80	120	140	180	180	220	275	315
200	225	20	45	45	90	90	135	155	200	200	240	305	350
225	250	25	50	50	100	100	150	170	215	215	265	330	380
250	280	25	55	55	110	110	165	185	240	240	295	370	420
280	315	30	60	60	120	120	180	205	265	265	325	410	470
315	355	30	65	65	135	135	200	225	295	295	360	455	520
355	400	35	75	75	150	150	225	255	330	330	405	510	585
400	450	45	85	85	170	170	255	285	370	370	455	565	650
450	500	50	95	95	190	190	285	315	410	410	505	625	720

❶ For bearings with normal clearance, only NA is added to bearing numbers. Ex. NU310NA

Table 8.9 Axial internal clearance for double row and duplex tapered roller bearings (metric series)

Nominal bore diameter *d* mm		Contact angle $\alpha \leqq 27^\circ$ ($e \leqq 0.76$)							
		C2		CN		C3		C4	
over	incl.	min	max	min	max	min	max	min	max
18	24	25	75	75	125	125	170	170	220
24	30	25	75	75	125	145	195	195	245
30	40	25	95	95	165	165	235	210	280
40	50	20	85	85	150	175	240	240	305
50	65	20	85	110	175	195	260	280	350
65	80	20	110	130	220	240	325	325	410
80	100	45	150	150	260	280	390	390	500
100	120	45	175	175	305	350	480	455	585
120	140	45	175	175	305	390	520	500	630
140	160	60	200	200	340	400	540	520	660
160	180	80	220	240	380	440	580	600	740
180	200	100	260	260	420	500	660	660	820
200	225	120	300	300	480	560	740	720	900
225	250	160	360	360	560	620	820	820	1,020
250	280	180	400	400	620	700	920	920	1,140
280	315	200	440	440	680	780	1,020	1,020	1,260
315	355	220	480	500	760	860	1,120	1,120	1,380
355	400	260	560	560	860	980	1,280	1,280	1,580
400	500	300	600	620	920	1,100	1,400	1,440	1,740

Note1: This table applies to bearings contained in the catalog. For information concerning other bearings or bearings using US customary unit, please contact NTN Engineering.
2: The correlation of axial internal clearance (Δ_a) and radial internal clearance (Δ_r) is expressed as $\Delta_r = 0.667 \cdot e \cdot \Delta_a$.
e: Constant (see dimensions table)
3: Bearing series 329X, 330, 322C and 323Cdo not apply to the table.

II

Unit µm

Bearing with tapered bore												Nominal bore diameter d mm	
C9NA❷		C0NA❷		C1NA		C2NA		NA❶		C3NA			
min	max	min	max	min	max	min	max	min	max	min	max	over	incl.
5	5	7	17	10	20	20	30	35	45	45	55	—	10
5	10	7	17	10	20	20	30	35	45	45	55	10	18
5	10	7	17	10	20	20	30	35	45	45	55	18	24
5	10	10	20	10	25	25	35	40	50	50	60	24	30
5	12	10	20	12	25	25	40	45	55	55	70	30	40
5	15	10	20	15	30	30	45	50	65	65	80	40	50
5	15	10	20	15	35	35	50	55	75	75	90	50	65
10	20	15	30	20	40	40	60	70	90	90	110	65	80
10	25	20	35	25	45	45	70	80	105	105	125	80	100
10	25	20	35	25	50	50	80	95	120	120	145	100	120
15	30	25	40	30	60	60	90	105	135	135	160	120	140
15	35	30	45	35	65	65	100	115	150	150	180	140	160
15	35	30	45	35	75	75	110	125	165	165	200	160	180
20	40	30	50	40	80	80	120	140	180	180	220	180	200
20	45	35	55	45	90	90	135	155	200	200	240	200	225
25	50	40	65	50	100	100	150	170	215	215	265	225	250
25	55	40	65	55	110	110	165	185	240	240	295	250	280
30	60	45	75	60	120	120	180	205	265	265	325	280	315
30	65	45	75	65	135	135	200	225	295	295	360	315	355
35	75	50	90	75	150	150	225	255	330	330	405	355	400
45	85	60	100	85	170	170	255	285	370	370	455	400	450
50	95	70	115	95	190	190	285	315	410	410	505	450	500

❷ C9NA, C0NA and C1NA are applied only to precision bearings of Class 5 and higher.

Unit µm

Contact angle $\alpha > 27^\circ$ ($e > 0.76$)								Nominal bore diameter d mm	
C2		CN		C3		C4			
min	max	min	max	min	max	min	max	over	incl.
10	30	30	50	50	70	70	90	18	24
10	30	30	50	60	80	80	100	24	30
10	40	40	70	70	100	90	120	30	40
10	40	40	70	80	110	110	140	40	50
10	40	50	80	90	120	130	160	50	65
10	50	60	100	110	150	150	190	65	80
20	70	70	120	130	180	180	230	80	100
20	70	70	120	150	200	210	260	100	120
20	70	70	120	160	210	210	260	120	140
30	100	100	160	180	240	240	300	140	160
—	—	—	—	—	—	—	—	160	180
—	—	—	—	—	—	—	—	180	200
—	—	—	—	—	—	—	—	200	225
—	—	—	—	—	—	—	—	225	250
—	—	—	—	—	—	—	—	250	280
—	—	—	—	—	—	—	—	280	315
—	—	—	—	—	—	—	—	315	355
—	—	—	—	—	—	—	—	355	400
—	—	—	—	—	—	—	—	400	500

Table 8.10 Radial internal clearance of spherical roller bearings

Nominal bore diameter d mm		Bearing with cylindrical bore									
		C2		CN		C3		C4		C5	
over	incl.	min	max	min	max	min	max	min	max	min	max
14	18	10	20	20	35	35	45	45	60	60	75
18	24	10	20	20	35	35	45	45	60	60	75
24	30	15	25	25	40	40	55	55	75	75	95
30	40	15	30	30	45	45	60	60	80	80	100
40	50	20	35	35	55	55	75	75	100	100	125
50	65	20	40	40	65	65	90	90	120	120	150
65	80	30	50	50	80	80	110	110	145	145	180
80	100	35	60	60	100	100	135	135	180	180	225
100	120	40	75	75	120	120	160	160	210	210	260
120	140	50	95	95	145	145	190	190	240	240	300
140	160	60	110	110	170	170	220	220	280	280	350
160	180	65	120	120	180	180	240	240	310	310	390
180	200	70	130	130	200	200	260	260	340	340	430
200	225	80	140	140	220	220	290	290	380	380	470
225	250	90	150	150	240	240	320	320	420	420	520
250	280	100	170	170	260	260	350	350	460	460	570
280	315	110	190	190	280	280	370	370	500	500	630
315	355	120	200	200	310	310	410	410	550	550	690
355	400	130	220	220	340	340	450	450	600	600	750
400	450	140	240	240	370	370	500	500	660	660	820
450	500	140	260	260	410	410	550	550	720	720	900
500	560	150	280	280	440	440	600	600	780	780	1,000
560	630	170	310	310	480	480	650	650	850	850	1,100
630	710	190	350	350	530	530	700	700	920	920	1,190
710	800	210	390	390	580	580	770	770	1,010	1,010	1,300
800	900	230	430	430	650	650	860	860	1,120	1,120	1,440
900	1,000	260	480	480	710	710	930	930	1,220	1,220	1,570
1,000	1,120	290	530	530	780	780	1,020	1,020	1,330	1,330	1,720
1,120	1,250	320	580	580	860	860	1,120	1,120	1,460	1,460	1,870
1,250	1,400	350	640	640	950	950	1,240	1,240	1,620	1,620	2,080

II

Table 8.11 Axial internal clearance of four points contact ball bearings

Unit μm

Nominal bore diameter d mm		C2		CN		C3		C4	
over	incl.	min	max	min	max	min	max	min	max
17	40	26	66	56	106	96	146	136	186
40	60	36	86	76	126	116	166	156	206
60	80	46	96	86	136	126	176	166	226
80	100	56	106	96	156	136	196	186	246
100	140	66	126	116	176	156	216	206	266
140	180	76	156	136	196	176	236	226	296
180	220	96	176	156	216	196	256	246	316

II

Unit μm

Bearing with tapered bore										Nominal bore diameter d mm	
C2		CN		C3		C4		C5			
min	max	min	max	min	max	min	max	min	max	over	incl.
—	—	—	—	—	—	—	—	—	—	14	18
15	25	25	35	35	45	45	60	60	75	18	24
20	30	30	40	40	55	55	75	75	95	24	30
25	35	35	50	50	65	65	85	85	105	30	40
30	45	45	60	60	80	80	100	100	130	40	50
40	55	55	75	75	95	95	120	120	160	50	65
50	70	70	95	95	120	120	150	150	200	65	80
55	80	80	110	110	140	140	180	180	230	80	100
65	100	100	135	135	170	170	220	220	280	100	120
80	120	120	160	160	200	200	260	260	330	120	140
90	130	130	180	180	230	230	300	300	380	140	160
100	140	140	200	200	260	260	340	340	430	160	180
110	160	160	220	220	290	290	370	370	470	180	200
120	180	180	250	250	320	320	410	410	520	200	225
140	200	200	270	270	350	350	450	450	570	225	250
150	220	220	300	300	390	390	490	490	620	250	280
170	240	240	330	330	430	430	540	540	680	280	315
190	270	270	360	360	470	470	590	590	740	315	355
210	300	300	400	400	520	520	650	650	820	355	400
230	330	330	440	440	570	570	720	720	910	400	450
260	370	370	490	490	630	630	790	790	1,000	450	500
290	410	410	540	540	680	680	870	870	1,100	500	560
320	460	460	600	600	760	760	980	980	1,230	560	630
350	510	510	670	670	850	850	1,090	1,090	1,360	630	710
390	570	570	750	750	960	960	1,220	1,220	1,500	710	800
440	640	640	840	840	1,070	1,070	1,370	1,370	1,690	800	900
490	710	710	930	930	1,190	1,190	1,520	1,520	1,860	900	1,000
530	770	770	1,030	1,030	1,300	1,300	1,670	1,670	2,050	1,000	1,120
570	830	830	1,120	1,120	1,420	1,420	1,830	1,830	2,250	1,120	1,250
620	910	910	1,230	1,230	1,560	1,560	2,000	2,000	2,470	1,250	1,400

8.3 Preload

Normally, bearings are used with a slight internal clearance under operating conditions. However, in some applications, bearings are given an initial load; this means that the bearings' internal clearance is negative before operation. This is called "preload" and is commonly applied to angular ball bearings and tapered roller bearings.

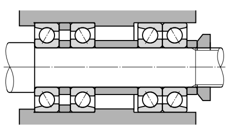
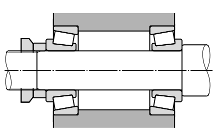
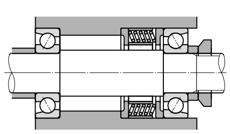
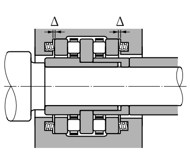
8.3.1 Purpose of preload

The following results are obtained by constant elastic compressive force applied to the contact points of rolling elements and raceway by providing preload.

(1) Bearing's rigidity increases, internal clearance tends not to be produced even when heavy load is applied.
(2) The particular frequency of the bearing increases and is becomes suitable for high-speed rotation.
(3) Shaft runout is suppressed; rotation and position precision are enhanced.
(4) Vibration and noise are controlled.
(5) Sliding of rolling elements by turning, spinning, or pivoting, is controlled and smearing is reduced.
(6) Fretting produced by external vibration is prevented.

Applying excessive preload could result in reduction of life, abnormal heating, or increase in turning torque. You should therefore consider the objectives before determining the amount of preload.

Table 8.12 Preloading methods and characteristics

Method	Basic pattern	Applicable bearings	Object	Characteristics	Applications
Fixed position preload		Angular contact ball bearings	Maintaining accuracy of rotating shaft, preventing vibration increasing rigidity	Preloading is accomplished by a predetermined offset of the rings or by using spacers. For the standard preload see **Table 8.13**.	Grinding machines, lathes, milling machines, measuring instruments
		Tapered roller bearings, thrust ball bearings, angular contact ball bearings	Increasing bearing rigidity	Preload is accomplished by adjusting a threaded screw. The amount of preload is set by measuring the starting torque or axial displacement.	Lathes, milling machines, differential gears of automotives, printing machines, wheel axles
Constant pressure preload		Angular contact ball bearings, deep groove ball bearings, tapered roller bearings (high speed)	Maintaining accuracy and preventing vibration and noise with a constant amount of preload without being affected by loads or temperature	Preloading is accomplished by using coil or belleville springs. for deep groove ball bearings: 4∼10 d N 0.4∼1.0 d {kgf} d: Shaft diameter mm for angular contact ball bearings: see Table 8.13.	Internal grinding machines, electric motors, high speed shafts in small machines, tension reels
		Spherical roller thrust bearings, cylindrical roller thrust bearings, thrust ball bearings	Preload is primarily used to prevent smearing of opposite axial load side when bearing an axial load.	Preload is accomplished by using coil or belleville springs. Recommended preloads are as follows: for thrust ball bearings: $T_1=0.42\,(nC_{oa})^{1.9}\times10^{-13}$ N $=3.275(nC_{oa})^{1.9}\times10^{-13}$ {kgf} $T_2=0.00083\,C_{oa}$ N {kgf} which ever is greater for spherical roller thrust bearings, cylindrical roller thrust bearing $T=0.025\,C_{oa}^{0.8}$ N $=0.0158\,C_{oa}^{0.8}$ {kgf}	Rolling mills, extruding machines

Note: In the above formulas
T = preload, N {kgf}
n = number of revolutions, min^{-1}
C_{oa} = basic static axial load rating, N {kgf}

II

8.3.2 Preloading methods and amounts

The most common method of applying preload on a bearing is change the relative position of the inner and outer rings of the bearing in the axial direction while applying an axial load between bearings on opposing sides. There are two types of preload: fixed position preload and constant pressure preload.

The basic pattern, purpose and characteristics of bearing preloads are shown in **Table 8.12**. The fixed position preload is effective for positioning the two bearings and also for increasing the rigidity. Due to the use of a spring for the constant pressure preload, the preloading amount can be kept constantly, even when the distance between the two bearings fluctuates under the influence of operating heat and load.

Also, the standard preloading amount for the paired angular contact ball bearings is shown in **Table 8.13**. Light and normal preload is applied to prevent general vibration, and medium and heavy preload is applied especially when rigidity is required.

8.3.3 Preload and rigidity

The increased rigidity effect preloading has on bearings is shown in **Fig. 8.2**. When the offset inner rings of the two paired angular contact ball bearings are pressed together, each inner ring is displaced axially by the amount δ_o and is thus given a preload, F_o, in the direction. Under this condition, when external axial load F_a is applied, bearing I will have an increased displacement by the amount δ_a and bearing II's displacement will decrease. At this time the loads applied to bearing I and II are F_{I} and F_{II}, respectively.

Under the condition of no preload, bearing I will be displaced by the amount δ_b when axial load F_a is applied. Since the amount of displacement, δ_a, is less than δ_b, it indicates a higher rigidity for δ_a.

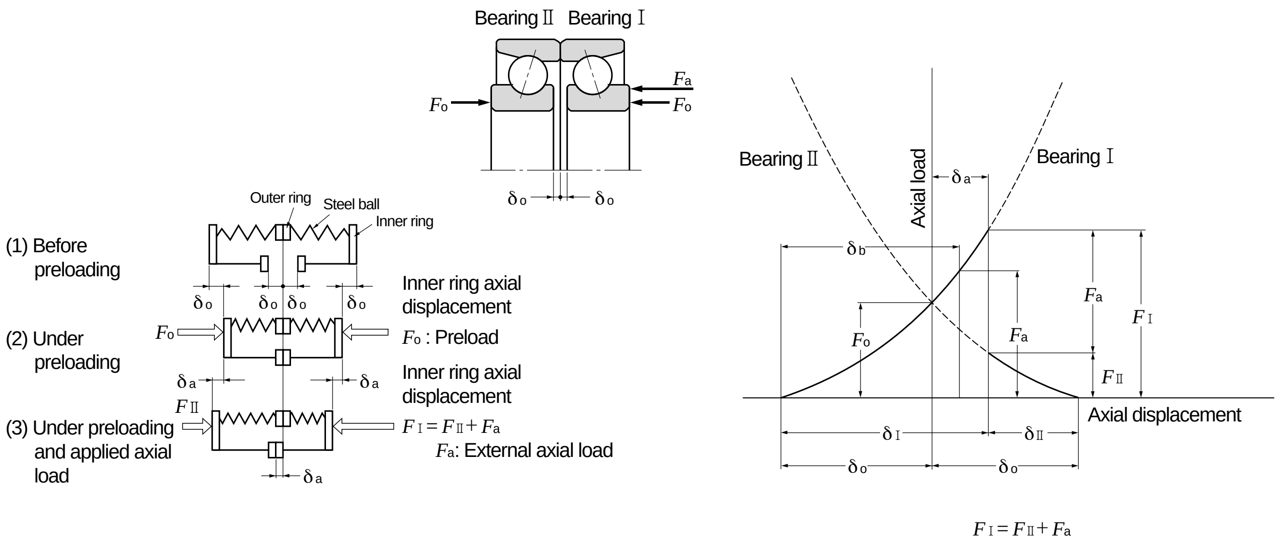


Fig. 8.2 Fixed position preload model diagram and preload diagram

Table 8.13 The normal preload of duplex angular contact ball bearings

Nominal bore diameter *d* mm		Bearing									
		78C				79C, HSB9C				70C, BNT0,	
over	inch	Low GL	Normal GN	Central GM	Heavy GH	Low GL	Normal GN	Central GM	Heavy GH	Low GL	Normal GN
-	12	-	-	-	-	-	-	-	-	20{2}	29{3}
12	18	-	-	-	-	-	-	-	-	20{2}	29{3}
18	32	10{1}	29{3}	78{8}	147{15}	20{2}	49{5}	98{10}	196{20}	29{3}	78{8}
32	40	10{1}	29{3}	78{8}	147{15}	29{3}	78{8}	196{20}	294{30}	49{5}	147{15}
40	50	20{2}	49{5}	98{10}	196{20}	39{4}	98{10}	245{25}	490{50}	49{5}	147{15}
50	65	29{3}	98{10}	196{20}	390{40}	49{5}	118{12}	294{30}	590{60}	98{10}	196{20}
65	80	29{3}	98{10}	196{20}	390{40}	78{8}	196{20}	390{40}	785{80}	98{10}	294{30}
80	90	49{5}	147{15}	294{30}	590{60}	98{10}	245{25}	490{50}	980{100}	147{15}	390{40}
90	95	49{5}	147{15}	294{30}	590{60}	98{10}	245{25}	490{50}	980{100}	147{15}	390{40}
95	100	49{5}	147{15}	294{30}	590{60}	118{12}	294{30}	685{70}	1,470{150}	147{15}	390{40}
100	105	49{5}	147{15}	294{30}	590{60}	118{12}	294{30}	685{70}	1,470{150}	196{20}	590{60}
105	110	78{8}	196{20}	490{50}	980{100}	118{12}	294{30}	685{70}	1,470{150}	196{20}	590{60}
110	120	78{8}	196{20}	490{50}	980{100}	147{15}	390{40}	880{90}	1,960{200}	196{20}	590{60}
120	140	98{10}	294{30}	590{60}	1,270{130}	196{20}	490{50}	980{100}	2,450{250}	294{30}	785{80}
140	150	147{15}	390{40}	785{80}	1,470{150}	245{25}	685{70}	1,470{150}	2,940{300}	294{30}	785{80}
150	160	147{15}	390{40}	785{80}	1,470{150}	245{25}	685{70}	1,470{150}	2,940{300}	490{50}	980{100}
160	170	147{15}	490{50}	980{100}	1,960{200}	245{25}	685{70}	1,470{150}	2,940{300}	490{50}	980{100}
170	180	147{15}	490{50}	980{100}	1,960{200}	294{30}	880{90}	1,960{200}	3,900{400}	490{50}	980{100}
180	190	196{20}	590{60}	1,270{130}	2,450{250}	294{30}	880{90}	1,960{200}	3,900{400}	590{60}	1,470{150}
190	200	196{20}	590{60}	1,270{130}	2,450{250}	490{50}	1,270{130}	2,940{300}	5,900{600}	590{60}	1,470{150}

Nominal bore diameter *d* mm		Bearing						
		79, HSB9			70, HSB0			
over	inch	Normal GN	Central GM	Heavy GH	Low GL	Normal GN	Central GM	Heavy GH
-	12	39{4}	78{8}	147{15}	29{3}	78{8}	147{15}	196{20}
12	18	49{5}	147{15}	196{20}	29{3}	78{8}	147{15}	294{30}
18	32	98{10}	196{20}	294{30}	49{5}	147{15}	294{30}	490{50}
32	40	147{15}	294{30}	590{60}	78{8}	294{30}	590{60}	880{90}
40	50	196{20}	390{40}	635{70}	78{8}	294{30}	590{60}	980{100}
50	65	245{25}	490{50}	785{80}	147{15}	490{50}	880{90}	1,470{150}
65	80	390{40}	785{80}	1,180{120}	147{15}	590{60}	1,470{150}	1,960{200}
80	90	490{50}	980{100}	1,470{150}	196{20}	880{90}	1,960{200}	2,940{300}
90	95	490{50}	980{100}	1,470{150}	196{20}	880{90}	1,960{200}	2,940{300}
95	100	685{70}	1,274{130}	1,960{200}	196{20}	880{90}	1,960{200}	2,940{300}
100	105	685{70}	1,274{130}	1,960{200}	294{30}	980{100}	2,450{250}	3,900{400}
105	110	685{70}	1,274{130}	1,960{200}	294{30}	980{100}	2,450{250}	3,900{400}
110	120	880{90}	1,780{180}	2,940{300}	294{30}	980{100}	2,450{250}	3,900{400}
120	140	980{100}	1,960{200}	3,450{350}	490{50}	1,470{150}	3,450{350}	5,900{600}
140	150	1,270{130}	2,450{250}	4,400{450}	490{50}	1,470{150}	3,450{350}	5,900{600}
150	160	1,270{130}	2,450{250}	4,400{450}	685{70}	2,450{250}	4,900{500}	8,800{900}
160	170	1,270{130}	2,450{250}	4,400{450}	685{70}	2,450{250}	4,900{500}	8,800{900}
170	180	1,780{180}	3,450{350}	5,900{600}	685{70}	2,450{250}	4,900{500}	8,800{900}
180	190	1,780{180}	3,450{350}	5,900{600}	880{90}	3,450{350}	6,850{700}	9,800{1,000}
190	200	2,450{250}	4,900{500}	7,850{800}	880{90}	3,450{350}	6,850{700}	9,800{1,000}

II

Unit N {kgf}

series HSB0C Central GM	HSB0C Heavy GH	72C, BNT2 Low GL	72C, BNT2 Normal GN	72C, BNT2 Central GM	72C, BNT2 Heavy GH	73C Low GL	73C Normal GN	73C Central GM	73C Heavy GH
98{10}	147{15}	20{2}	49{5}	98{10}	196{20}	29{3}	78{8}	147{15}	294{30}
98{10}	196{20}	20{2}	49{5}	147{15}	294{30}	29{3}	78{8}	196{20}	390{40}
147{15}	294{30}	49{5}	98{10}	294{30}	490{50}	76{8}	147{15}	390{40}	685{70}
294{30}	590{60}	78{8}	196{20}	490{50}	785{80}	98{10}	294{30}	590{60}	980{100}
294{30}	685{70}	98{10}	294{30}	590{60}	980{100}	145{15}	390{40}	980{100}	1,960{200}
490{50}	980{100}	147{15}	390{40}	785{80}	1,470{150}	196{20}	590{60}	1,470{150}	2,940{300}
685{70}	1,470{150}	196{20}	490{50}	980{100}	1,960{200}	294{30}	785{80}	1,960{200}	3,900{400}
980{100}	1,960{200}	294{30}	685{70}	1,470{150}	2,940{300}	390{40}	980{100}	2,450{250}	4,900{500}
980{100}	1,960{200}	294{30}	685{70}	1,960{200}	3,900{400}	390{40}	980{100}	2,950{300}	5,900{600}
980{100}	1,960{200}	294{30}	685{70}	1,960{200}	3,900{400}	390{40}	980{100}	2,950{300}	5,900{600}
1,470{150}	2,450{250}	390{40}	980{100}	2,450{250}	4,900{500}	590{60}	1,470{150}	3,450{350}	6,850{700}
1,470{150}	2,450{250}	390{40}	980{100}	2,450{250}	4,900{500}	590{60}	1,470{150}	3,450{350}	6,850{700}
1,470{150}	2,450{250}	390{40}	980{100}	2,450{250}	4,900{500}	590{60}	1,470{150}	3,450{350}	6,850{700}
1,960{200}	3,900{400}	490{50}	1,470{150}	2,940{300}	5,900{600}	785{80}	1,960{200}	4,400{450}	8,800{900}
1,960{200}	3,900{400}	490{50}	1,470{150}	2,940{300}	5,900{600}	785{80}	1,960{200}	4,400{450}	8,800{900}
2,450{250}	5,900{600}	685{70}	1,960{200}	4,400{450}	7,850{800}	880{90}	2,450{250}	5,900{600}	9,800{1,100}
2,450{250}	5,900{600}	685{70}	1,960{200}	4,400{450}	7,850{800}	880{90}	2,450{250}	5,900{600}	9,800{1,100}
2,450{250}	5,900{600}	685{70}	1,960{200}	4,400{450}	7,850{800}	880{90}	2,450{250}	5,900{600}	9,800{1,100}
3,450{350}	6,850{700}	785{80}	2,450{250}	4,900{500}	9,800{1,000}	980{100}	2,940{300}	6,850{700}	11,800{1,200}
3,450{350}	6,850{700}	785{80}	2,450{250}	4,900{500}	9,800{1,000}	980{100}	2,940{300}	6,850{700}	11,800{1,200}

Unit N {kgf}

series 72, 72B Low GL	72, 72B Normal GN	72, 72B Central GM	72, 72B Heavy GH	73, 73B Low GL	73, 73B Normal GN	73, 73B Central GM	73, 73B Heavy GH
29{3}	98{10}	196{20}	294{30}	49{5}	147{15}	294{30}	390{40}
29{3}	98{10}	294{30}	390{40}	49{5}	147{15}	390{40}	490{50}
78{8}	196{20}	490{50}	785{80}	98{10}	294{30}	590{60}	980{100}
98{10}	390{40}	880{90}	1,470{150}	147{15}	490{50}	980{100}	1,960{200}
147{15}	590{60}	980{100}	1,960{200}	196{20}	785{80}	1,470{150}	2,450{250}
196{20}	785{80}	1,470{150}	2,940{300}	294{30}	980{100}	2,450{250}	3,900{400}
294{30}	980{100}	2,450{250}	3,900{400}	390{40}	1,470{150}	3,450{350}	4,900{500}
490{50}	1,470{150}	2,940{300}	4,900{500}	590{60}	1,960{200}	3,900{400}	5,880{600}
490{50}	1,960{200}	3,900{400}	5,900{600}	590{60}	2,450{250}	4,900{500}	6,854{700}
490{50}	1,960{200}	3,900{400}	5,900{600}	590{60}	2,450{250}	4,900{500}	6,860{700}
590{60}	2,450{250}	4,900{500}	7,850{800}	685{70}	2,940{300}	5,900{600}	8,800{900}
590{60}	2,450{250}	4,900{500}	7,850{800}	685{70}	2,940{300}	5,900{600}	8,800{900}
590{60}	2,450{250}	4,900{500}	7,850{800}	685{70}	2,940{300}	5,900{600}	8,800{900}
785{80}	2,940{300}	5,900{600}	9,800{1,000}	880{90}	3,900{400}	7,850{800}	11,800{1,200}
785{80}	2,940{300}	5,900{600}	9,800{1,000}	880{90}	3,900{400}	7,850{800}	11,800{1,200}
880{90}	3,900{400}	7,850{800}	11,800{1,200}	980{100}	4,400{450}	8,800{900}	13,700{1,400}
880{90}	3,900{400}	7,850{800}	11,800{1,200}	980{100}	4,400{450}	8,800{900}	13,700{1,400}
880{90}	3,900{400}	7,850{800}	11,800{1,200}	980{100}	4,400{450}	8,800{900}	13,700{1,400}
980{100}	4,400{450}	8,800{900}	13,700{1,400}	1,470{150}	5,900{600}	11,800{1,200}	15,700{1,600}
980{100}	4,400{450}	8,800{900}	13,700{1,400}	1,470{150}	5,900{600}	11,800{1,200}	15,700{1,600}

9. Allowable Speed

As rotational speed of the bearing increase, the temperature of the bearing also rises due to heat produced inside the bearing by friction. This causes damage to the bearing such as seizure, and the bearing will be unable to continue stable operation. Therefore, the maximum speed at which it is possible for the bearing to continuously operate without the generation of excessive heat beyond specified limits, is called the **allowable speed** (min^{-1}).

The allowable speed of a bearing depends on the type of bearing, bearing dimensions, type of cage, load, lubricating conditions, and cooling conditions.

The bearing dimensions table gives approximate allowable rotational speeds for grease and oil lubrication. The values are based on the following:

- The bearing must have the proper internal clearance prescribed in the NTN Engineering standard design specifications and must be properly installed.
- A quality lubricant must be used. The lubricant must be replenished and changed when necessary.
- The bearing must be operated at normal operating temperature under ordinary load conditions ($P \leqq 0.09\ C_r$, $F_a / F_a \leqq 0.3$).

If load is $P \leqq 0.04\ C_{or}$, the rolling elements may not turn smoothly. If so, please contact NTN Engineering for more information. Allowable rotational speed for deep groove ball bearings with contact seal (LLU type) or low-torque seal (LLH type) is determined according to the circumferential speed of the seal.

For bearings to be used under heavier than normal load conditions, the allowable speed values listed in the bearing tables must be multiplied by an adjustment factor. The adjustment factors f_L and f_C are given in **Figs. 9.1** and **9.2**.

Also, when radial bearings are mounted on vertical shafts, lubricant retentions and cage guidance are not favorable compared to horizontal shaft mounting.

Therefore, the allowable speed should be reduced to **approximately 80% of the listed speed**.

For speeds other than those mentioned above, and for which data is incomplete, please consult NTN Engineering.

If rotational speed is to exceed allowable rotational speed given in the dimensions table, it will require special considerations such as using a bearing for which cage specifications, internal clearance and precision have been thoroughly checked. It will also require adopting forced circulation, jet oil or mist oil lubrication as the lubrication method.

Under such high speed operating conditions, when special care is taken, the standard allowable speeds given in the bearing tables can be adjusted upward. The maximum speed adjustment values, f_B, by which the bearing table speeds can be multiplied, are shown in **Table 9.1**. However, for any application requiring speeds in excess of the standard allowable speed, please consult NTN Engineering.

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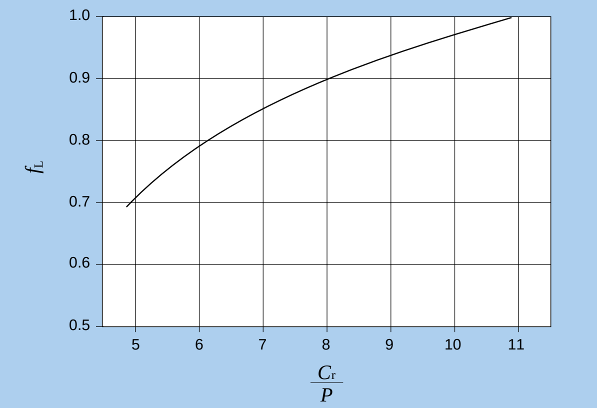

Fig. 9.1 Value of adjustment factor f_L depends on bearing load

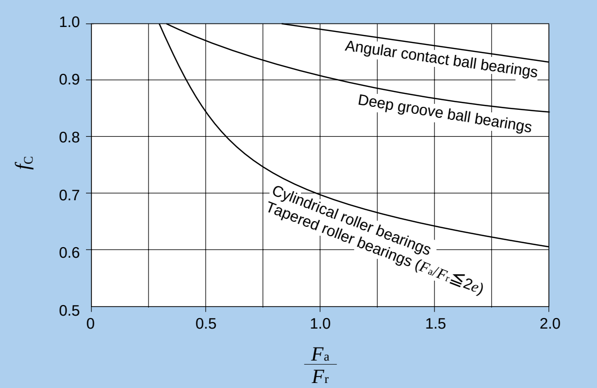

Fig. 9.2 Value of adjustment factor f_C depends on combined load

Table 9.1 Adjustment factor, f_B, for allowable number of revolutions

Type of bearing	Adjustment factor f_B
Deep groove ball bearings	3.0
Angular contact ball bearings	2.0
Cylindrical roller bearings	2.5
Tapered roller bearings	2.0

10. Friction and Temperature Rise

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10.1 Friction

One of the main functions required of a bearing is that it must have low friction. Under normal operating conditions rolling bearings have a much smaller friction coefficient than the slide bearings, especially starting friction.

The friction coefficient for rolling bearings is expressed by formula (10.1).

$$\mu = \frac{2M}{Pd} \quad \cdots\cdots (10.1)$$

where,

μ : Friction coefficient
M : Friction moment, N • mm {kgf • fmm}
P : Load, N {kgf}
d : Bearing bore diameter, mm

Although the dynamic friction coefficient for rolling bearings varies with the type of bearings, load, lubrication, speed, and other factors; for normal operating conditions, the approximate friction coefficients for various bearing types are listed in **Table 10.1**.

Table 10.1 Friction coefficient for bearings (reference)

Bearing type	Coefficient $\mu \times 10^{-3}$
Deep groove ball bearings	1.0～1.5
Angular contact ball bearings	1.2～1.8
Self-aligning ball bearings	0.8～1.2
Cylindrical roller bearings	1.0～1.5
Needle roller bearings	2.0～3.0
Tapered roller bearings	1.7～2.5
Spherical roller bearings	2.0～2.5
Thrust ball bearings	1.0～1.5
Thrust roller bearings	2.0～3.0

10.2 Temperature rise

Almost all friction loss in a bearing is transformed into heat within the bearing itself and causes the temperature of the bearing to rise. The amount of thermal generation caused by friction moment can be calculated using formula (10.2).

$$\left.\begin{aligned} Q &= 0.105 \times 10^{-6} Mn \ \text{N} \\ &= 1.03 \times 10^{-6} Mn \ \{\text{kgf}\} \end{aligned}\right\} \quad \cdots\cdots (10.2)$$

where,

Q : Thermal value, kW
M : Friction moment, N • mm {kgf • fmm}
n : Rotational speed, min^{-1}

Bearing operating temperature is determined by the equilibrium or balance between the amount of heat generated by the bearing and the amount of heat conducted away from the bearing. In most cases the temperature rises sharply during initial operation, then increases slowly until it reaches a stable condition and then remains constant. The time it takes to reach this stable state depends on the amount of heat produced, heat capacity/diffusion of the shaft and bearing housing, amount of lubricant and method of lubrication. If the temperature continues to rise and does not become constant, it must be assumed that there is some improper function.

Possible causes of abnormal temperature include bearing misalignment (due to moment load or incorrect installation), insufficient internal clearance, excessive preload, too much or too little lubricant, or heat produced from sealed units. Check the mechanical equipment, and if necessary, remove and inspect the bearing.

11. Lubrication

11.1 Purpose of lubrication

The purpose of bearing lubrication is to prevent direct metallic contact between the various rolling and sliding elements. This is accomplished through the formation of a thin oil (or grease) film on the contact surfaces. However, for rolling bearings, lubrication has the following advantages:

(1) **Reduction of friction and wear**
(2) **Dissipation of friction heat**
(3) **Prolonged bearing life**
(4) **Prevention of rust**
(5) **Protection against harmful elements**

In order to exhibit these effects, a lubrication method that matches service conditions. In addition to this, a quality lubricant must be selected, the proper amount of lubricant must be used and the bearing must be designed to prevent foreign matter from getting in or lubricant from leaking out.

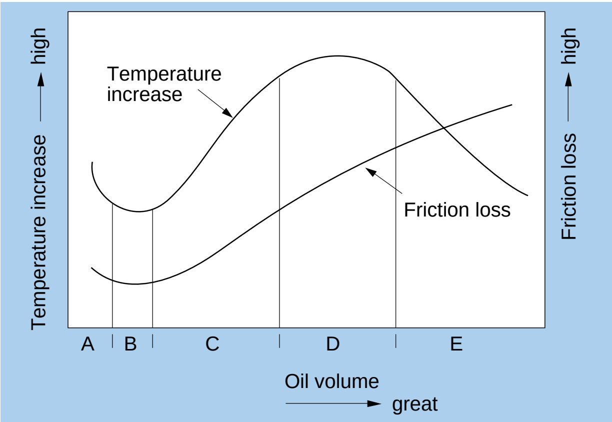

Fig. 11.1

Table 11.1 Oil volume, friction loss, bearing temperature (See Fig. 11.1)

Range	Characteristics	Lubrication method
A	When oil volume is extremely low, direct metallic contact occurs in places between the rolling elements and raceway surfaces. Bearing abrasion and seizing occur.	—
B	A thin oil film develops over all surfaces, friction is minimal and bearing temperature is low.	Grease lubrication, oil mist, air-oil lubrication
C	As oil volume increases, heat buildup is balanced by cooling.	Circulating lubrication
D	Regardless of oil volume, temperature increases at a fixed rate.	Circulating lubrication
E	As oil volume increases, cooling predominates and bearing temperature decreases.	Forced circulation lubrication, Oil jet lubrication

Fig. 11.1 shows the relationship between oil volume, friction loss, and bearing temperature. **Table 11.1** details the characteristics of this relationship.

11.2 Lubrication methods and characteristics

Lubrication method for bearings can be roughly divided into grease and oil lubrication. Each of these has its own features, so the lubrication method that best offers the required function must be selected.
The characteristic are shown in **Table 11.2**.

Table 11.2 Comparison of grease lubrication and oil lubrication characteristics

Concern \ Method	Grease lubrication	Oil lubrication
Handling	◎	△
Reliability	○	◎
Cooling effect	×	○ (Circulation necessary)
Seal structure	○	△
Power loss	○	○
Environment contamination	○	△
High speed rotation	×	○

◎ : Very good ○ : Good △ : Fair × : Poor

11.3 Grease lubrication

Grease lubricants are relatively easy to handle and require only the simplest sealing devices. For these reasons, grease is the most widely used lubricant for rolling bearings. It is used a bearing that is pre-sealed with grease (sealed/shield bearing), or if using an unsealed bearing, fill the bearing and housing with the proper amount of grease, and replenish or change the grease regularly.

11.3.1 Types and characteristics of grease

Lubricating grease are composed of either a mineral oil base or a synthetic oil base. To this base a thickener and other additives are added. The properties of all greases are mainly determined by the kind of base oil used and by the combination of thickening agent and various additives. **Table 11.5** shows general grease varieties and characteristics, and **Table 11.6** shows grease brand names and their natures. (See pages A-74 and A-75.) As performance characteristics of even the same type of grease will vary widely from brand to brand, **it is necessary to check the manufacturers' data when selecting a grease.**

(1) Base oil

Mineral oil or synthetics such as ester or ether oil are used as the base of the grease.

Mainly, the properties of any grease is determined by the properties of the base oil. Generally, greases with low

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viscosity base oil are best suited for low temperatures and high speeds; Grease using high-viscosity base oil has superior high-temperature and high-load characteristics.

(2) Thickening agents

Thickening agents are compounded with base oils to maintain the semi-solid state of the grease. Thickening agents consist of two types of bases, metallic soaps and non-soaps. Metallic soap thickeners include: lithium, sodium, calcium, etc.

Non-soap base thickeners are divided into two groups; inorganic (silica gel, bentonite, etc.) and organic (poly-urea, fluorocarbon, etc.).

The various special characteristics of a grease, such as limiting temperature range, mechanical stability, water resistance, etc. depend largely on the type of thickening agent used. For example, a sodium based grease is generally poor in water resistance properties, while greases with bentone, poly-urea and other non-metallic soaps as the thickening agent are generally superior in high temperature properties.

(3) Additives

Various additives are added to greases to improve various properties and efficiency. For example, there are anti-oxidents, high-pressure additives (EP additives), rust preventives, and anti-corrosives.

For bearings subject to heavy loads and/or shock loads, a grease containing high-pressure additives should be used. For comparatively high operating temperatures or in applications where the grease cannot be replenished for long periods, a grease with an oxidation stabilizer is best to use.

(4) Consistency

Consistency is an index that indicates hardness and fluidity of grease. The higher the number, the softer the grease is. The consistency of a grease is determined by the amount of thickening agent used and the viscosity of the base oil. For the lubrication of rolling bearings, greases with the NLGI consistency numbers of 1, 2, and 3 are used.

General relationships between consistency and application of grease are shown in **Table 11.3**.

(5) Mixing of greases

When greases of different kinds are mixed together, the consistency of the greases will change (usually softer), the operating temperature range will be lowered, and other changes in characteristics will occur. As a rule, grease should not be mixed with grease of any other brand.

However, if different greases must be mixed, at least greases with the same base oil and thickening agent should be selected.

Table 11.3 Consistency of grease

NLGI Consis-tency No.	JIS (ASTM) 60 times blend consistency	Applications
0	355～385	For centralized greasing use
1	310～340	For centralized greasing use
2	265～295	For general use and sealed bearing use
3	220～250	For general use and high temperature use
4	175～205	For special use

11.3.2 Amount of grease

The amount of grease used in any given situation will depend on many factors relating to the size and shape of the housing, space limitations, bearing's rotating speed and type of grease used.

As a rule of thumb, bearings should be filled to 30 to 40% of their space and housing should be filled 30 to 60%.

Where speeds are high and temperature rises need to be kept to a minimum, a reduced amount of grease should be used. **Excessive amount of grease cause temperature rise which in turn causes the grease to soften and may allow leakage. With excessive grease fills oxidation and deterioration may cause lubricating efficiency to be lowered.**

Moreover, the standard bearing space can be found by formula (11.1)

$$V = K \cdot W \quad \cdots\cdots (11.1)$$

where,
V : Quantity of bearing space open type (approx.), cm^3
K : Bearing space factor (see value of K in **Table 11.4**)
W : Mass of bearing, kg

Table 11.4 Bearing space factor K

Bearing type	Cage type	K
Ball bearings ❶	Pressed cage	61
NU-type cylindrical roller bearings ❷	Pressed cage Machined cage	50 36
N-type cylindrical roller bearings ❸	Pressed cage Machined cage	55 37
Tapered roller bearings	Pressed cage	46
Spherical roller bearings	Pressed cage Machined cage	35 28

❶ Does not apply top 160 series bearings.
❷ Does not apply to NU4 series bearings.
❸ Does not apply to N4 series bearings.

Table 11.5 Grease varieties and characteristics

Grease name	Lithium grease			Sodium grease (Fiber grease)	Calcium compound base grease
Thickener	Li soap			Na soap	Ca+Na soap Ca+Li soap
Base oil	Mineral oil	Diester oil	Silicone oil	Mineral oil	Mineral oil
Dropping point ˚C	170 ～ 190	170 ～ 190	200 ～ 250	150 ～ 180	150 ～ 180
Operating temperature range ˚C	-30 ～ +130	-50 ～ +130	-50 ～ +160	-20 ～ +130	-20 ～ +120
Mechanical stability	Excellent	Good	Good	Excellent ～ Good	Excellent ～ Good
Pressure resistance	Good	Good	Poor	Good	Excellent ～ Good
Water resistance	Good	Good	Good	Good ～ Poor	Good ～ Poor
Applications	Widest range of applications. Grease used in all types of rolling bearings.	Excellent low temperature and wear characteristics. Suitable for small sized and miniature bearings.	Suitable for high and low temperatures. Unsuitable for heavy load applications due to low oil film strength.	Some emulsification when water is introduced. Excellent characteristics at relatively high temperatures.	Excellent pressure resistance and mechanical stability. Suitable for bearings receiving shock loads.

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Table 11.6 Grease brands and their nature

Manufacturer	Brand name	NTN code	Thickener	Base oil
Showa Shell Sekiyu	Alvania Grease 2	2A	Lithium	Mineral oil
	Alvania Grease 3	3A	Lithium	Mineral oil
	Alvania Grease RA	4A	Lithium	Mineral oil
	Alvania EP Grease 2	8A	Lithium	Mineral oil
	Aero Shell Grease 7	5S	Microgel	Diester
Kyodo Yushi	Multemp PS No. 2	1K	Lithium	Diester
	Multemp SRL	5K	Lithium	Tetraesterdiester
	E5	L417	Urea	Ether
Esso Sekiyu	Temprex N3 / Unilex N3	2E	Complex Li	Synthetic hydrocarbon
	Beacon 325	3E	Lithium	Diester
NOK Kluber	Isoflex Super LDS18	6K	Lithium	Diester
	Barrierta JFE552	LX11	Fluoride	Fluoride oil
	Grease J	L353	Urea	Ester
Toray Dow Corning, Silicone	SH33L	3L	Lithium	Methyl phenyl oil
	SH44M	4M	Lithium	Methyl phenyl oil
Nippon Oil	Multi Nok wide No. 2	6N	Sodium lithium	Diester mineral oil
	U-4	L412	Urea	Synthetic hydrocarbon + dialkyldiphenyl ether
Nihon Grease	MP-1	L448	Diurea	PAO + ester
Idemitsu Kosan	Apolo Autolex A	5A	Lithium	Mineral oil
Mobil Sekiyu	Mobile Grease 28	9B	Bentone	Synthetic hydrocarbon
Cosmo Oil	Cosmo Wide Grease WR3	2M	Na terephthalate	Diester mineral oil
Daikin	Demnum L200	LX23	PTFE	Fluoride oil

Note: For nature, see the manufacturer's catalog.

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Aluminum grease	Non-soap base grease	
Al soap	Bentone, silica gel, urea, carbon black, fluorine compounds, etc.	
Mineral oil	Mineral oil	Synthetic oil
70 ～ 90	250 or above	250 or above
-10 ～ +80	-10 ～ +130	-50 ～ +200
Good ～ Poor	Good	Good
Good	Good	Good
Good	Good	Good
Excellent adhesion	Can be used in a wide range of low to high temperatures. Shows excellent heat resistance, cold resistance, chemical resistance, and other characteristics when matched with a suitable base oil and thickener.	
Suitable for bearings receiving vibration	Grease used in all types of rolling bearings.	

Base oil viscosity	Consistency	Dropping point˚C	Operating temperature˚C	Color	Characteristics
37.8˚C 140mm^2/s	273	181	−25～120	Amber	All-purpose grease
37.8˚C 140mm^2/s	232	183	−25～135	Amber	All-purpose grease
37.8˚C 45mm^2/s	252	183	−40～120	Amber	For low temperature
98.9˚C 15.3mm^2/s	276	187	−20～110	Brown	All-purpose extreme-pressure
98.9˚C 3.1mm^2/s	288	Min. 260	−73～149	Yellow-brown	MIL-G-23827
37.8˚C 15.3mm^2/s	265～295	190	−55～130	White	For low temperature and low torque
40˚C 26mm^2/s	250	192	−40～150	White	Wide range
40˚C 72.3mm^2/s	300	240	−30～180	White	For high temperature
40˚C 113mm^2/s	220～250	Min. 300	−30～160	Green	For high temperature
40˚C 11.5mm^2/s	265～295	177	−60～120	Brown	For low temperature and low torque
40˚C 16.0mm^2/s	265～295	Min. 180	−60～130	Yellow-green	For low temperature and low torque
40˚C 400mm^2/s	290	—	−35～250	White	
40˚C 75mm^2/s	—	280	−20～180	Gray-white	For high temperature
25˚C 100mm^2/s	300	200	−70～160	Light red-gray	For low temperature
40˚C 32mm^2/s	260	210	−40～180	Brown	For high temperature
37.8˚C 30.9mm^2/s	265～295	215	−40～135	Light brown	Wide range
40˚C 58mm^2/s	255	260	−40～180	Milk-white	For high temperature
40˚C 40.6mm^2/s	243	254	−40～150	Light brown	Wide range
37.8˚C 50mm^2/s	265～295	192	−25～150	Yellow	All-purpose grease
40˚C 28mm^2/s	315	Min. 260	−62～177	Red	MIL-G-81322C Wide range
37.8˚C 30.1mm^2/s	265～295	Min. 230	−40～150	Light brown	Wide range
40˚C 200mm^2/s	280	—	−60～300	White	

11.3.3 Grease replenishment

As the lubricating efficiency of grease declines with the passage of time, fresh grease must be re-supplied at proper intervals. The replenishment time interval depends on the type of bearing, dimensions, bearing's rotating speed, bearing temperature, and type of grease.

An easy reference chart for calculating grease replenishment intervals is shown in **Fig. 11.2**.

This chart indicates the replenishment interval for standard rolling bearing grease when used under normal operating conditions.

As operating temperatures increase, the grease re-supply interval should be shortened accordingly.

Generally, for every 10˚C increase in bearing temperature above 80˚C, the relubrication period is reduced by exponent "1/1.5".

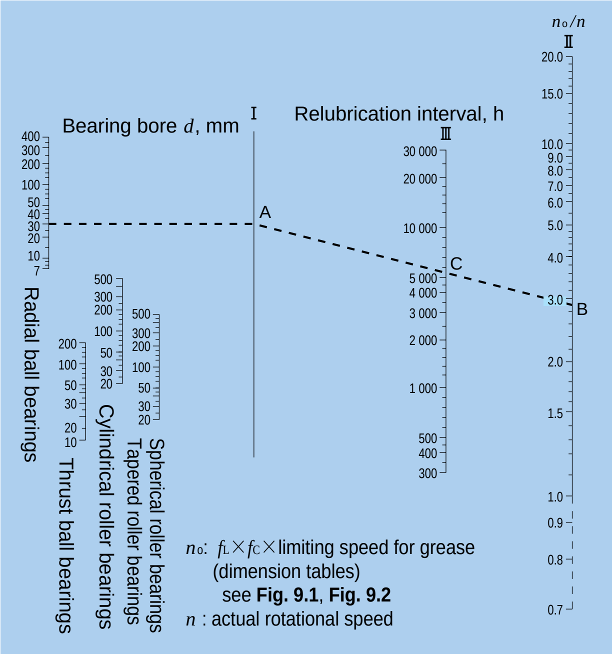

Fig. 11.2 Diagram for relubrication interval of greasing

(Example)
Find the grease relubrication time limit for deep groove ball bearing **6206**, with a radial load of 2.0 kN {204kgf} operating at 3,600 min^{-1}.

C_r / P_r = 19.5/2.0 kN = 9.8 from **Fig. 11.1**, the ajustment factor, f_L, is 0.96.

Allowable rotational speed from the dimensions tables for bearing 6206 is 11,000 min^{-1}. Allowable rotational speed n_o for a 2.0 kN {204 kgf} radial load is:

$$n_o = 0.96 \times 11{,}000 = 10{,}560 \text{ min}^{-1}$$

therefore, $\dfrac{n_o}{n} = \dfrac{10{,}560}{3{,}600} = 2.93$

The point where vertical line Ⅰ intersects a horizontal line drawn from the point equivalent of d = 30 for the radial ball bearing shown in **Fig. 11.2** shall be point A. Find intersection point C where vertical line Ⅱ intersects the straight line formed by joining point B (n_o/n = 2.93) with A with a straight line. It shows that grease life in this case is approximately 5,500 hours.

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11.4 Solid grease (For bearings with solid grease)

"Solid grease" is a lubricant composed mainly of lubricating grease and ultra-high polymer polyethylene. Solid grease has the same viscosity as grease at normal temperature, If heated once and cooled (this process is referred to as "calcination") the grease hardens while maintaining a large quantity of lubricant. The result of this solidification is that the grease does not easily leak from the bearing, even when the bearing is subjected to strong vibrations or centrifugal force.

Bearings with solid grease are available in two types: the spot-pack type in which solid grease is injected into the cage, and the full-pack type in which all empty space around the rolling elements is filled with solid grease.

Spot-pack solid grease is standard for deep groove ball bearings, small diameter ball bearings, and bearing units. Full-pack solid grease is standard for self-aligning ball bearings, spherical roller bearings, and needle roller bearings.

Primary advantages:

(1) Grease leakage is minimal.

(2) Low bearing torque with spot-pack type solid grease

For more details, please refer to NTN special catalog of **Solid grease bearings**.

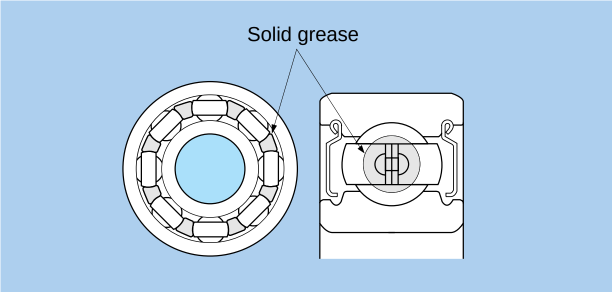


Fig. 11.3 Deep groove ball bearing with spot-pack solid grease (Z shield) (Standard for deep groove ball bearings)

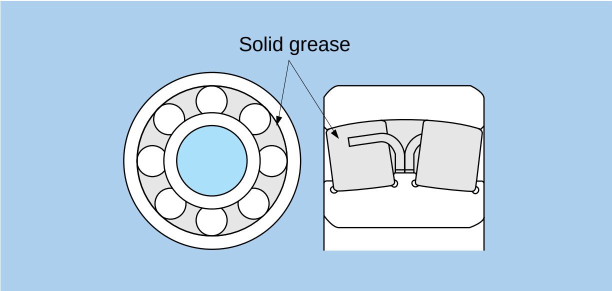
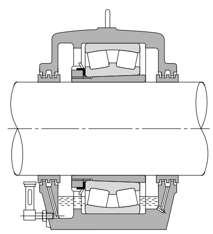
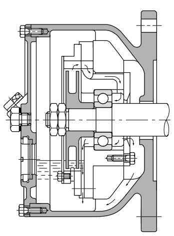
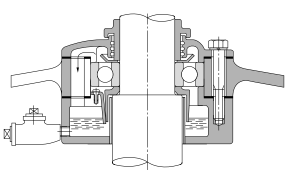
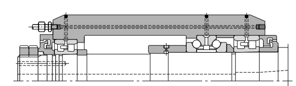
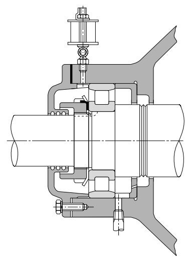
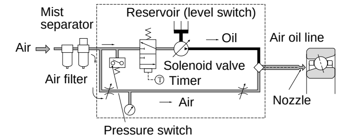
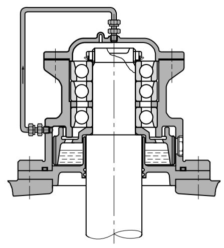
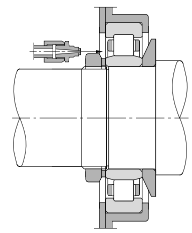


Fig. 11.4 Spherical roller bearing with full-pack solid grease (Standard for spherical roller bearings)

11.5 Oil lubrication

Oil lubrication is suitable for applications requiring that bearing-generated heat or heat applied to the bearing from other sources be carried away from the bearing and dissipated to the outside. **Table 11.7** shows the main methods of oil lubrication.

Table 11.7 Oil lubrication methods

Lubrication method	Example	Lubrication method	Example
(Oil bath lubrication) ● Oil bath lubrication is the most generally used method of lubrication and is widely used for low to moderate rotation speed applications. ● For horizontal shaft applications, oil level should be maintained at approximately the center of the lowest rolling element, according to the oil gauge, when the bearing is at rest. For vertical shafts at low speeds, oil level should be maintained at 50 - 80% submergence of the rolling elements.		**(Disc lubrication)** ● In this method, a partially submerged disc rotates and pulls oil up into a reservoir from which it then drains down through the bearing, lubricating it.	
(Oil spray lubrication) ● In this method, an impeller or similar device mounted on the shaft draws up oil and sprays it onto the bearing. This method can be used at considerably high speeds.		**(Oil mist lubrication)** ● Using pressurized air, lubricating oil is atomized before passing through the bearing. ● Due to the low lubricant resistance, this method is well suited to high speed applications.	
(Drip lubrication) ● In this method, oil is collected above the bearing and allowed to drip down into the housing where it becomes a lubricating mist as it strikes the rolling elements. Another version allows only slight amounts of oil to pass through the bearing. ● Used at relatively high speeds for light to moderate load applications. ● In most cases, oil volume is a few drops per minute.		**(Air-oil lubrication)** ● In this method, the required minimum amount of lubricating oil is measured and fed to each bearing at ideal intervals using compressed air. ● With fresh lubricating oil constantly being fed to the bearing, and with the cooling effect of the compressed air, bearing temperature rise can be minimized. ● Because the required oil quantity is infinitesimal, the working environment can be kept clean. Air-oil lubrication units are available from NTN.	
(Circulating lubrication) ● Used for bearing cooling applications or for automatic oil supply systems in which the oil supply is centrally located. ● One of the advantages of this method is that oil cooling devices and filters to maintain oil purity can be installed within the system. ● In order for oil to thoroughly lubricate the bearing, oil inlets and outlets must be provided on opposite sides of the bearing.		**(Oil jet lubrication)** ● This method lubricates by injecting oil under high pressure directly into the side of the bearing. This is a reliable system for high speed, high temperature or otherwise severe conditions. ● Used for lubricating the bearings in jet engines, gas turbines, and other high speed equipment. ● Under-race lubrication for machine tools is one example of this type of lubrication.	

11.5.1 Selection of lubricating oil

Under normal operating conditions, **spindle oil**, **machine oil**, **turbine oil**, and other mineral oils are widely used for the lubrication of rolling bearings. However, for temperatures **above 150**˚C or **below -30**˚C, synthetic oils such as **diester oil**, **silicone oil**, and **fluorocarbon oil** are used.

For lubricating oils, viscosity is one of the most important properties and determines an oil's lubricating efficiency. If viscosity is too low, formation of the oil film will be insufficient, and damage will occur to the raceways of the bearing. If viscosity is too high, viscous resistance will also be great and result in temperature increases and friction loss. In general, for higher speed applications a lower viscosity oil should be used; for heavier load applications, a higher viscosity oil should be used.

In regard to operating temperature, **Table 11.8** lists the required oil viscosity for different types of rolling bearings. **Fig. 11.5** is an oil viscosity - operating temperature comparison chart for the purpose of selecting a lubrication oil with viscosity characteristics appropriate to an application.

Table 11.9 lists the selection standards for lubricating oil viscosity with reference to bearing operating conditions.

Table 11.8 Required lubricating oil viscosity for bearings

Bearing type	Dynamic viscosity mm^2/s
Ball bearings, Cylindrical roller bearings, Needle roller bearings	13
Spherical roller bearings, Tapered roller bearings, Needle roller thrust bearings	20
Self-aligning roller thrust bearings	30

11.5.2 Oil quantity

In forced oil lubrication systems, the heat radiated away by the housing and surrounding parts plus the heat carried away by the lubricating oil is approximately equal to the amount of heat generated by the bearing and other sources.

For general housing applications, the required quantity of oil can be found by formula (11.2).

$$Q = K \cdot q \quad \cdots\cdots\cdots\cdots (11.2)$$

where,

Q: Quantity of oil for one bearing cm^3/min.
K: Allowable oil temperature rise factor (**Table 11.10**)
q: Amount of lubrication determined by diagram cm^3/min. (**Fig. 11.4**)

Because the amount of heat radiated will vary according to the type of housing, for actual operation it is advisable that the quantity of oil calculated by formula

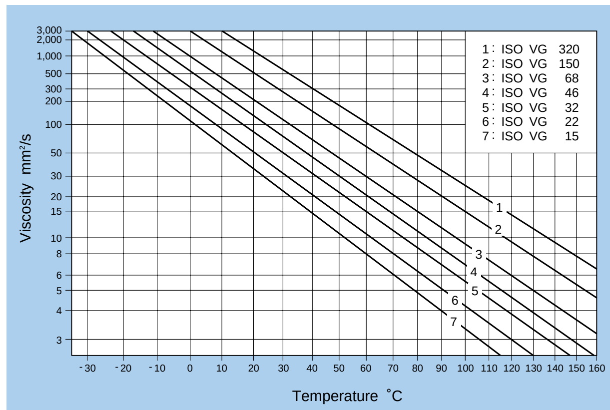


FIg. 11.5 Relation between lubricating oil viscosity and temperature

Table 11.8 Selection standards for lubricating oils (Reference)

Bearing operating temperature ˚C	d_n-value	Lubricating oil ISO viscosity grade (VG)		Suitable bearing
		Normal load	Heavy load or shock load	
−30〜 0	Up to allowable rotational speed	22，32	46	All types
0〜 60	Up to 15,000	46，68	100	All types
	15,000 〜80,000	32，46	68	All types
	80,000 〜150,000	22，32	32	All types but thrust ball bearings
	150,000〜500,000	10	22，32	Single row radial ball bearings, cylindrical roller bearings
60〜100	Up to 15,000	150	220	All types
	15,000 〜80,000	100	150	All types
	80,000 〜150,000	68	100，150	All types but thrust ball bearings
	150,000〜500,000	32	68	Single row radial ball bearings, cylindrical roller bearings
100 〜150	Up to allowable rotational speed	320		All types
0〜 60	Up to allowable rotational speed	46，68		Self-aligning roller bearings
60〜100	Up to allowable rotational speed	150		

Note 1: Applied when lubrication method is either oil bath or circulating lubrication.
2: Please consult NTN Engineering in cases where operating conditions fall outside the range covered by this table.

Table 11.9 Factor *K*

Expelled oil temp minus supplied oil temp ˚C	*K*
10	1.5
15	1
20	0.75
25	0.6

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(11.2) be multiplied by a factor or 1.5 or 2.0. Then, the amount of oil can be adjusted to correspond to actual operating conditions.

Furthermore, if it is assumed for calculation purposes that no heat is radiated by the housing, and that all bearing heat is removed by the oil, then the value for shaft diameter, $d = 0$.

(Example) For tapered roller bearing **30220U** mounted on a flywheel shaft with a radial load of 9.5 kN {969 kgf}, operating at 1,800 r/min, what is the amount of lubricating oil ‘ *Q* ’ required to keep the bearing temperature rise below 15˚C.

$d = 100$ mm,

$dn = 100 \times 1{,}800 = 18 \times 10^4$

From **Fig. 11.6** $q = 180\text{cm}^3$ / min

Assume the bearing temperature is approximately equal to the expelled oil temperature,

from **Table 11.10**, since $K = 1$

$Q = 1 \times 180 = 180\text{cm}^3$ / min

11.5.3 Relubrication intervals

The intervals at which lubricating oil should be changed varies depending upon operating conditions, oil quantity, and type of oil used. In general, for oil bath lubrication where the operating temperature is 50˚C or less, oil should be replaced once a year. When the operating temperature is between 80˚C – 100˚C, oil should be replaced at least every three months. For important equipment, it is advisable that lubricating efficiency and oil purity deterioration be checked regularly to determine when oil replacement is necessary.

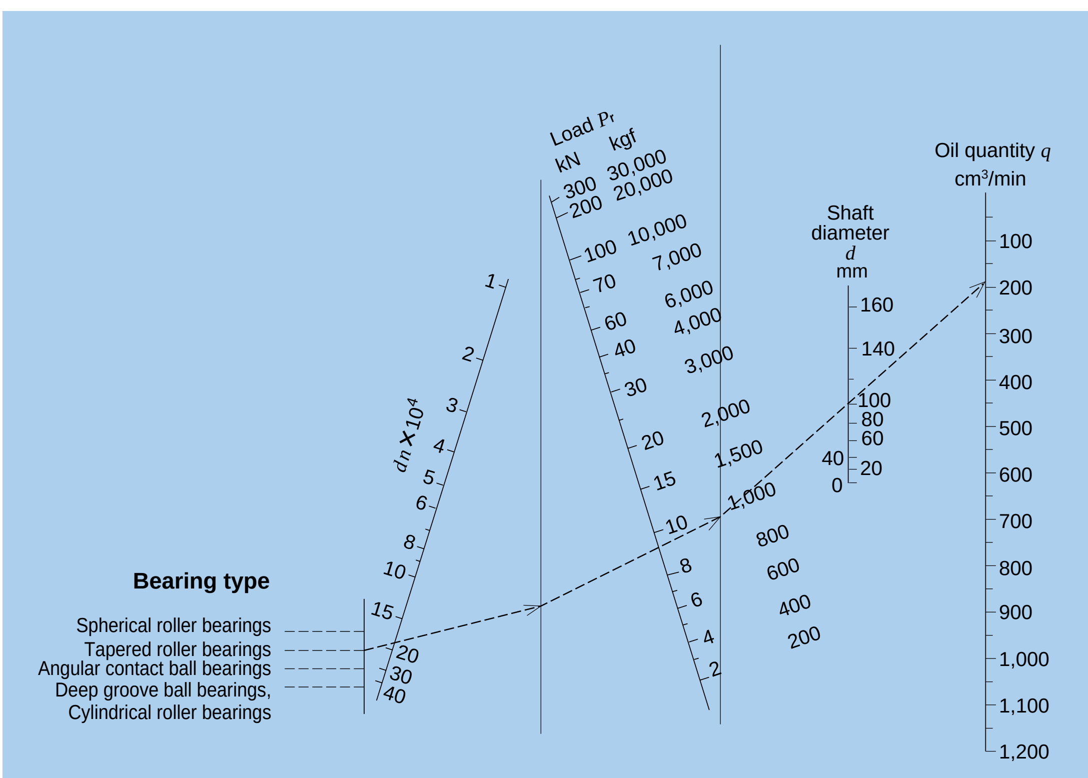


Fig. 11.6 Oil quantity guidelines

12. External bearing sealing devices

External seals have two main functions: to prevent lubricating oil from leaking out, and, to prevent dust, water, and other contaminants from entering the bearing. When selecting a seal, the following factors need to be taken into consideration: the type of lubricant (oil or grease), seal peripheral speed, shaft fitting errors, space limitations, seal friction and resultant heat increase, and cost.

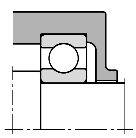
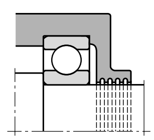
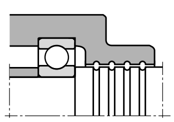
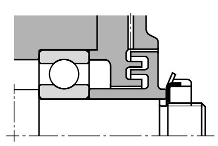
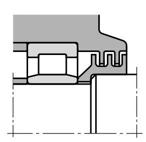
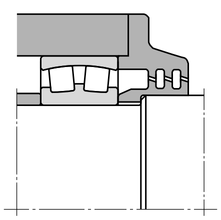
Sealing devices for rolling bearings fall into two main classifications: non-contact seals and contact seals.

- **Non-contact seals**: Non-contact seals utilize a small clearance between the shaft and the housing cover. Therefore friction is negligible, making them suitable for high speed applications.
  In order to improve sealing capability, clearance spaces are often filled with lubricant.

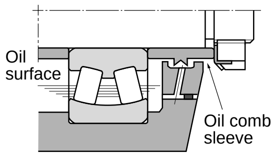
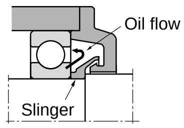
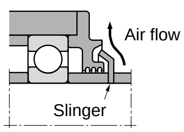
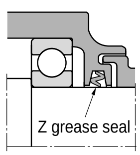
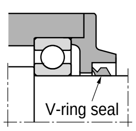
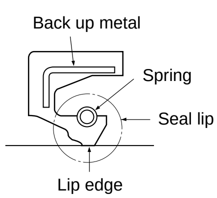
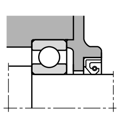
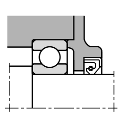
- **Contact seals**: A contact seal is a seal whereby a formed synthetic rubber lip on a steel plate is pressed against the shaft. Contact seals are generally far superior to non-contact seals in sealing efficiency, although their friction torque and temperature rise coefficients are higher. Furthermore, because the lip portion of a contact seal slides while in contact with the shaft, the allowable seal peripheral speed varies depending on seal type.

Lubrication is required in the place where the seal lip makes contact with the shaft. Ordinary bearing lubricant can also be used for this purpose.

The following chart lists the special characteristics of seals and other points to be considered when choosing an appropriate seal.

<table>
<tr><th>Type</th><th>Seal construction</th><th>Name</th><th colspan="2">Seal characteristics and selection considerations</th></tr>
<tr><td rowspan="6">Non-contact seals</td><td></td><td>Clearance seal</td><td>This is an extremely simple seal design with a small radial clearance.</td><td rowspan="3">Cautionary points regarding selection
• In order to improve sealing efficiency, clearances between the shaft and housing should be minimized. However, care should be taken to confirm shaft/bearing rigidity and other factors to avoid direct shaft-housing contact during operation.
Oil groove clearance (reference)<table><tr><th>Shaft diameter mm</th><th>Clearance mm</th></tr><tr><td>Up to 50</td><td>0.2∼0.4</td></tr><tr><td>50 or above</td><td>0.5∼1.0</td></tr></table>• Oil groove width, depth (reference)
width : 2∼5 mm
depth : 4∼5 mm
• Three or more oil grooves should be provided.
• Sealing efficiency can be further improved by filling the oil groove portion with grease of which the consistency grade is 150 to 200.
• Grease is generally used as the lubricant for labyrinth seals, and, except in low speed applications, is commonly used together with other sealing devices.</td></tr>
<tr><td></td><td>Oil groove seal
(oil grooves on housing side)</td><td>Several concentric oil grooves are provided on the housing inner diameter to greatly improve the sealing effect. When the grooves are filled with lubricant, the intrusion of contaminants from the outside is prevented.</td></tr>
<tr><td></td><td>Oil groove seal
(oil grooves on shaft and housing side)</td><td>Oil grooves are provided on both the shaft outer diameter and housing inner diameter for a seal with even greater sealing efficiency.</td></tr>
<tr><td></td><td>Axial labyrinth seal</td><td>This seal has a labyrinth passageway on the axial side of the housing.</td><td rowspan="3">Cautionary points regarding selection
• In order to improve sealing efficiency, labyrinth passageway clearances should be minimized. However, care should be taken to confirm shaft/bearing rigidity, fit, internal clearances and other factors to avoid direct contact between labyrinth projections during operation.
Labyrinth clearance (reference)<table><tr><th rowspan="2">Shaft diameter mm</th><th colspan="2">Clearance mm</th></tr><tr><th>Radial direction</th><th>Axial direction</th></tr><tr><td>−∼ 50</td><td>0.2∼0.4</td><td>1.0∼2.0</td></tr><tr><td>50∼200</td><td>0.5∼1.0</td><td>3.0∼5.0</td></tr></table>• Sealing efficiency can be further improved by filling the labyrinth passageway with grease of which the consistency grade is 150 to 200.
• Labyrinth seals are suitable for high speed applications.</td></tr>
<tr><td></td><td>Radial labyrinth seal</td><td>A labyrinth passageway is affixed to the radial side of the housing. For use with split housings. This offers better sealing efficiency than axial labyrinth seals.</td></tr>
<tr><td></td><td>Aligning labyrinth seal</td><td>The seal's labyrinth passageway is slanted and has sufficient clearance to prevent contact between the housing projections and the shaft even as the shaft realigns.</td></tr>
</table>

External bearing sealing devices

Type	Seal construction	Name	Seal characteristics and selection considerations
Non-contact seals		**Oil comb sleeve**	In this design, lubricating oil that makes its way out of the housing along the shaft is thrown off by projections on the oil comb sleeve and recirculated.
		**Slinger provided in the housing**	Seal type whereby a slinger is provided in the housing that prevents lubricant from leaking by centrifugal force produced by rotation.
		**Slinger provided outside the housing**	By mounting a slinger on the outside of the housing, centrifugal force helps to prevent dust and other solid contaminants from entering.
Contact seals		**Z grease seal**	In cross section resembling the letter "Z," this seal's empty spaces are filled with grease. The seal is commonly used with a plummer block (bearing housing).
		**V-ring seal**	This design enhances sealing efficiency with a lip that seals from the axial direction. With the aid of centrifugal force, this seal also offers effective protection against dust, water, and other contaminants entering the bearing. Can be used for both oil and grease lubrication. At seal peripheral speeds in excess of 12 m/s, seal ring fit is lost due to centrifugal force, and a clamping band is necessary to hold it in place.
	  For dust proof  For preventing lubricant leakage	**Oil seal**	Oil seals are widely used, and their shapes and dimensions are standardized under JIS B 2402. In this design, a ring-shaped spring is installed in the lip section. As a result, optimal contact pressure is exerted between the lip edge and shaft surface, and sealing efficiency is good. When the bearing and oil seal are in close proximity, the internal clearance of the bearing may be reduced by heat produced by the oil seal. In addition to considering the heat generated by contact seals at various peripheral speeds, internal bearing clearances must also be selected with caution. Depending on its orientation, the seal may function to prevent lubricant from leaking out or foreign matter from getting in.

Cautionary points regarding selection (non-contact seals)

- Seal type whereby a slinger that utilizes centrifugal force is provided on the rotating shaft.
- If mounted on the inside of the housing, the slinger should function to seal in lubricant by centrifugal force produced by rotation.
- If mounted on the outside of the housing, the slinger should function to seal out foreign matter by fan effect produced by rotation.
- These seal types are commonly employed together with other sealing devices.

Cautionary points regarding selection (oil seal)

Shaft surface roughness (reference)

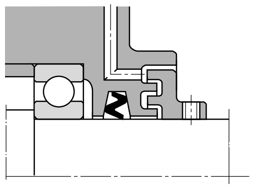
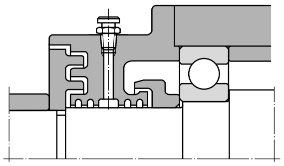
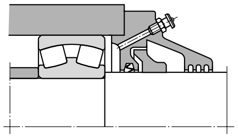
Peripheral speed m/s	Surface roughness Ra	Surface roughness Rmax
～ 5	0.8a	3.2s
5～10	0.4a	1.6s
10～	0.2a	0.8s

Shaft material (reference)

Material	Machine structural carbon steel, Low carbon alloy steel, Stainless steel
Surface hardness	HRC 40 or higher necessary HRC 55 or higher advisable
Processing method	Final grinding without repeat (moving), or buffed after hard chrome plating

Allowable speed/temperature according to seal type/material (reference)

Seal type/material		Allowable peripheral speed m/s $\left(V(m/s)=\frac{\pi \times d(mm) \times n(r/min)}{60{,}000}\right)$	Allowable temp °C
Oil seals	Nitrile rubber	16 or less	-25 ～+120
	Acrylic rubber	26 or less	-15 ～+150
	Fluorinated rubber	32 or less	-30 ～+200
Z-seal	Nitrile rubber	6 or less	-25 ～+120
V-ring	Nitrile rubber	40 or less	-25 ～+120

Type	Seal construction	Name	Seal characteristics and selection considerations
Combination seals		**Z-seal + Labyrinth seal**	This is an example of an axial labyrinth seal which has been combined with a Z-seal to increase its sealing efficiency. The axial labyrinth seal is affixed to the shaft with a setting bolt or other method. In the diagram on the left, both the direction of the Z-seal and the labyrinth seal are oriented to keep dust and other contaminants out of the bearing. Because a Z-seal has been incorporated, the allowable peripheral speed should not exceed 6 m/s.
		**Labyrinth seal + Oil groove seal + Slinger**	This is an example of a combination of three different non-contact seals. It has the advantage of preventing both lubricant leakage from inside the bearing and infiltration of dust and other contaminants from the outside. It is widely used on mining equipment and as a sealing system with plummer blocks in extremely dusty application conditions.
		**Oil groove seal + Slinger + Z-seal**	This is an example where an oil groove seal and slinger have been combined with a Z-seal to increase its sealing efficiency. In the diagram on the left, all three seals have been oriented to keep dust and other contaminants out of the bearing. The combination is widely used on mining equipment and as a sealing system with plummer blocks in extremely dusty application conditions.

II

13. Bearing Materials

13.1 Raceway and rolling element materials

While the contact surfaces of a bearing's raceways and rolling elements are subjected to repeated heavy stress, they still must maintain high precision and rotational accuracy. To accomplish this, the raceways and rolling elements must be made of a material that has high hardness, is resistant to rolling fatigue, is wear resistant, and has good dimensional stability. The most common cause of fatigue in bearings is the inclusion of non-metallic impurities in the steel. Non-metallic inclusion includes hard oxides that can cause fatigue crack. Clean steel with minimal non-metallic inclusion must therefore be used.

For all NTN bearings, steel low in oxygen content and non-metallic impurities, then refined by a vacuum degassing process as well as outside hearth smelting, is used. For bearings requiring especially high reliability and long life, steels of even higher in purity, such as vacuum melted steel (VIM, VAR) and electro-slag melted steel (ESR), are used.

1) High/mid carbon alloy steel

In general, steel varieties which can be hardened not just on the surface but also deep hardened by the so-called "through hardening method" are used for the raceways and rolling elements of bearings. Foremost among these is high carbon chromium bearing steel, which is widely used. For large type bearings and bearings with large cross sectional dimensions, induction hardened bearing steel incorporating manganese or molybdenum is used. Also in use is mid-carbon chromium steel incorporating silicone and manganese, which gives it hardening properties comparable to high carbon chromium steel.

Table 13.1 gives chemical composition of representative high carbon chrome bearing steel that meets JIS standards. SUJ2 is frequently used. SUJ3 with enhanced hardening characteristics containing a large quantity of Mn is used for large bearings. SUJ5 is SUJ3 to which Mo has been added to further enhance hardening characteristics, and is used for oversized bearings or bearings with thick walls.

The chemical composition of SUJ2 is equivalent to AISI 52100 (US) and DIN 100Cr6 (Germany).

2) Case hardened (carburizing) steel

Carburizing hardens the steel from the surface to the proper depth, forming a relatively soft core. This provides hardness and toughness, making the material suitable for impact loads. NTN uses case hardened steel for almost all of its tapered roller bearings. In terms of case hardened steel for NTN's other bearings, chromium steel and chrome molybdenum steel are used for small to medium sized bearings, and nickel chrome molybdenum steel is used for large sized bearings.

Table 13.2 gives the chemical composition of representative JIS case hardened steel.

3) Heat resistant bearing steel

When bearings made of ordinary high carbon chromium steel which have undergone standard heat treatment are used at temperatures above 120˚C for long durations, unacceptably large dimensional changes can occur. For this reason, a dimension stabilizing treatment (TS treatment) has been devised for very high temperature applications. This treatment however reduces hardness of the material, thereby reducing rolling fatigue life. (See item 3.3.2 on page A-18.)

For standard high temperature bearings used at temperatures from 150˚C – 200˚C, the addition of silicone to the steel improves heat resistance and results in a bearing with excellent rolling fatigue life with minimal dimensional change or softening at high temperatures.

A variety of heat resistant steels are also incorporated in bearings to minimize softening and dimensional changes when used at high temperatures. Two of these are high speed molybdenum steel and high speed tungsten steel. For bearings requiring heat resistance in high speed applications, there is also heat resistant case hardening molybdenum steel. (refer to **Table 13.3**)

4) Corrosion resistant bearing steel

For applications requiring high corrosion resistance, stainless steel is used. To achieve this corrosion resistance a large proportion of the alloying element chrome is added to martensite stainless steel. (**Table 13.4**)

5) Induction hardened steel

Besides the use of surface hardening steel, induction hardening is also utilized for bearing raceway surfaces, and for this purpose mid-carbon steel is used for its lower carbon content instead of through hardened steel. For induction hardening of the deep layers required for larger bearings and bearings with large surface dimensions, mid-carbon steel is fortified with chrome and molybdenum.

6) Other bearing materials

For ultra high speed applications and applications requiring very high level corrosion resistance, ceramic bearing materials such as Si_3N_4 are also available.

13.2 Cage materials

Bearing cage materials must have the strength to withstand rotational vibrations and shock loads. These materials must also have a low friction coefficient, be light weight, and be able to withstand bearing operation temperatures.

For small and medium sized bearings, pressed cages of cold or hot rolled steel with a low carbon content of approx. 0.1% are used. However, depending on the application, austenitic stainless steel is also used.

Machined cages are generally used for large bearings. Carbon steel for machine structures or high-strength cast brass is frequently used for the cages, but other materials such as aluminum alloy are also used.

Tables 13.5 and **13.6** give the chemical composition for these representative cage materials.

Besides high-strength brass, medium carbon nickel, chrome and molybdenum that has been hardened and tempered at high temperatures are also used for bearings used in aircraft. The materials are often plated with silver to enhance lubrication characteristics.

High polymer materials that can be injection molded are also widely used for cages. Polyamide resin reinforced with glass fibers is generally used. Cages made of high-polymer materials are lightweight and corrosion resistant. They also have superior damping and characteristics and lubrication performance. **Heat resistant polyimide resins now enable the production of cages that perform well in applications ranging between -40˚C – 120˚C.** However, they are not recommended for use at temperatures exceeding 120˚C.

Table 13.1 Chemical composition of representative high carbon chrome bearing steels

Standard	Symbol	Chemical composition (%)							Remarks
		C	Si	Mn	P	S	Cr	Mo	
JIS G 4805	SUJ2	0.95～1.10	0.15～0.35	Max. 0.50	Max. 0.025	Max. 0.025	1.30～1.60	Max. 0.08	
	SUJ3	0.95～1.10	0.40～0.70	0.90～1.15	Max. 0.025	Max. 0.025	0.90～1.20	Max. 0.08	
	SUJ5	0.95～1.10	0.40～0.70	0.90～1.15	Max. 0.025	Max. 0.025	0.90～1.20	0.10～0.25	
ASTM A295	52100	0.98～1.10	0.15～0.35	0.25～0.45	Max. 0.025	Max. 0.025	1.30～1.60	Max. 0.10	SUJ2 equivalent
ASTM A485	Grade 1	0.90～1.05	0.45～0.75	0.95～1.25	Max. 0.025	Max. 0.025	0.90～1.20	Max. 0.10	SUJ3 equivalent
	Grade 3	0.95～1.10	0.15～0.35	0.65～0.90	Max. 0.025	Max. 0.025	1.10～1.50	0.20～0.30	SUJ5 equivalent

Table 13.2 Chemical composition of representative case hardened steel (carburizing steel)

Standard	Symbol	Chemical composition (%)							
		C	Si	Mn	P	S	Ni	Cr	Mo
JIS G 4104	SCr420	0.18～0.23	0.15～0.35	0.60～0.85	Max. 0.030	Max. 0.030	—	0.90～1.20	—
JIS G 4105	SCM420	0.18～0.23	0.15～0.35	0.60～0.85	Max. 0.030	Max. 0.030	—	0.90～1.20	0.15～0.30
JIS G 4103	SNCM220	0.17～0.23	0.15～0.35	0.60～0.90	Max. 0.030	Max. 0.030	0.40～0.70	0.40～0.65	0.15～0.30
	SNCM420	0.17～0.23	0.15～0.35	0.40～0.70	Max. 0.030	Max. 0.030	1.60～2.00	0.40～0.65	0.15～0.30
	SNCM815	0.12～0.18	0.15～0.35	0.30～0.60	Max. 0.030	Max. 0.030	4.00～4.50	0.70～1.00	0.15～0.30
ASTM A534	5120	0.17～0.22	0.15～0.35	0.70～0.90	Max. 0.030	Max. 0.040	—	0.70～0.90	—
	4118	0.18～0.23	0.15～0.35	0.70～0.90	Max. 0.030	Max. 0.040	—	0.40～0.60	0.08～0.15
	8620	0.18～0.23	0.15～0.35	0.70～0.90	Max. 0.030	Max. 0.040	0.40～0.70	0.40～0.60	0.15～0.25
	4320	0.17～0.22	0.15～0.35	0.45～0.65	Max. 0.030	Max. 0.040	1.65～2.00	0.40～0.60	0.20～0.30
	9310	0.08～0.13	0.15～0.35	0.45～0.65	Max. 0.025	Max. 0.025	3.00～3.50	1.00～1.40	0.08～0.15

Table 13.3 Chemical composition of high-speed steel

Standard		Chemical composition (%)											
		C	Si	Mn	P	S	Cr	Mo	V	Ni	Cu	Co	W
AMS	6491 (M50)	0.77～0.85	Max. 0.25	Max. 0.35	Max. 0.015	Max. 0.015	3.75～4.25	4.00～4.50	0.90～1.10	Max. 0.15	Max. 0.10	Max. 0.25	Max. 0.25
	5626	0.65～0.80	0.20～0.40	0.20～0.40	Max. 0.030	Max. 0.030	3.75～4.50	Max. 1.00	0.90～1.30	—	—	—	17.25～18.25
	2315 (M50NiL)	0.11～0.15	0.10～0.25	0.15～035	Max. 0.015	Max. 0.010	4.00～4.25	4.00～4.50	1.13～1.33	3.20～3.60	Max. 0.10	Max. 0.25	Max. 0.25

Table 13.4 Chemical composition of stainless steel

Standard	Symbol	Chemical composition (%)						
		C	Si	Mn	P	S	Cr	Mo
JIS G 4303	SUS440C	0.95～1.20	Max. 1.00	Max. 1.00	Max. 0.040	Max. 0.030	16.00～18.00	Max. 0.75
AISI	440C	0.95～1.20	Max. 1.00	Max. 1.00	Max. 0.040	Max. 0.030	16.00～18.00	Max. 0.75

Table 13.5 Chemical composition of steel plate for pressed cages and carbon steel for machined cages

	Standard	Symbol	Chemical composition (%)						
			C	Si	Mn	P	S	Ni	Cr
Pressed retainer	JIS G 3141	SPCC	—	—	—	—	—	—	—
	JIS G 3131	SPHC	—	—	—	Max. 0.050	Max. 0.050	—	—
	BAS 361	SPB2	0.13～0.20	Max. 0.04	0.25～0.60	Max. 0.030	Max. 0.030	—	—
	JIS G 4305	SUS304	Max. 0.08	Max. 1.00	Max. 2.00	Max. 0.045	Max. 0.030	8.00～10.50	18.00～20.00
Machined retainer	JIS G 4051	S25C	0.22～0.28	0.15～0.35	0.30～0.60	Max. 0.030	Max. 0.035	—	—

Table 13.6 Chemical composition of high-strength cast brass for machined cages

Standard	Symbol	Chemical composition (%)							Impurities	
		Cu	Zn	Mn	Fe	Al	Sn	Ni	Pb	Si
JIS H 5120	CAC301	55.0～60.0	33.0～42.0	0.1～1.5	0.5～1.5	0.5～1.5	Max. 1.0	Max. 1.0	Max. 0.4	Max. 0.1

14. Shaft and Housing Design

Depending upon the design of a shaft or housing, the shaft may be influenced by an unbalanced load or other factors which can then cause large fluctuations in bearing efficiency. For this reason, it is necessary to pay attention to the following when designing shaft and housing:

1) Bearing arrangement selection; most effective fixing method for bearing arrangement
2) Selection of shoulder height and fillet radius of housing and shaft.
3) Shape precision and dimensions of fitting; area runout tolerance of shoulder.
4) Machining precision and mounting error of housing and shaft suitable for allowable alignment angle and inclination of bearing.

14.1 Fixing of bearings

When fixing a bearing in position on a shaft or housing, there are many instances where the interference fit alone is not enough to hold the bearing in place. Bearings must be fixed in place by various methods so that they do not move axially when placed under load.

Moreover, **even bearings which are not subjected to axial loads (such as cylindrical roller bearings, etc.), must be fixed in place axially because of the potential for ring displacement due to shaft deflection by moment load which may cause damage.**

Table 14.1 shows general bearing fixing methods, and **Table 14.2** shows fixing methods for bearings with tapered bores.

Table 14.1 General bearing fixing methods

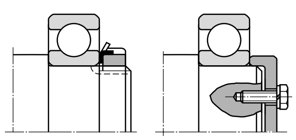
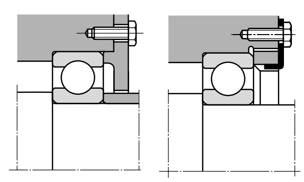
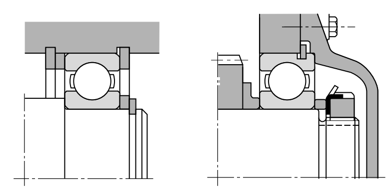
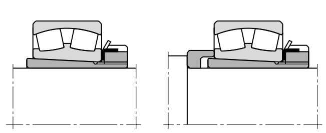
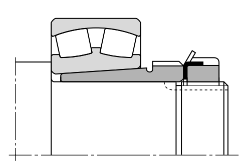
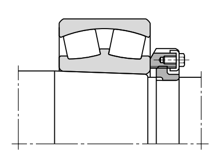
Inner ring clamp	Outer ring clamp	Snap ring
		
The most common method of fixing bearings in place is to use clamping nuts or bolts to hold the bearing or housing abutment against the ring end face.		Use of snap rings regulated under JIS B 2804, B 2805, and B 2806, makes construction very simple. However, interference with chamfers, bearing installation dimensions, and other related specifications must be considered carefully. Snap rings are not suitable for applications requiring high accuracy and where the snap ring receives large axial loads.

Table 14.2 Fixing methods for bearings with tapered bores

Adapter sleeve mounting	Withdrawal sleeve mounting	Split ring mounting
		
When installing bearings on cylindrical shafts, adapter sleeves or withdrawal sleeves can be used to fix bearings in place axially. The adapter sleeve is fastened in place by frictional force between the shaft and inner diameter of the sleeve.		For installation of tapered bore bearings directly on tapered shafts, the bearing is held in place by a split ring inserted into a groove on the shaft, and is fixed in place by a split ring nut or screw.

14.2 Bearing fitting dimensions

14.2.1 Abutment height and fillet radius

The shaft and housing abutment height (h) should be larger than the bearings' maximum allowable chamfer dimensions ($r_{s\ max}$), and the abutment should be designed so that it directly contacts the flat part of the bearing end face. The fillet radius (r_a) must be smaller than the bearing's minimum allowable chamfer dimension ($r_{s\ min}$) so that it does not interfere with bearing seating. **Table 14.3** lists abutment height (h) and fillet radius (r_a).

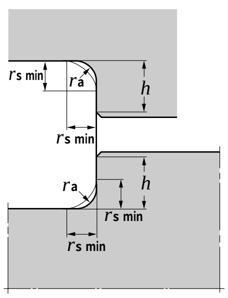
For bearings to be applied to very large axial loads as well, shaft abutments (h) should be higher than the values in the table.

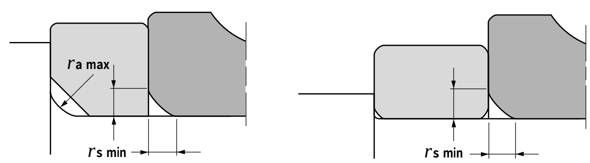
14.2.2 For spacer and ground undercut

In cases where a fillet radius ($r_{a\ max}$) larger than the bearing chamfer dimension is required to strengthen the shaft or to relieve stress concentration (**Fig. 14.1a**), or where the shaft abutment height is too low to afford adequate contact surface with the bearing (**Fig. 14.1b**), spacers may be used effectively.

Relief dimensions for ground shaft and housing fitting surfaces are given in **Table 14.4**.

II





(a) (b)

Fig. 14.1 Bearing mounting with spacer

Table 14.3 Fillet radius and abutment height Unit mm

$r_{s\ min}$	$r_{as\ max}$	h (min)	
		Normal use❶	Special use❷
0.05	0.05	0.3	
0.08	0.08	0.3	
0.1	0.1	0.4	
0.15	0.15	0.6	
0.2	0.2	0.8	
0.3	0.3	1.25	1
0.6	0.6	2.25	2
1	1	2.75	2.5
1.1	1	3.5	3.25
1.5	1.5	4.25	4
2	2	5	4.5
2.1	2	6	5.5
2.5	2	6	5.5
3	2.5	7	6.5
4	3	9	8
5	4	11	10
6	5	14	12
7.5	6	18	16
9.5	8	22	20
12	10	27	24
15	12	32	29
19	15	42	38

❶ If bearing supports large axial load, the height of the shoulder must exceed the value given here.
❷ Used when axial load is light. These values are not suitable for tapered roller bearings, angular ball bearings and spherical roller bearings.

Note: $r_{as\ max}$ maximum allowable fillet radius.

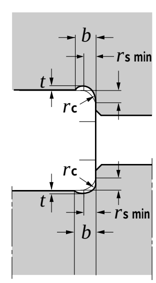


Table 14.4 Relief dimensions for ground shaft

Unit mm

$r_{s\ min}$	Relief dimensions b	t	r_c
1	2	0.2	1.3
1.1	2.4	0.3	1.5
1.5	3.2	0.4	2
2	4	0.5	2.5
2.1	4	0.5	2.5
2.5	4	0.5	2.5
3	4.7	0.5	3
4	5.9	0.5	4
5	7.4	0.6	5
6	8.6	0.6	6
7.5	10	0.6	7

II

14.2.3 Thrust bearings and fitting dimensions

For thrust bearings, it is necessary to make the raceway disc back face sufficiently broad in relation to load and rigidity, and fitting dimensions from the dimension tables should be adopted. (**Figs. 14.2** and **14.3**)

For this reason, shaft and abutment heights will be larger than for radial bearings. (Refer to dimension tables for all thrust bearing fitting dimensions.)

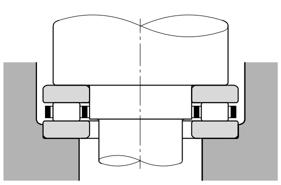


Fig. 14.2

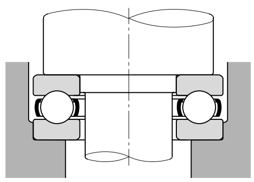


Fig. 14.3

14.3 Shaft and housing accuracy

Table 14.5 shows the accuracies for shaft and housing fitting surface dimensions and configurations, as well as fitting surface roughness and abutment squareness for normal operating conditions.

Table 14.5 Shaft and housing accuracy

Characteristics		Shaft	Housing
Dimensional accuracy		IT6 (IT5)	IT7 (IT5)
Roundness (max.) Cylindricity		IT3	IT4
Abutment squareness		IT3	IT3
Fitting surface roughness	Small size bearings	0.8a	1.6a
	Mid-large size bearings	1.6a	3.2a

Note: For precision bearings (P4, P5 accuracy), it is necessary to increase the circularity and cylindricity accuracies in this table by approximately 50%. For more specific information, please consult the NTN precision rolling bearing catalog.

14.4 Allowable bearing misalignment

A certain amount of misalignment of a bearing's inner and outer rings occurs as a result of shaft flexure, shaft or housing finishing irregularities, and minor installation error. In situations where the degree of misalignment is liable to be relatively large, self-aligning ball bearings, spherical roller bearings, bearing units and other bearings with aligning properties are advisable. Although allowable misalignment will vary according to bearing type, load conditions, internal clearances, etc., **Table 14.6** lists some general misalignment standards for normal applications. In order to avoid shorter bearing life and cage failure, it is necessary to maintain levels of misalignment below these standard levels.

Table 14.6 Bearing type and allowable misalignment/alignment allowance

Allowable misalignment	
Deep groove ball bearings	1/1,000～1/300
Angular contact ball bearings	
Single row	1/1,000
Multi row	1/10,000
back to back arrangement	1/10,000
Face to face arrangement	1/1,000
Cylindrical roller bearings	
Bearing series 2, 3, 4	1/1,000
Bearing series 22, 23, 49, 30	1/2,000
Tapered roller bearings	
Single row/back to back arrangement	1/2,000
Face-to-face arrangement	1/1,000
Needle roller bearings	1/2,000
Thrust bearings (excluding self-aligning roller thrust bearings)	1/10,000
Alignment allowance	
Self-aligning ball bearings	1/20～1/15
Spherical roller bearings	1/50～1/30
Self-aligning roller thrust bearings	1/30
Ball bearing units	
Without cover	1/30
With cover	1/50

15. Bearing Handling

Bearings are precision parts and, in order to preserve their accuracy and reliability, care must be exercised in their handling.
In particular, bearing cleanliness must be maintained, sharp impacts avoided, and rust prevented.

15.1 Bearing storage

Most rolling bearings are coated with a rust prevent oil before being packed and shipped, and they should be stored at room temperature with a relative humidity of less than 60%.

15.2 Installation

When bearings are being installed on shafts or in housings, the bearing rings should never be struck directly with a hammer or a drift, as shown in **Fig. 15.1**, because damage to the bearing may result. **Any force applied to the bearing should always be evenly distributed over the entire bearing ring face.** Also, when fitting both rings simultaneously, applying pressure to one ring only, as shown in **Fig. 15.2**, should be avoided because indentations in the raceway surface may be caused by the rolling elements, or other internal damage may result.

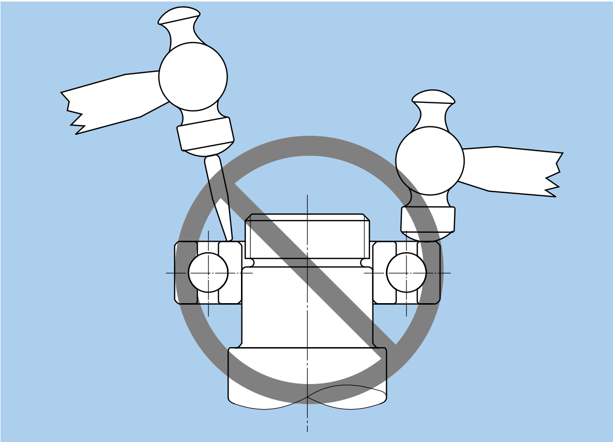


Fig. 15.1

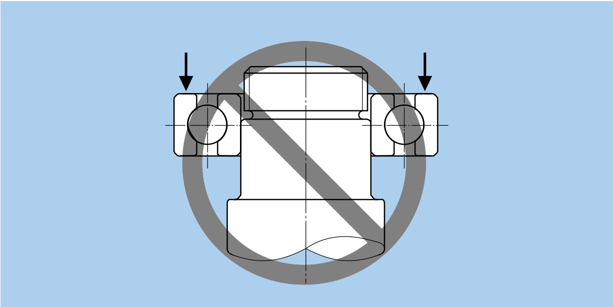


Fig. 15.2

15.2.1 Installation preparations

Bearings should be fitted in a clean, dry work area. Especially for small and miniature bearings, a "clean room" should be provided as any contamination particles in the bearing will greatly affect bearing efficiency.

All dirt, burrs or metal filings must be removed from the shaft, housing and tools used for mounting the bearings. Shaft and housing fitting surfaces should also be checked for roughness, dimensional and design accuracy, and to ensure that they are within allowable tolerance limits.

Bearings should not be unwrapped until just prior to installation. Normally, bearings to be used with grease lubricant can be installed as is, without removing the rust prevent oil. However, for bearings which will use oil lubricant, or in cases where mixing the grease and rust prevent oil would result in loss of lubrication efficiency, the rust prevent oil should be removed by washing with benzene or petroleum solvent and dried before installation. Bearings should also be washed and dried before installation if the package has been damaged or there are other chances that the bearings have been contaminated. **Double shielded bearings and sealed bearings should never be washed.**

15.2.2 Installing cylindrical bore bearings

For bearings with relatively small interference, the entire circumference of the raceway can be uniformly press-fit at room temperature as shown in **Fig. 15.3**. Usually, bearings are installed by striking the sleeve with a hammer; however, when installing a large number of bearings, a mechanical or hydraulic press should be used.

When installing non-separable bearings on a shaft and in a housing simultaneously, a pad which distributes the fitting pressure evenly over the inner and outer rings is used as shown in **Fig. 15.4**. If the fitting is too tight or bearing size is large, a considerable amount of force is required to install the bearing at room temperature. Installation can be facilitated by heating and expanding the inner ring beforehand. The required relative temperature difference between the inner ring and the shaft depends on the amount of interference and the shaft fitting surface diameter. **FIg. 15.5** shows the relation between the bearing inner bore diameter temperature differential and the amount of thermal expansion. **In any**

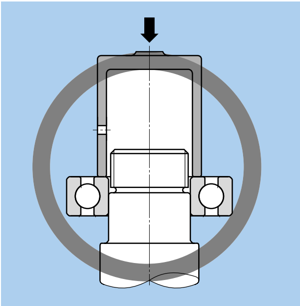


Fig. 15.3 **Fitting sleeve pressure against inner ring**

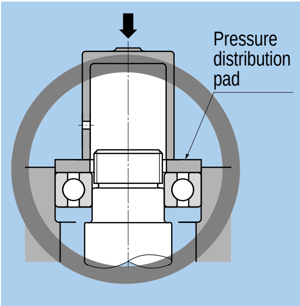


Fig. 15.4 **Fitting sleeve pressure against inner /outer ring simultaneously**

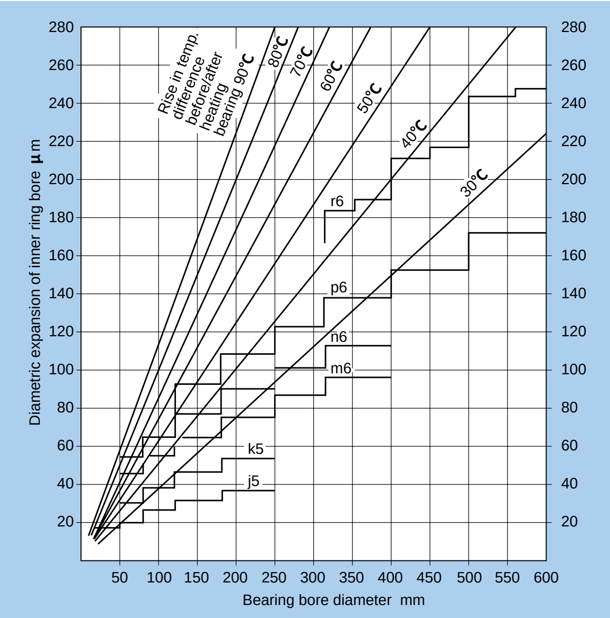


Fig. 15.5 Temperature required for heat-fitting inner ring

event, bearings should never be heated above 120˚C.

The most commonly used method of heating bearings is to immerse them in hot oil. This method must not be used for sealed bearings or shield bearings with grease sealed inside.

To avoid overheating parts of the bearings they should never be brought into direct contact with the heat source, but instead should be suspended inside the heating tank or placed on a wire grid.

If heating the bearing with air in a device such as a thermostatic chamber, the bearing can be handled while dry.

For heating the inner rings of NU, NJ or NUP cylindrical and similar type bearings without any ribs or with only a single rib, an induction heater can be used to quickly heat bearings in a dry state **(must demagnetize).**

When heated bearings are installed on shafts, the inner rings must be held against the shaft abutment until the bearing has been cooled in order to prevent clearance between the ring and the abutment face.

As shown in **Fig. 15.6**, a removal pawl, or tool, can also be used to dismount the inner ring when using the induction heating method described above.

15.2.3 Installation of tapered bore bearings

Small type bearings with tapered bores are installed over a tapered shaft, withdrawal sleeves, or adapter sleeves by driving the bearing into place using a locknut. The locknut is tightened using a hammer or impact wrench. (**Fig. 15.7**)

Large size bearings require considerable fitting force and must be installed hydraulically.

In **Fig. 15.8** the fitting surface friction and nut tightening torque needed to install bearings with tapered bores directly onto tapered shafts are decreased by injecting

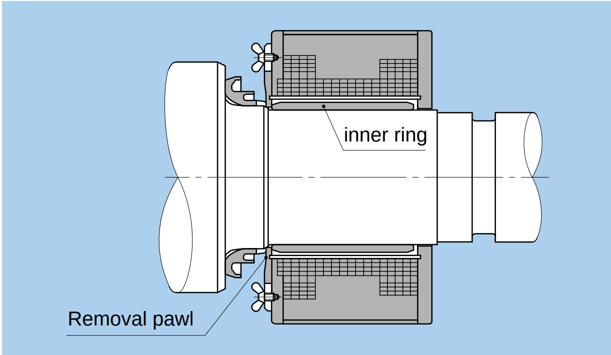
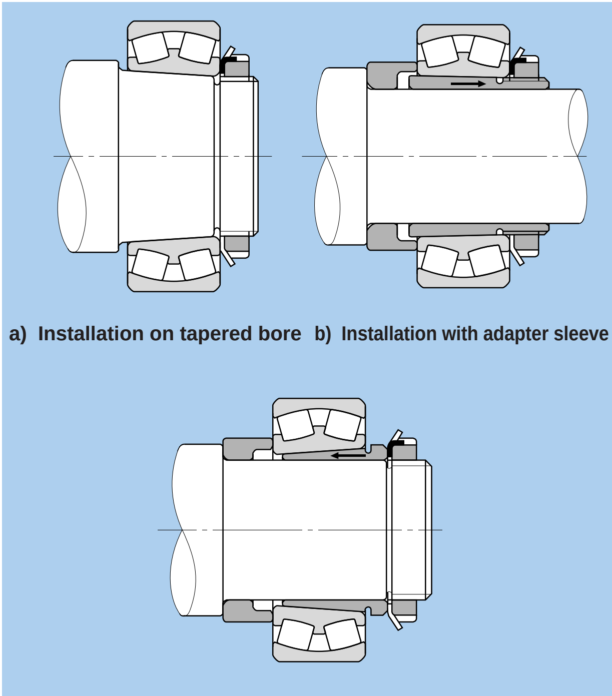


Fig. 15.6 Removal of inner ring using an induction heater



a) Installation on tapered bore b) Installation with adapter sleeve

c) Installation using withdrawal sleeve

Fig. 15.7 Installation methods using locknuts

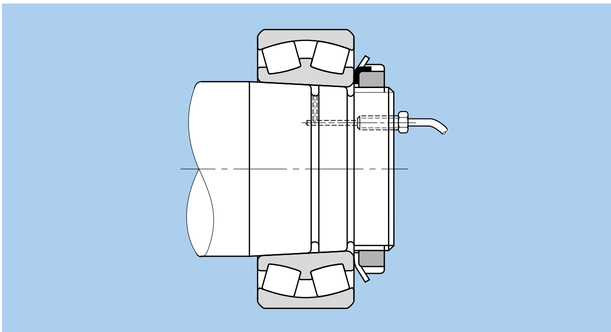
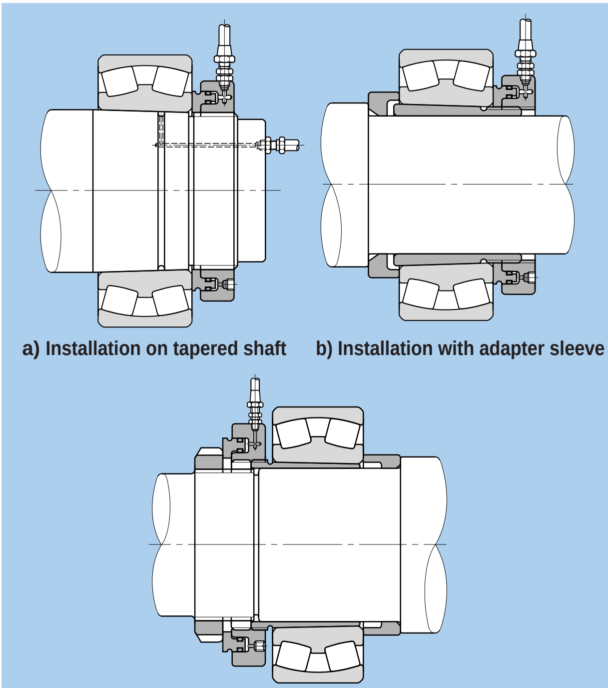


Fig. 15.8 Installation utilizing oil injection



a) Installation on tapered shaft b) Installation with adapter sleeve

c) Installation with withdrawal sleeve

Fig. 15.9 Installation using hydraulic nut

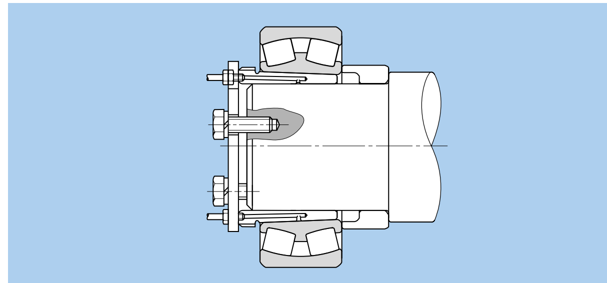


Fig. 15.10 Installation using hydraulic withdrawal sleeve

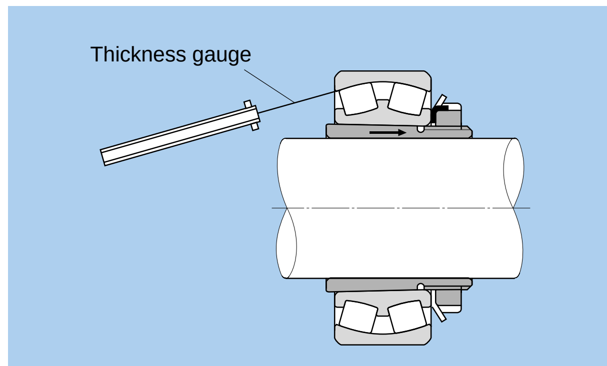


Fig. 15.11 Internal clearance measurement method for spherical roller bearings

high pressure oil between the fitting surfaces.

Fig. 15.9 a) shows one method of installation where a hydraulic nut is used to drive the bearing onto a tapered shaft.

Fig. 15.9 b) and **c**) show installation using a hydraulic nut with adapter sleeves and withdrawal sleeves.

Fig. 15.10 shows an installation method using a hydraulic withdrawal sleeve.

With tapered bore bearings, as the inner ring is driven axially onto the shaft or adapter or withdrawal sleeve, the interference will increase and the bearing internal radial clearance will decrease. Interference can be estimated by measuring decrease in internal radial clearance. As shown in **Fig. 15.11**, the internal radial clearance between the rollers and outer ring of spherical roller bearings should be measured with a thickness gauge under no load while the rollers are held in the correct position. Instead of using the decrease in amount of internal radial clearance to estimate the interference, it is possible to estimate by measuring the distance the bearing has been driven onto the shaft.

For spherical roller bearings, **Table 15.1** indicates the appropriate interference which will be achieved as a result of the internal radial clearance decrease, or the distance the bearing has been driven onto the shaft.

For conditions such as heavy loads, high speeds, or when there is a large temperature differential between the inner and outer rings, etc. which require large interference fits, bearings which have a minimum internal radial clearance of C3 or greater should be used. **Table 15.1** lists the maximum values for internal radial clearance decrease and axial displacement. For these applications, the remaining clearance must be greater than the minimum allowable residual clearance listed in **Table 15.1**.

15.2.4 Installation of outer ring

Even for tight interference fits, the outer rings of small type bearings can be installed by driving them into housings at room temperature. For large interference type bearings, the housing can be heated before installing the bearing, or the bearing's outer ring can be cooled with dry ice, etc. before installing. If dry ice or other cooling agent is used, atmospheric moisture will condense on bearing surfaces, and therefore appropriate rust preventative measures are necessary.

15.3 Internal clearance adjustment

As shown in **Fig. 15.12**, for angular contact ball bearings and tapered roller bearings the desired amount of axial internal clearance can be set at the time of installation by tightening or loosening the adjustment nut.

To adjust the suitable axial internal clearance or amount of bearing preload, the internal clearance can be measured while tightening the adjusting nut as shown in **Fig. 15.13**. Other methods are to check rotation torque by rotating the shaft or housing while adjusting the nut, or to insert shims of the proper thickness as shown in **Fig. 15.14**.

II

Table 15.1 Installation of tapered bore spherical roller bearings

Units mm

Nominal bearing bore diameter *d*		Reduction of radial internal clearance		Axial displacement drive up				Minimum allowable residual clearance		
				Taper, 1:12		Taper, 1:30				
over	incl.	Min	Max	Min	Max	Min	Max	CN	C3	C4
30	40	0.02	0.025	0.35	0.4	—	—	0.015	0.025	0.04
40	50	0.025	0.03	0.4	0.45	—	—	0.02	0.03	0.05
50	65	0.03	0.035	0.45	0.6	—	—	0.025	0.035	0.055
65	80	0.04	0.045	0.6	0.7	—	—	0.025	0.04	0.07
80	100	0.045	0.055	0.7	0.8	1.75	2.25	0.035	0.05	0.08
100	120	0.05	0.06	0.75	0.9	1.9	2.25	0.05	0.065	0.1
120	140	0.065	0.075	1.1	1.2	2.75	3	0.055	0.08	0.11
140	160	0.075	0.09	1.2	1.4	3	3.75	0.055	0.09	0.13
160	180	0.08	0.1	1.3	1.6	3.25	4	0.06	0.1	0.15
180	200	0.09	0.11	1.4	1.7	3.5	4.25	0.07	0.1	0.16
200	225	0.1	0.12	1.6	1.9	4	4.75	0.08	0.12	0.18
225	250	0.11	0.13	1.7	2	4.25	5	0.09	0.13	0.2
250	280	0.12	0.15	1.9	2.4	4.75	6	0.1	0.14	0.22
280	315	0.13	0.16	2	2.5	5	6.25	0.11	0.15	0.24
315	355	0.15	0.18	2.4	2.8	6	7	0.12	0.17	0.26
355	400	0.17	0.21	2.6	3.3	6.5	8.25	0.13	0.19	0.29
400	450	0.2	0.24	3.1	3.7	7.75	9.25	0.13	0.2	0.31
450	500	0.21	0.26	3.3	4	8.25	10	0.16	0.23	0.35
500	560	0.24	0.3	3.7	4.6	9.25	11.5	0.17	0.25	0.36
560	630	0.26	0.33	4	5.1	10	12.5	0.2	0.29	0.41
630	710	0.3	0.37	4.6	5.7	11.5	14.5	0.21	0.31	0.45
710	800	0.34	0.43	5.3	6.7	13.3	16.5	0.23	0.35	0.51
800	900	0.37	0.47	5.7	7.3	14.3	18.5	0.27	0.39	0.57
900	1,000	0.41	0.53	6.3	8.2	15.8	20.5	0.3	0.43	0.64
1,000	1,120	0.45	0.58	6.8	8.7	17	22.5	0.32	0.48	0.7
1,120	1,250	0.49	0.63	7.4	9.4	18.5	24.5	0.34	0.54	0.77

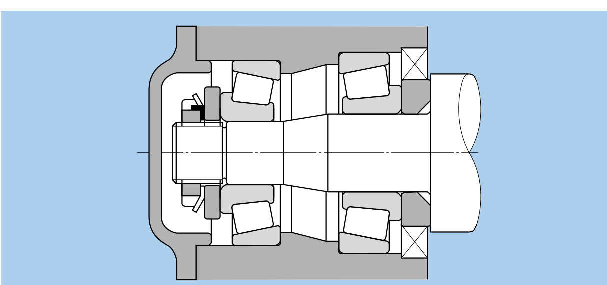


Fig. 15.12 Axial internal clearance adjustment

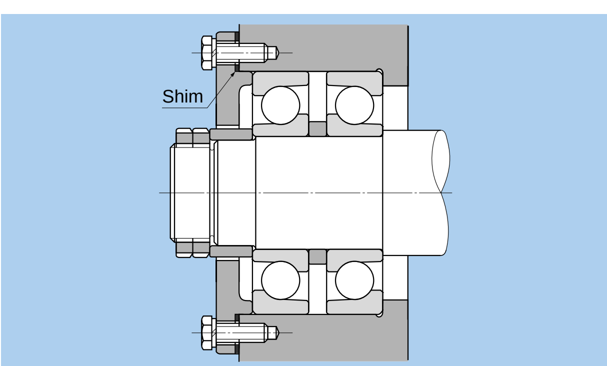


Fig. 15.14 Internal clearance adjustment using shims

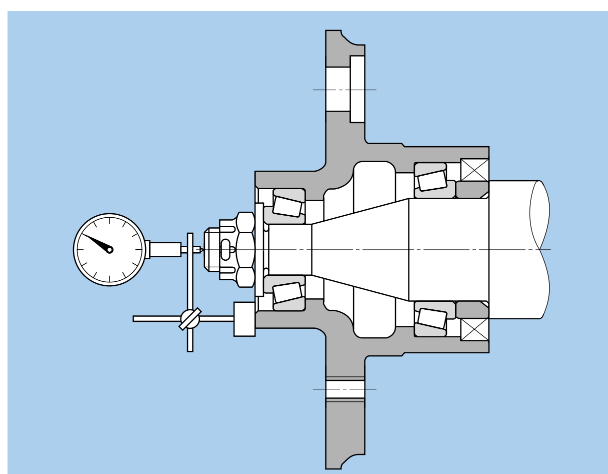


Fig. 15.13 Measurement of axial internal clearance adjustment

15.4 Post installation running test

To insure that the bearing has been properly installed, a running test is performed after installation is completed. The shaft or housing is first rotated by hand and if no problems are observed low speed, no load power test is performed. If no abnormalities are observed, **the load and speed are gradually increased to operating conditions**. **During the test if any unusual noise, vibration, or temperature rise is observed the test should be stopped and examine the equipment. If necessary, the bearing should be disassembled for inspection.**

To check bearing running noise, the sound can be amplified and the type of noise ascertained with a listening instrument placed against the housing. A clear, smooth and continuous running sound is normal. A high, metallic or irregular sound indicates some error in function. Vibration can be accurately checked with a vibration measuring instrument, and the amplitude and frequency characteristics measured guantitatively.

Usually the bearing temperature can be estimated from the housing surface temperature. However, if the bearing outer ring is accessible through oil inlets, etc., the temperature can be more accurately measured.

Under normal conditions, bearing temperature rises with operation time and then reaches a stable operating temperature after a certain period of time. If the temperature does not stable and continues to rise, or if there is a sudden temperature rise, or if the temperature is extremely high, the bearing should be inspected.

15.5 Bearing disassembly

Bearings are often removed as part of periodic inspection procedures or during the replacement of other parts. However, the shaft and housing are almost always reinstalled, and in more than a few cases the bearings themselves are reused. These bearings, shafts, housings, and other related parts must be designed to prevent damage during disassembly procedures, and the proper disassembly tools must be employed. When removing raceways with interference, pulling force should be applied to the raceway only. **Do not remove the raceway through the rolling elements.**

15.5.1 Disassembly of bearings with cylindrical bores

For small type bearings, the pullers shown in **Fig. 15.15 a**) and **b**) or the press method shown in **Fig. 15.16** can be used for disassembly. When used properly, these methods can improve disassembly efficiency and prevent damage to bearings.

To facilitate disassembly procedures, attention should be given to planning the designs of shafts and housings, such as providing extraction grooves on the shaft and housing for puller claws as shown **Figs. 15.17** and **15.18**. Threaded bolt holes should also be provided in housings to facilitate the pressing out of outer rings as shown in **Fig. 15.19**.

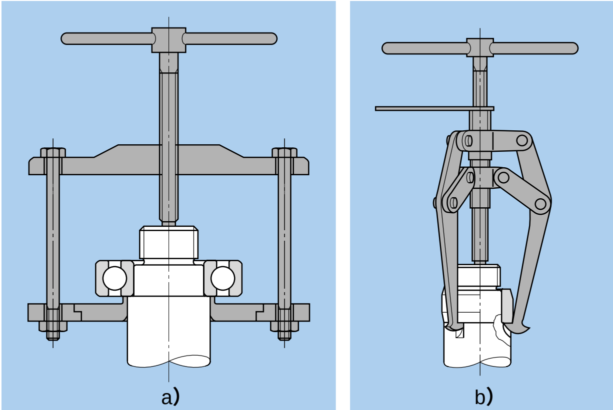


Fig. 15.15 Puller disassembly

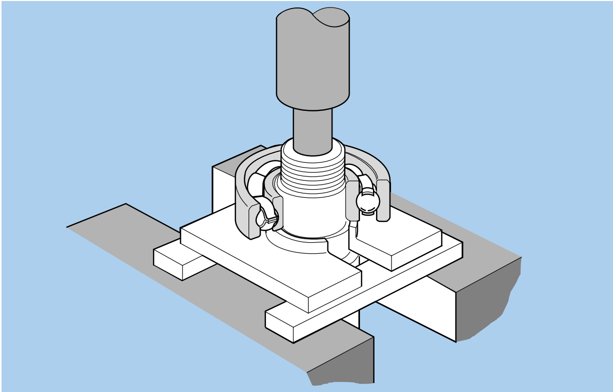


Fig. 15.16 Press disassembly

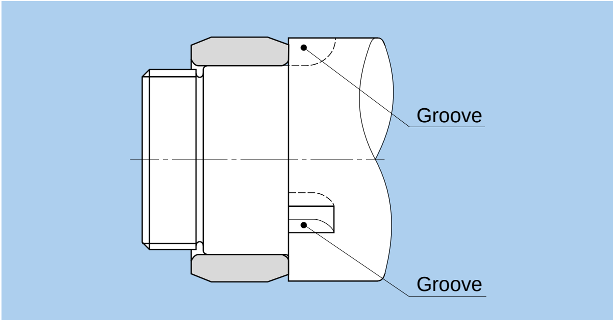


Fig. 15.17 Extracting grooves

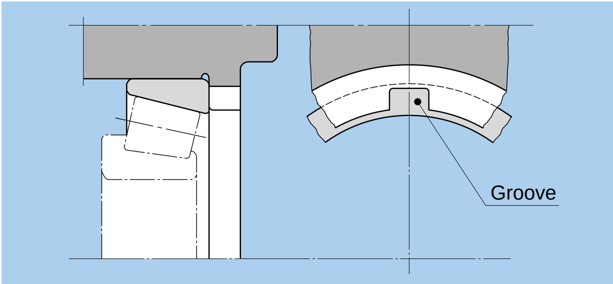


Fig. 15.18 Extraction groove for outer ring disassembly

II

Large bearings, installed with tight fits, and having been in service for a long period of time, will likely have developed fretting corrosion on fitting surfaces and will require considerable dismounting force. In such instances, dismounting friction can be reduced by injecting oil under high pressure between the shaft and inner ring surfaces as shown in **Fig. 15.20**.

For NU, NJ and NUP type cylindrical roller bearings, the induction heating unit shown in **Fig. 15.6** can be used to facilitate removal of the inner ring by means of thermal expansion. This method is highly efficient for frequent disassembly of bearings with identical dimensions.

15.5.2 Disassembly of bearings with tapered bores

Small bearings installed using an adapter are removed by loosening the locknut, placing a block on the edge of the inner ring as shown in **Fig. 15.21**, and tapping with a hammer. Bearings which have been installed with withdrawal sleeves can be disassembled by tightening down the lock nut as shown in **Fig. 15.22**.

For large type bearings on tapered shafts, adapters, or withdrawal sleeves, disassembly is greatly facilitated by hydraulic methods. **Fig. 15.23** shows the case where the bearing is removed by applying hydraulic pressure on the fitting surface of a bearing installed on a tapered shaft.

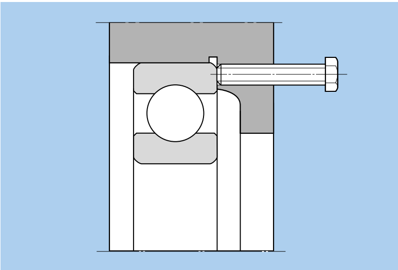


Fig. 15.19 Outer ring disassembly bolt

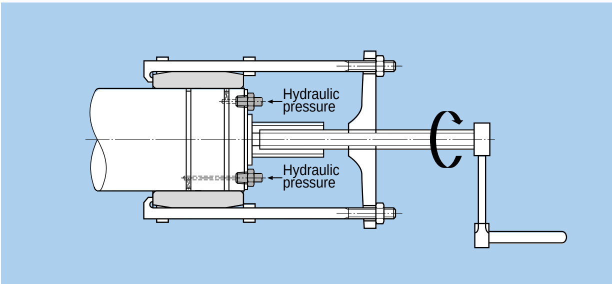


Fig. 15.20 Removal by hydraulic pressure

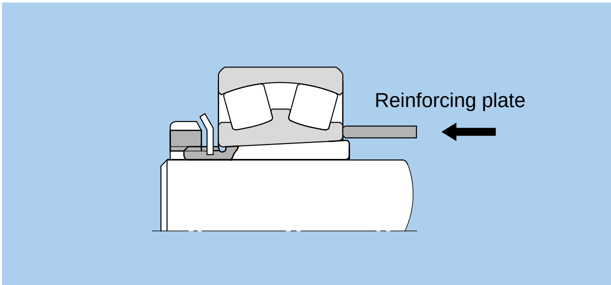


Fig. 15.21 Disassembly of bearing with adapter

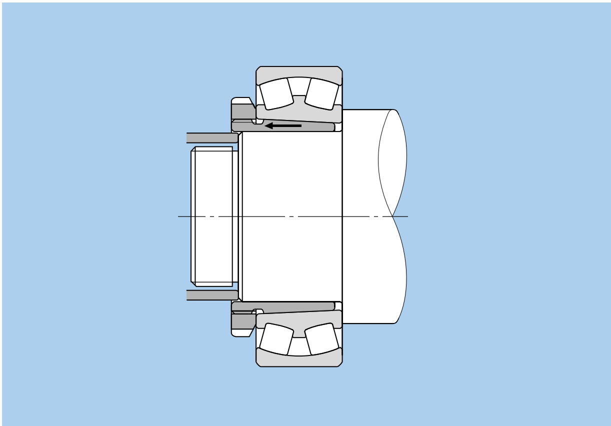


Fig. 15.22 Disassembly of bearing with withdrawal sleeve

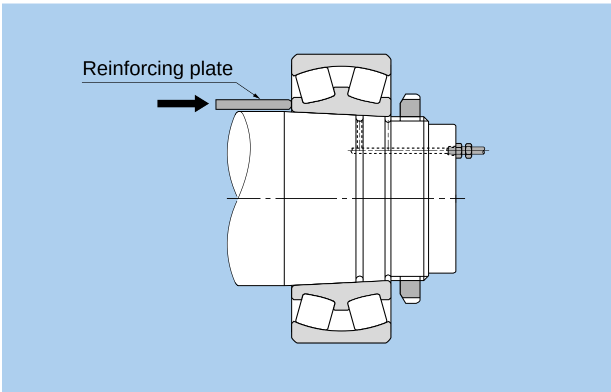
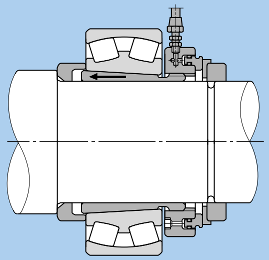
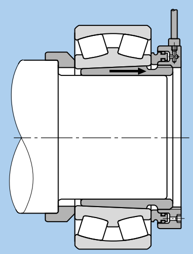


Fig. 15.23 Removal of bearing by hydraulic pressure

Fig. 15.24 shows two methods of disassembling bearings with adapters or withdrawal sleeves using a hydraulic nut. **Fig. 15.25** shows a disassembly method using a hydraulic withdrawal sleeve where high pressure oil is injected between fitting surfaces and a nut is then employed to remove the sleeve.



a) Disassembly of adapter sleeve



b) Disassembly of withdrawal sleeve

fig. 15.24 Disassembly using hydraulic nut

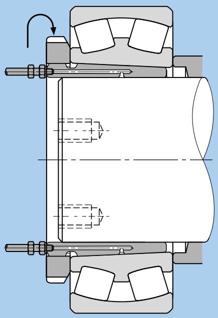


Fig. 15.25 Disassembly using hydraulic withdrawal sleeve

15.6 Bearing maintenance and inspection

In order to get the use the bearing to its full potential and keep it in good working condition as long as possible, maintenance and inspections should be performed. Doing so will enable early detection of any problems with the bearing.

This will enable you to prevent bearing failure before it happens, and will enhance productivity and cost performance.

The following measures are often taken as a general method of maintaining and managing bearings.

Maintenance management requires inspection items and frequency for performing routine inspections be determined according to the importance of the device or machine.

15.6.1 Inspection of machine while running

The interval for replenishing and replacing lubricant is determined by a study of lubricant nature and checking the bearing temperature, noise and vibration.

15.6.2 Observation of bearing after use

Take note of any problem that may appear after the bearing is used or when performing routine inspections, and take measures for preventing reoccurrence of any damage discovered. For types of bearing damage and countermeasures for preventing damage, see section 16.

II

16. Bearing Damage and Corrective Measures

If handled correctly, bearings can generally be used for a long time before reaching their fatigue life. If damage occurs prematurely, the problem could stem from improper bearing selection, handling or lubrication. In this occurs, take note of the type of machine on which the bearings is used, the place where it is mounted, service conditions and surrounding structure. By investigating several possible causes surmised from the type of damage and condition at the time the damage occurred, it is possible to prevent the same kind of damage from reoccurring. **Table 16.1** gives the main causes of bearing damage and remedies for correcting the problem.

Table 16.1 Bearing damage, main causes of bearing damage and remedies for correcting the problem

Description			
Flaking	Surface of the raceway and rolling elements peels away in flakes Conspicuous hills and valleys form soon afterward.  	Causes	● Excessive load, fatigue life, improper handling ● Improper mounting. ● Improper precision in the shaft or housing. ● Insufficient clearance. ● Contamination. ● Rust. ● Improper lubrication ● Drop in hardness due to abnormally high temperatures.
		Correction	● Select a different type of bearing. ● Reevaluate the clearance. ● Improve the precision of the shaft and housing. ● Review application conditions. ● Improve assembly method and handling. ● Reevaluate the layout (design) of the area around the bearing. ● Review lubricant type and lubrication methods.
Seizure	The bearing heats up and becomes discolored. Eventually the bearing will seize up. 	Causes	● Insufficient clearance (including clearances made smaller by local deformation). ● Insufficient lubrication or improper lubricant. ● Excessive loads (excessive preload). ● Skewed rollers. ● Reduction in hardness due to abnormal temperature rise
		Correction	● Riview lubricant type and quantity. ● Check for proper clearance. (Increase clearances.) ● Take steps to prevent misalignment. ● Review application conditions. ● Improve assembly method and handling.
Cracking and notching	Localized flaking occurs. Little cracks or notches appear.  	Causes	● Excessive shock loads. ● Improper handling (use of steel hammer, cutting by large particles of foreign matter) ● Formation of decomposed surface layer due to improper lubrication ● Excessive interference. ● Large flaking. ● Friction cracking. ● Imprecision of mounting mate (oversized fillet radius)
		Correction	● Review lubricant (friction crack prevention). ● Select proper interference and review materials. ● Review service conditions. ● Improve assembly procedures and take more care in handling.

Table 16.1 Bearing damage, main causes of bearing damage and remedies for correcting the problem

Description

Cage damage

Rivets break or become loose resulting in cage damage.


Causes

- Excessive moment loading.
- High speed or excessive speed fluctuations.
- Inadequate lubrication.
- Impact with foreign objects.
- Excessive vibration.
- Improper mounting. (Mounted misaligned)

Correction

- Reevaluation of lubrication conditions.
- Review of cage type selection.
- Investigate shaft and housing rigidity.
- Review service conditions.
- Improve assembly method and handling.

Rolling path skewing

Abrasion or an irregular, rolling path skewing left by rolling elements along raceway surfaces.


Causes

- Shaft or housing of insufficient accuracy.
- Improper installation.
- Insufficient shaft or housing rigidity.
- Shaft whirling caused by excessive internal bearing clearances.

Correction

- Reinspect bearing's internal clearances.
- Review accuracy of shaft and housing finish.
- Review rigidity of shaft and housing.

Smearing and scuffing

The surface becomes rough and some small deposits form. Scuffing generally refers to roughness on the race collar and the ends of the rollers.


Causes

- Inadequate lubrication.
- Entrapped foreign particles.
- Roller skewing due to a misaligned bearing.
- Bare spots in the collar oil film due to large axial loading.
- Surface roughness.
- Excessive slippage of the rolling elements.

Correction

- Reevaluation of the lubricant type and lubrication method.
- Bolster sealing performance.
- Review preload.
- Review service conditions.
- Improve assembly method and handling

Rust and corrosion

The surface becomes either partially or fully rusted, and occasionally rust even occurs along the rolling element pitch lines.


Causes

- Poor storage conditions.
- Poor packaging.
- Insufficient rust inhibitor.
- Penetration by water, acid, etc.
- Handling with bare hands.

Correction

- Take measures to prevent rusting while in storage.
- Periodically inspect the lubricating oil.
- Improve sealing performance.
- Improve assembly method and handling.

Table 16.1 Bearing damage, main causes of bearing damage and remedies for correcting the problem

Description			
Fretting	There are two types of fretting. In one, a rusty wear powder forms on the mating surfaces. In the other, brinelling indentations form on the raceway at the rolling element pitch.  	Causes	• Insufficient interference. • Small bearing oscillation angle. • Insufficient lubrication.(unlubricated) • Fluctuating loads. • Vibration during transport, vibration while stopped.
		Correction	• Select a different kind of bearing. • Select a different type of lubricant. • Review the interference and apply a coat of lubricant to fitting surface. • Pack the inner and outer rings separately for transport.
Wear	The surfaces wear and dimensional deformation results. Wear is often accompanied by roughness and scratches. 	Causes	• Entrapment of foreign particles in the lubricant. • Inadequate lubrication. • Skewed rollers.
		Correction	• Review lubricant type and lubrication methods. • Improve sealing performance. • Take steps to prevent misalignment.
Electrolytic corrosion	Pits form on the raceway. The pits gradually grow into ripples.  	Causes	• Electric current flowing through the rollers.
		Correction	• Create a bypass circuit for the current. • Insulate the bearing.
Dents and scratches	Scoring during assembly, gouges due to hard foreign objects, and surface denting due to mechanical shock.  	Causes	• Entrapment of foreign objects. • Bite-in on the flaked-off side. • Dropping or other mechanical shocks due to careless handling. • Assembled misaligned.
		Correction	• Improve handling and assembly methods. • Bolster sealing performance. (measures for preventing foreign matter from getting in) • Check area surrounding bearing. (when caused by metal fragments)

II

Table 16.1 Bearing damage, main causes of bearing damage and remedies for correcting the problem

Description			
Creeping	Surface becomes mirrored by sliding of inside and outside diameter surfaces. May by accompanied by discoloration or score.  	**Causes**	● Insufficient interference in the mating section. ● Sleeve not fastened down properly. ● Abnormal temperature rise. ● Excessive loads.
		Correction	● Reevaluate the interference. ● Reevaluate usage conditions. ● Review the precision of the shaft and housing. ● Raceway end panel scuffing
Speckles and discoloration	Luster of raceway surfaces is gone; surface is matted, rough, and / or evenly dimpled. Surface covered with minute dents.  	**Causes**	● Infiltration of bearing by foreign matter. ● Insufficient lubrication.
		Correction	● Reevaluation of lubricant type and lubrication method. ● Review sealing mechanisms. ● Examine lubrication oil purity. (filter may be excessively dirty, etc.)
Peeling	Patches of minute flaking or peeling (size, approx. 10 μm). Innumerable hair-line cracks visible though not yet peeling. (This type of damage frequently seen on roller bearings.) 	**Causes**	● Infiltration of bearing by foreign matter. ● Insufficient lubrication.
		Correction	● Reevaluation of lubricant type and lubrication method. ● Improve sealing performance. (to prevent infiltration of foreign matter) ● Take care to operate smoothly.

17. Technical data

17.1 Deep groove ball bearing radial internal clearances and axial internal clearances

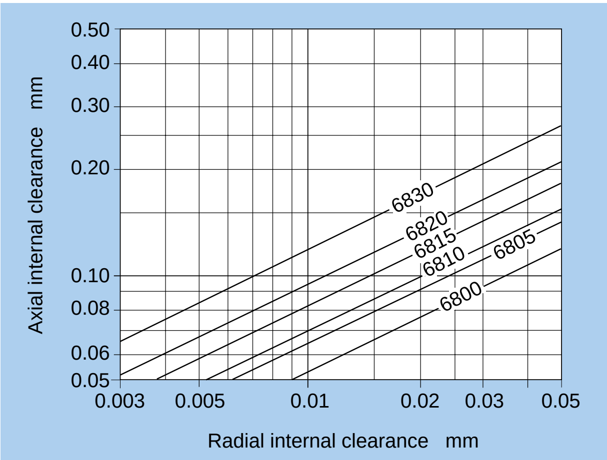


Fig. 17.1.1 Series 68 radial internal/axial internal clearances

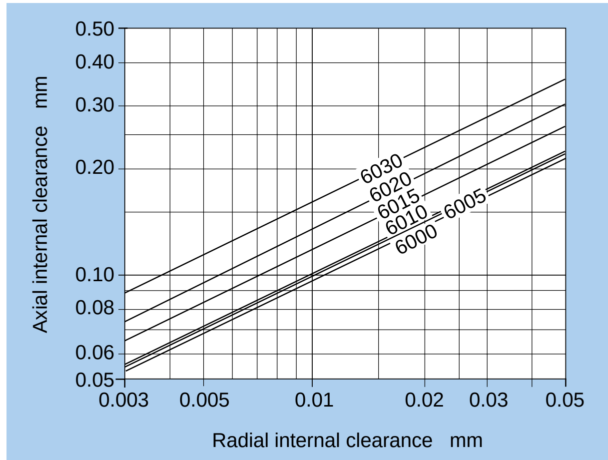


Fig. 17.1.3 Series 60 radial internal/axial internal clearances

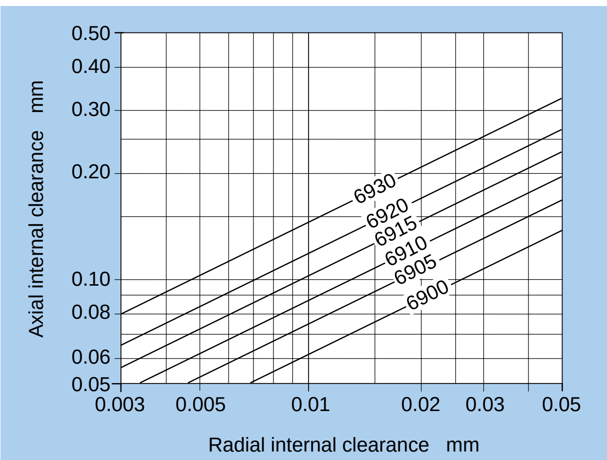


Fig. 17.1.2 Series 69 radial internal/axial internal clearances

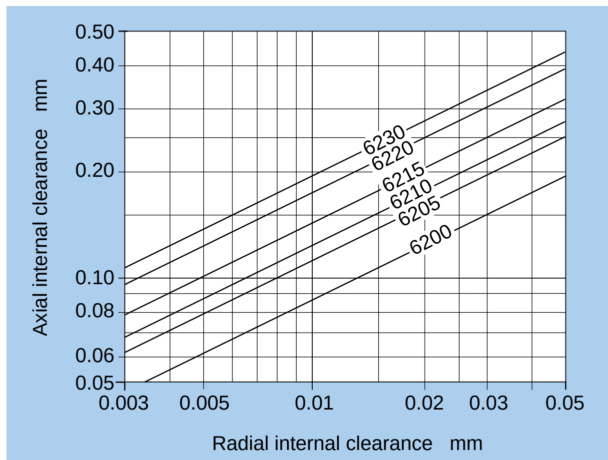


Fig. 17.1.4 Series 62 radial internal/axial internal clearances

17.2 Angular contact ball bearing axial load and axial displacement

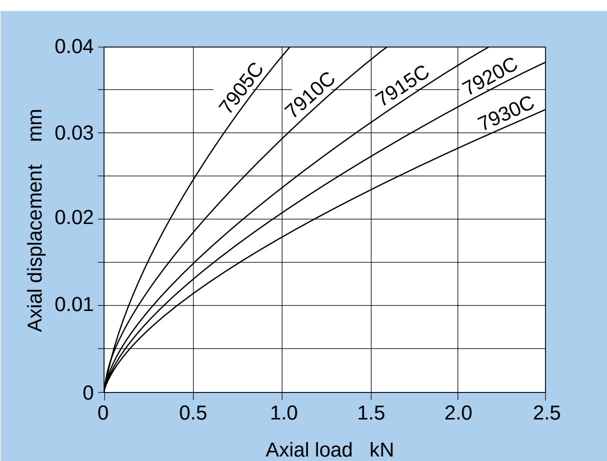


Fig. 17.2.1 Series 79 C axial load and axial displacement

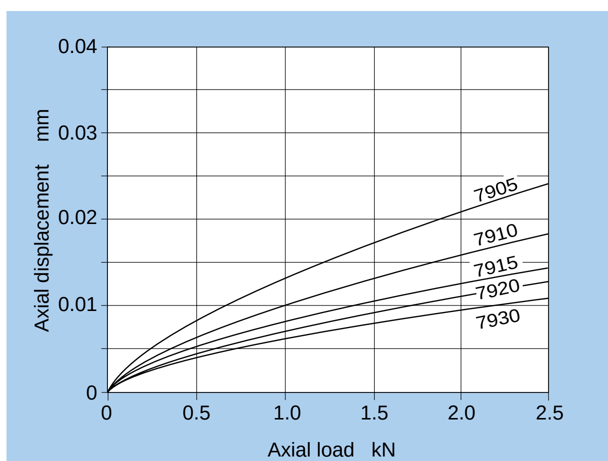


Fig. 17.2.2 Series 79 axial load and axial displacement

II

※This data is based on typical dimensions. NTN do not guarantee at this data.

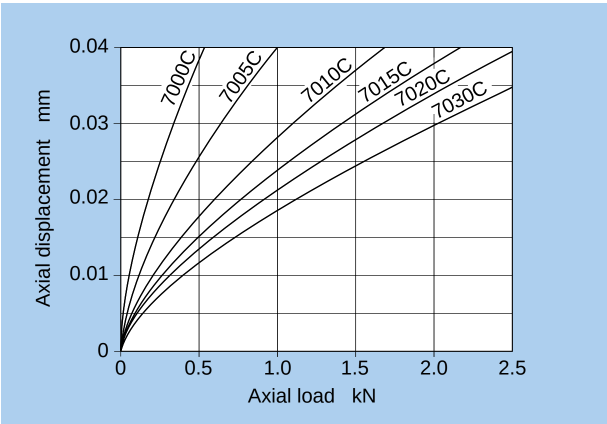


Fig. 17.2.3 Series 70 C axial load and axial displacement

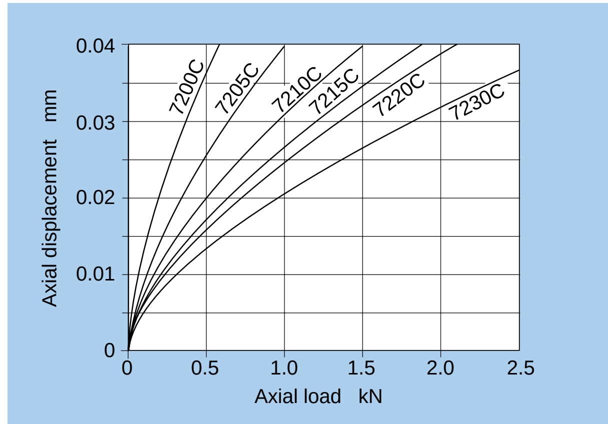


Fig. 17.2.6 Series 72 C axial load and axial displacement

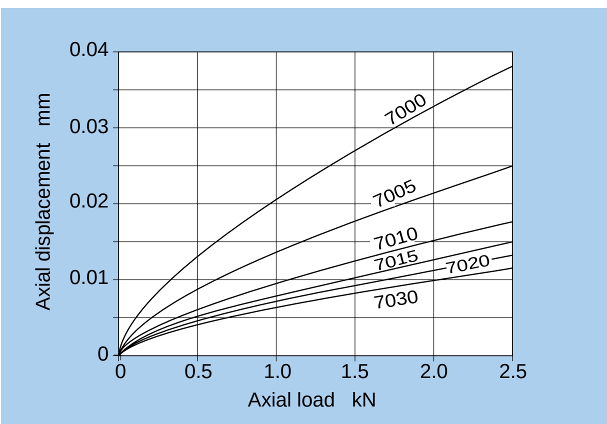


Fig. 17.2.4 Series 70 axial load and axial displacement

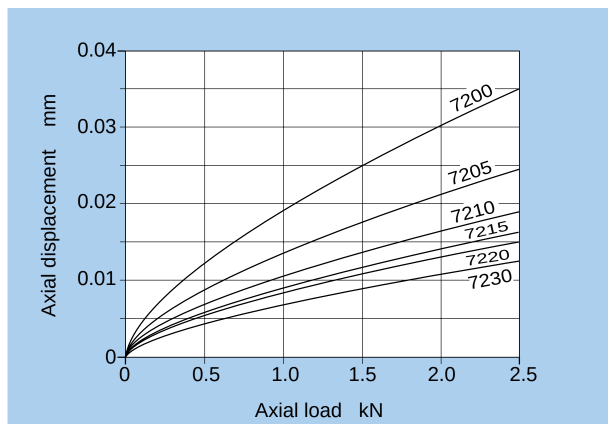


Fig. 17.2.7 Series 72 axial load and axial displacement

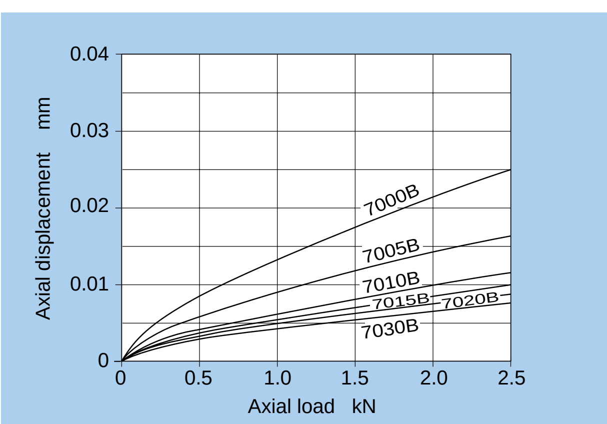


Fig. 17.2.5 Series 70 B axial load and axial displacement

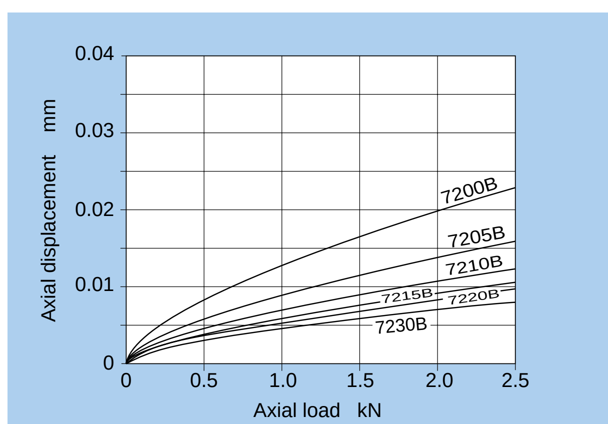


Fig. 17.2.8 Series 72 B axial load and axial displacement

※This data is based on typical dimensions. NTN do not guarantee at this data.

17.3 Tapered roller bearing axial load and axial displacement

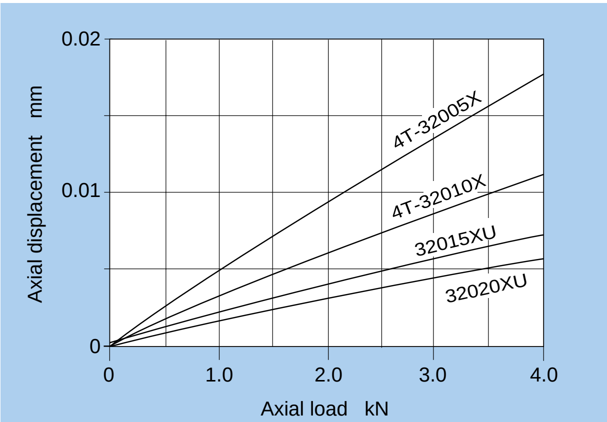


FIg. 17.3.1 Series 320 axial load and axial displacement

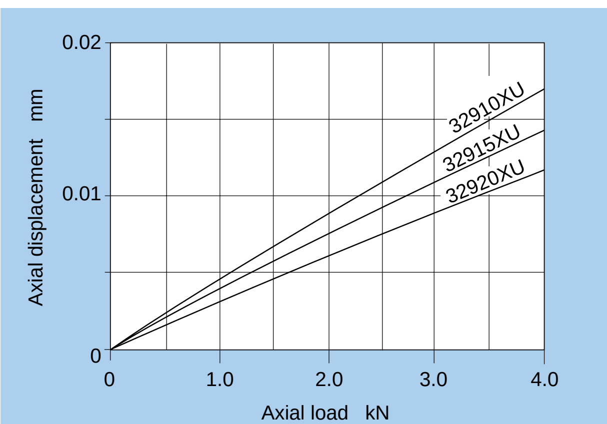


Fig. 17.3.2 Series 329 axial load and axial displacement

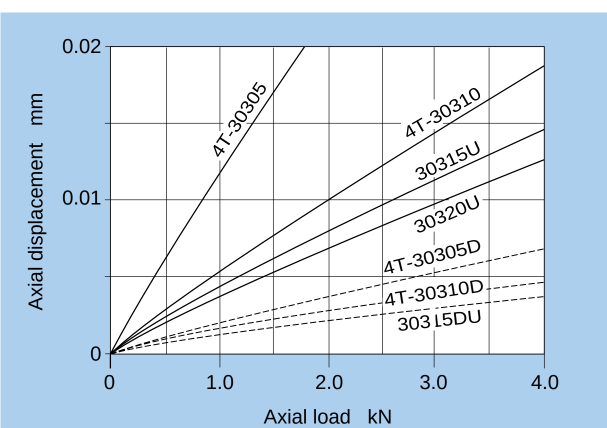


FIg. 17.3.3 Series 303/303 D axial load and axial displacement

Note: Values when bearing and housing are rigid bodies.
Axial displacement may become large depending on shape of shaft/housing and fitting conditions.

17.4 Allowable axial load for ball bearings

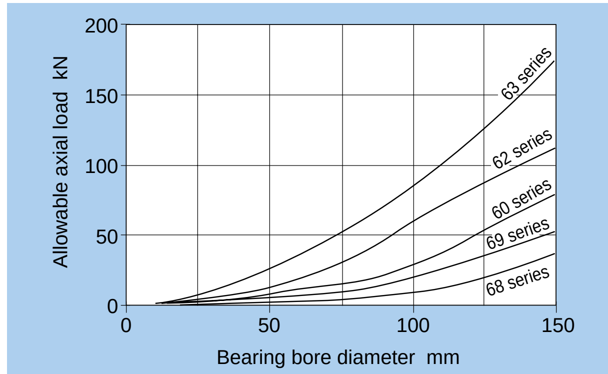


Fig. 17.4.1 Allowable axial load for deep groove ball bearings

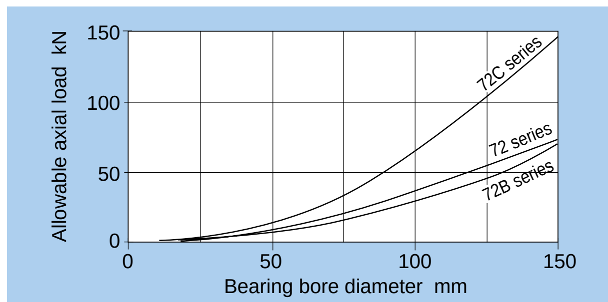


Fig. 17.4.2 Allowable axial load for angular contact ball bearings (72, 72B, 72C series)

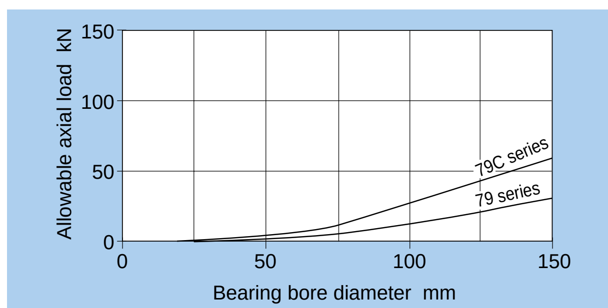


Fig. 17.4.3 Allowable axial load for angular contact ball bearings (79, 79C series)

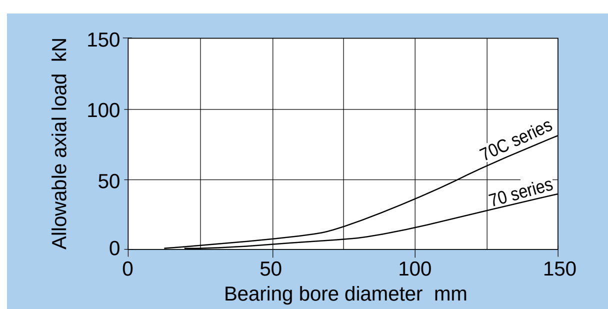


Fig. 17.4.4 Allowable axial load for angular contact ball bearings (70, 70C series)

Note: When an axial load acts upon deep groove or angular contact ball bearings, allowable axial load is the load whereby the contact ellipse exceeds the shoulder of the raceway.

II

17.5 Fitting surface pressure

Table 17.5.1 lists equations for calculating the pressure and maximum stress between fitting surfaces.

Table 17.5.2 can be used to determine the approximate average groove diameter for bearing inner and outer rings.

The effective interference, in other words the actual interference Δ_{deff} after fitting, is smaller than the apparent interference Δd derived from the measured valued for the bearing bore diameter and shaft. This difference is due to the roughness or variations of the finished surfaces to be fitted, and therefore it is necessary to assume the following reductions in effective interference:

For ground shafts: 1.0 ～ 2.5 μm
For lathed shafts : 5.0 ～ 7.0 μm

Table 17.5.1 Fitting surface pressure and maximum stress

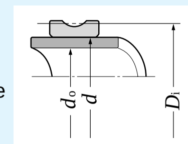
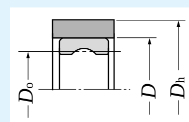
Fit conditions		Equation	Codes (units: N{kgf}, mm)
Fitting surface pressure MPa {kgf / mm²}	Solid steel shaft/ inner ring fit	$P = \frac{E}{2} \frac{\Delta_{deff}}{d} \left\{ 1 - \left(\frac{d}{D_i} \right)^2 \right\}$	d : Shaft diameter, inner ring bore diameter d_o : Hollow shaft inner diameter D_i : Inner ring average groove diameter Δ_{deff} : Effective interference E : Elasticity factor = 208,000 MPa {21,200 kgf / mm²} 
	Hollow steel shaft/ inner ring fit	$P = \frac{E}{2} \frac{\Delta_{deff}}{\Delta d} \frac{[1-(d/D_i)^2]\,[1-(d_o/d)^2]}{[1-(d_o/D_i)^2]}$	
	Steel housing/ outer ring fit	$P = \frac{E}{2} \frac{\Delta_{Deff}}{D} \frac{[1-(D_o/D)^2]\,[1-(D/D_h)^2]}{[1-(D_o/D_h)^2]}$	D : Housing inner diameter, bearing outer diameter D_o : Outer ring average groove diameter D_h : Housing outer diameter Δ_{Deff} : Effective interference 
Maximum stress MPa {kgf / mm²}	Shaft / inner ring fit	$\sigma_t \max = P \frac{1 + (d/D_i)^2}{1 - (d/D_i)^2}$	Inner ring bore diameter face maximum tangential stress
	Housing/ outer ring fit	$\sigma_t \max = P \frac{2}{1 - (D_o/D)^2}$	Outer ring inner diameter face maximum tangential stress

Table 17.5.2 Average groove diameter (approximate expression)

Bearing type		Average groove diameter	
		Inner ring (D_i)	Outer ring (D_o)
Deep groove ball bearings	All types	$1.05 \frac{4d + D}{5}$	$0.95 \frac{d + 4D}{5}$
Cylindrical roller bearings❶	All types	$1.05 \frac{3d + D}{4}$	$0.98 \frac{d + 3D}{4}$
Spherical roller bearings	All types	$\frac{2d + D}{3}$	$0.97 \frac{d + 4D}{5}$

d: Inner ring bore diameter mm D: Outer ring outer diameter mm
❶ Average groove diameter values shown for double-flange type.

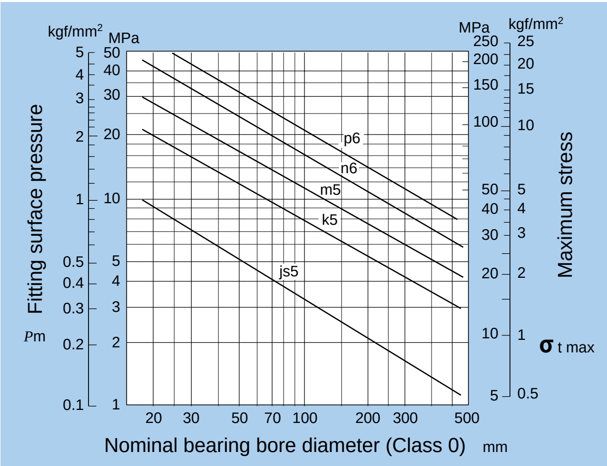


Fig. 17.5.1 Average fit interference as it relates to surface pressure P_m **and max. stress** $\sigma_{t\ max}$

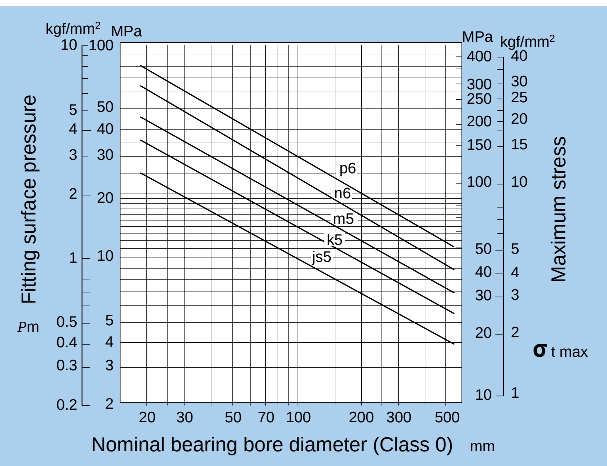


Fig. 17.5.2 Maximum fit interference as it relates to surface pressure P_m **and max. stress** $\sigma_{t\ max}$

❶ For recommended fitting, see page A-50.

17.6 Necessary press fit and pullout force

Equations (17.1) and (17.2) below can be used to calculate the necessary pullout force for press fit for inner rings and shafts or outer rings and housings.

For shaft and inner rings:

$$K_d = \mu \cdot P \cdot \pi \cdot d \cdot B \cdots\cdots\cdots\cdots\cdots\cdots (17.1)$$

For housing and outer rings:

$$K_D = \mu \cdot P \cdot \pi \cdot D \cdot B \cdots\cdots\cdots\cdots\cdots\cdots (17.2)$$

Where,

K_d : Inner ring press fit or pullout force N {kgf}
K_D : Outer ring press fit or pullout force N {kgf}
P : Fitting surface pressure MPa {kgf/mm²} (Refer to **Table 17.5.1**)
d : Shaft diameter, inner ring bore diameter mm
D : Housing inner diameter, outer ring outer diameter mm
B : Inner or outer ring width
μ : Sliding friction coefficient (Refer to **Table 17.6.1**)

Table 17.6.1 Press fit and pullout sliding friction coefficient

Type	μ
Inner (outer) ring press fit onto cylindrical shaft (bore)	0.12
Inner (outer) ring pullout from cylindrical shaft (bore)	0.18
Inner ring press fit onto tapered shaft or sleeve	0.17
Inner ring pullout from tapered shaft	0.14
Sleeve press fit onto shaft/bearing	0.30
Sleeve pullout from shaft/bearing	0.33

Precision Shaft Couplings

In the simplest of terms a coupling's purpose is to transfer rotational movement from one shaft to another. Reality is somewhat more complicated, though, as flexible shaft couplings have also to compensate for misalignment between the two shafts. This ability must be balanced with the need to be pliable in the planes of misalignment while still having the torsional strength to carry out the coupling's main function. This is known as the Compliance Mechanism where compliance is the capacity for allowing relative displacement.

Several factors should always be taken into consideration when looking to specify flexible shaft couplings. These are torsional stiffness, backlash, torque, life and attachment system. All of these have a bearing on coupling selection.

TORSIONAL STIFFNESS

This is the measure of resistance to torsional rotation in the coupling, and in applications such as closed loop velocity and motion control systems it needs to be high. Whereas in systems where the transmission is subject to shock loads, the coupling requires a low torsional stiffness, sometimes referred to as torsional damping.

BACKLASH

The free play between input and output shafts is commonly referred to as backlash. If rotation is constant then backlash has little impact. However, if the system requires changes in rotational direction, a dwell is created which in high-speed, short cycle applications can create noise and instability. In open-loop systems backlash will also cause loss of accuracy.

TORQUE

A coupling's torque capacity can be defined in several ways including nominal torque, reversing torque and peak torque. As far as Huco's products are concerned a coupling's capacity to transfer rotation under load is qualified by its peak torque rating. This figure is determined through Huco's testing procedures and is the maximum reversing torque applied over at least one million cycles without loss of performance. More information on Torque can be found on page 14

LIFE

The life expectancy of any flexible coupling is dependent on the individual application. Therefore, published performance values, which are based on extensive simulations, are intended as a guide. For instance, where perfect shaft alignment is the case, a coupling can sustain its peak torque value almost indefinitely. However, where misalignment extends beyond the recommended limits, failure can be induced in disc, bellows and helical beam couplings, while wear will be accelerated in universal joints and displacement couplings. Aluminium beam-type couplings will always have a finite fatigue limit when an alternating load is applied.

ATTACHMENT SYSTEMS

The simplest and most cost-effective method of attaching a coupling to a shaft is to use set screws which locate on flats or dimples on the shaft. Clamps may also be used and have the advantage that as the shaft diameter increases so does traction. For high integrity drive systems a key and keyway system should be employed.

II

Misalignment

Misalignment, or the variance between the intended position and attitude of two shafts, is normally the result of manufacturing tolerances, and quantifying misalignment is crucial when seeking to specify the correct coupling. As the misalignment increases, the transmissible torque and life expectancy of the coupling reduce exponentially. Therefore, understanding the nature and origins of misalignment is important to you as a design engineer.

The main types of misalignment are angular, radial, and axial displacement.

Factors that influence misalignment include thermal imbalances, wear, settlement and creep, and the influence of the last of these can, without correct maintenance, increase during the life of the coupling.

When determining alignment, measurements should always be taken when the system is cold and again when it is at operating temperature. Consideration should also be given to the class of tolerance being used in the assembly of the individual items. For example, the output shaft of a reduction gearbox with a die-cast housing with unmachined mounting faces and clearance holes for location purposes has a greater possibility for misalignment, than a face-mounted servo motor with machined registers.

Fig.1-10 *Sources of misalignment*

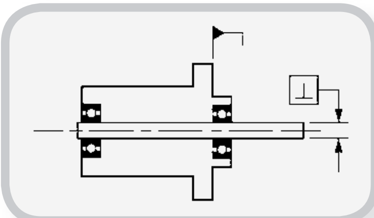


Fig.1 *Perpendicularity of shaft axis to mounting datum*

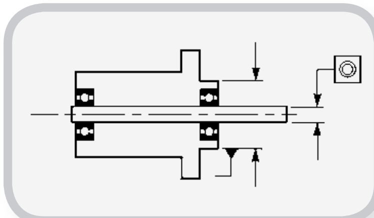


Fig.5 *Concentricity of shaft axis and locating register*

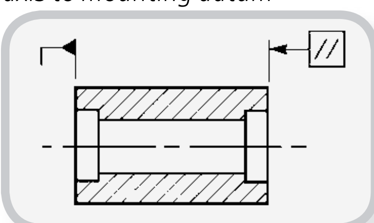


Fig.2 *Parallelism of mounting faces*

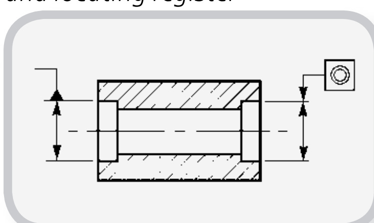


Fig.6 *Concentricity of counter bores. Clearances in locating registers*

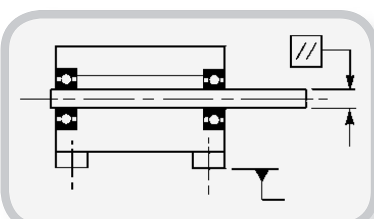


Fig.3 *Parallism of shaft axis to mounting datum*

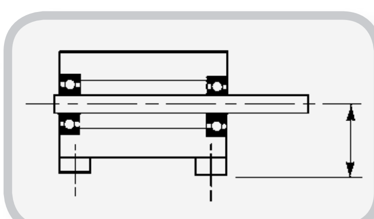


Fig.7 *Distance of shaft axis to mounting datum*

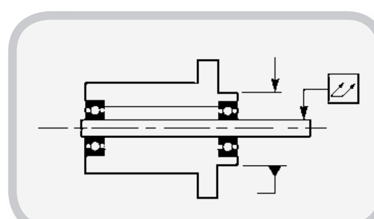


Fig.9 *Shaft run-out*

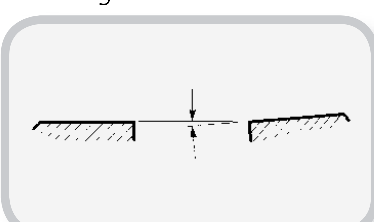


Fig.4 *Angular alignment of mounting surfaces*

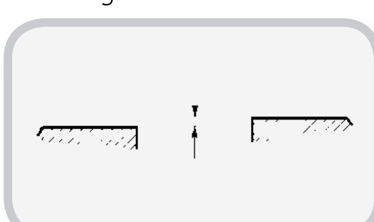


Fig.8 *Planar alignment of mounting surfaces*

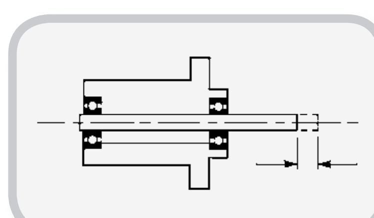


Fig.10 *Axial Displacement*

PREDICTING MISALIGNMENT

By prediction we really mean verifying the worst-case misalignment in any given situation, so as to be certain that the correction capability of the coupling is adequate. In essence shaft misalignment has three components: parallel; angular; and radial – each being three-dimensional. The following explanation and the accompanying graphics should help in clarifying this situation.

In simple terms a shaft with angular error describes a cone when it is rotated, and while mating shafts can converge and intersect on the critical plane it is unlikely. This gives rise to radial error, which is at its maximum when the axes are tangentially opposed on the sphere diameter

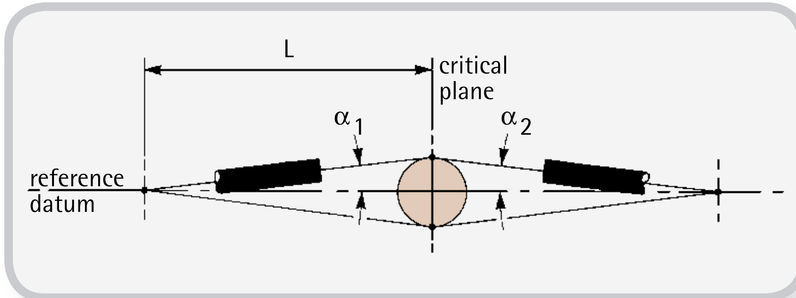


Fig.11 *Worst case angular misalignment =* $\alpha_1+\alpha_2$

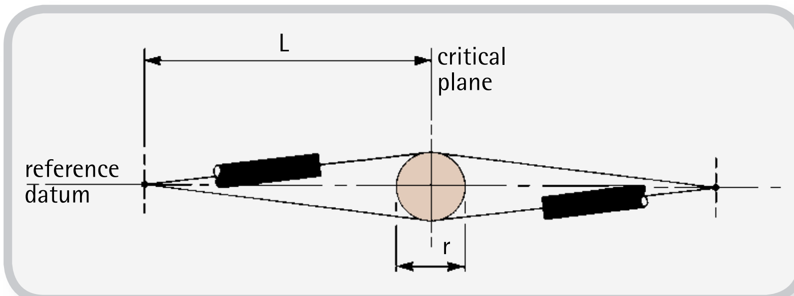


Fig.12 *Maximum radial error=* **r**

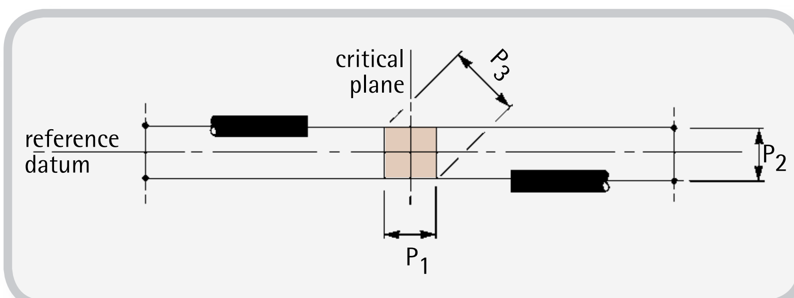


Fig.13 *Maximum parallel error =* P_3

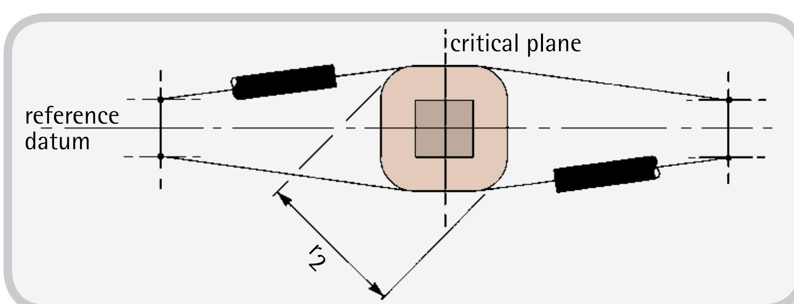


Fig.14 *Worst case radial error* $r_2 = p_3+r$

II

Selecting the ideal coupling

The choice of couplings available to today's engineers can be daunting, but follow our guidelines and you will arrive at the optimum coupling for your particular application.

- *Does the coupling provide adequate misalignment protection?*
- *Can it transmit the load torque?*
- *Do I need axial motion or axial stiffness?*
- *Can it sustain the required speed of rotation?*
- *Will it fit within the available space envelope?*
- *Can it operate at the designated ambient temperature?*
- *Does it provide torsional stiffness required for positional accuracy?*
- *Does it provide electrical isolation between the shafts?*
- *Will it have the required life expectancy?*

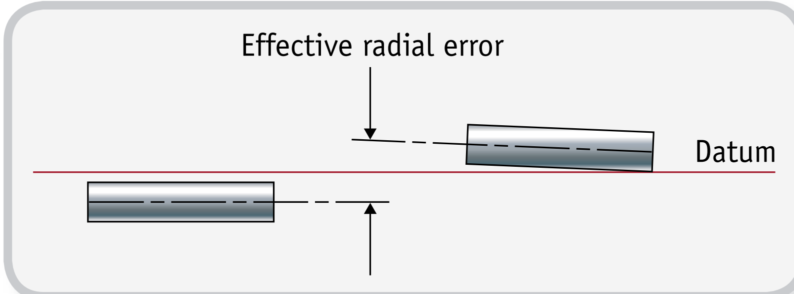


Fig.15 *Effective radial error.*

MISALIGNMENT COMPENSATION AND AXIAL MOTION

The ability to deal with misalignment and axial motion differentiates a flexible coupling from a simple rigid-type coupling. The particular mechanism used – bellows, membrane, flexible beam or sliding disc – determines the performance characteristic of the coupling, including its tolerance of misalignment or axial motion.

For instance, sliding disc and universal/lateral couplings can tolerate large misalignments, but at the cost of having their backlash-free life reduced. Bellows-type couplings, can absorb a high degree of axial motion but with a possible reduction in misalignment capacity. Membrane couplings, however, can be damaged beyond repair if axial motion exceeds the coupling's specification. That said, they can withstand large misalignments with little or no reduction in life expectancy.

Where misalignment is incidental, in other words caused simply by manufacturing tolerances, a more realistic measure is the effective radial error. This is the radial distance between the shafts' axes measured midway along the length of the coupling. Sometimes called the composite error, this can be crucial when determining a value for the maximum permissible misalignment.

Axial motion is often created as a result of axial clearances in the shaft bearings, or through thermal expansion. While it is usual to absorb this with a suitable coupling, it may, in some cases, be more beneficial to resist the motion, particularly if it has a positioning function. Couplings such as the universal/lateral type can be useful in such circumstances.

Flexible couplings are designed to protect shaft support bearings from destructive radial and thrust loads arising from misalignment and axial motion. In effect, all couplings resist these properties;

therefore, the conclusion is that those with least resistance will better protect the bearings. Figure 16 compares the radial bearing loads of some of the most popular couplings based on a nominal outside diameter of 25 mm, with the exception of the jaw coupling where a 30 mm diameter has been used.

LOAD TORQUE, INERTIA AND TORSIONAL STIFFNESS

In applications where couplings are used to drive frictional loads, for example, pumps, shutter doors and machinery, etc., the coupling's torsional stiffness is not a major factor as the angular synchronisation of the shafts is not an issue. However, when resonance is a problem, it is possible to reduce the coupling's torsional stiffness and so avoid conflict with the natural resonance of the machine.

This does not apply when the loads are inertial; typically position and velocity control systems where registration of input and output shafts is critical throughout the operating cycle. In these applications the three elements of motor, coupling and load combine to create a resonant system. The frequency of this system is controlled by the load inertia and the coupling's torsional stiffness. Increasing the inertia, or lowering the torsional stiffness, results in a lower resonant frequency.

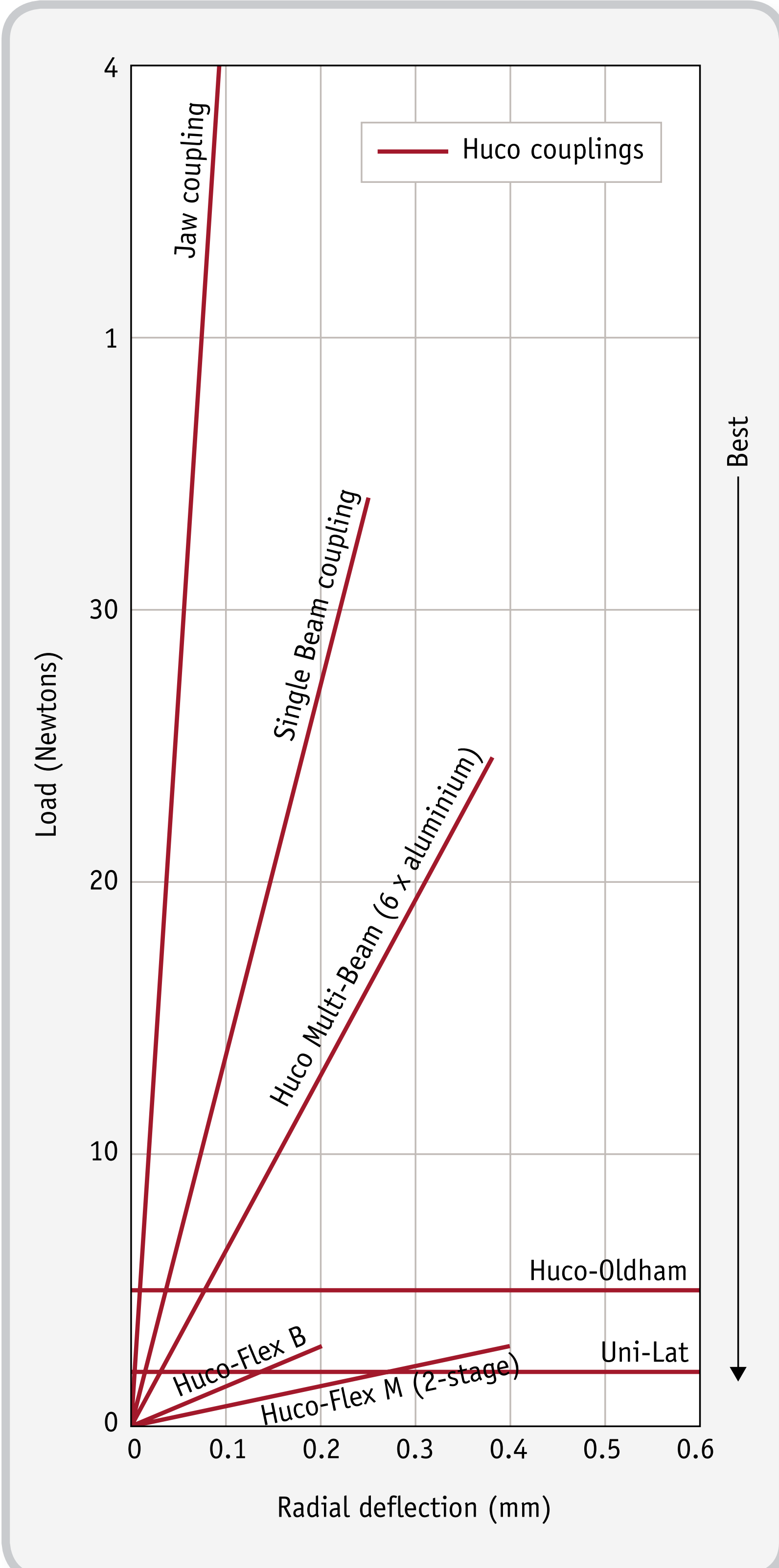


Fig.16

II

II

In order to control a resonant system you must work well below its resonant frequency. For example, imagine supporting a weight on an elastic band. You can control the weight's vertical movement if you move your hand slowly. Increase the speed and the weight barely moves.

Therefore, to improve responsiveness you require less elasticity or you need to reduce the weight. If you now substitute the elastic band with a coupling and the weight with an inertial load, you have an analogy of an inertial system.

To summarise, when the emphasis is on performance, you require a stiffer coupling in order to reduce settling times, improve positional accuracy and raise the upper limit of dynamic performance.

Torsional deflection (the inverse of torsional stiffness) for a number of the most popular couplings, based on a nominal outside diameter of 25 mm, with the exception of the jaw coupling where a 30 mm diameter has been used.

Selection criteria – which coupling does what?

FLEXIBLE SHAFT COUPLINGS

Flexible shaft-type couplings compensate for radial and angular misalignment though the flexure of a varying number of compliant elements. This type of coupling includes the multi-stage bellows, helical beam and radial slit concepts.

Points to bear in mind:

1) *The greater the number of elements, the greater the angular and radial misalignment capacity and the lower the torsional stiffness.*

2) *The forces required to effect compliance are broadly proportional to the torsional stiffness. The stiffer the coupling in torsion, the higher the resulting bearing loads.*

MEMBRANE (DISC) COUPLINGS

Thin pressed spring steel membranes act as the pivotal media in disc couplings. These are attached alternately to the drive and driven members, and provide flex to compensate for misalignment. Any torque is resolved to simple tensile stresses in the opposing segments of the membranes, which are free of residual stresses as no secondary forming operations are involved in their manufacture.

Another advantage of this type of coupling is their near-infinite life and dynamically balanced construction, making them suitable for applications where high rotational speed and high-level motion integrity are required. Typical applications include closed loop servo systems in machine tools, robots, scanners, centrifuges, turbines and dynamometers. When selecting a disc coupling, the user can specify modified spring rates, longer/shorter intermediate members and either keywayed or 'D' bore.

BELLOWS

The characteristics of the bellows coupling can be modified by varying the number and/or the wall thickness of the convolutions of the bellows. This type of coupling generally has high torsional stiffness and may be used in any drive system where high levels of motion integrity are essential. Typical applications include encoder drives in closed-loop servo systems. Coupling options include modified spring rates, along with keywayed and 'D' bores.

TWO-STAGE BELLOWS

As opposed to the multi-stage bellows, the two-stage version, as the name implies, has only two convolutions. Removing the additional convolutions has the effect of increasing torsional stiffness. By virtue of this benefit this type of coupling has limited misalignment capacity and is, therefore, ideally suited to high precision, high-resolution applications. These include main axis drives in closed loop velocity and position control systems, encoders, resolvers and tachogenerators. Options include keywayed and 'D' bores.

FLEXIBLE BEAM COUPLINGS

The beam coupling is made from one piece of material achieving its flexibility in all three modes; angular, radial and axial, by means of a slot or slots machined through the wall of the material. Most commonly, the slots are machined helically around the circumference of the coupling. Straight radial slots are also sometimes used.

Helical beam couplings may have one two or three start helices, a three-start helix providing the highest level of torsional stiffness and hence signal accuracy.

Even higher torsional stiffness can be achieved with straight radial beams, however, this is at the expense of radial and angular flexibility.

As with other types of coupling, increased radial compliance is achieved by joining together two flexible coupling 'stages' separated by a spacer.

II

PLASTIC DOUBLE LOOP COUPLINGS

This type of coupling uses a moulded plastic element permanently swaged to steel or stainless steel hubs to form an effective two-stage coupling with exceptional flexibility in all three modes. Ideal for transmitting rotation in small drives, this type of coupling works without any friction, wear or noise, although its low torsional stiffness makes it less suitable for high precision positioning applications.

JAW COUPLINGS

Most commonly used to provide some flexibility and misalignment compensation in high power transmission systems, the jaw type coupling achieves its flexibility by means of a plastic element sandwiched between metal hubs. If the hubs are not in true alignment the element deforms to accommodate this difference. Flexibility is somewhat limited as the plastic is compressed, storing energy like a spring, which creates a high resistive force that can cause excessive radial loads to be transmitted to the shaft bearings.

Points to bear in mind:

1) *The greater the distance between the pivotal planes (see figure 17 right), the greater the radial misalignment capacity.*

2) *Torsional stiffness reduces marginally depending on the length and stiffness of the intermediate member.*

3) *Angular misalignment capacity cannot be increased beyond the coupling's basic capability, irrespective of the distance between pivotal planes.*

LATERAL DISPLACEMENT COUPLINGS

By using an intermediate member that slides in a plane perpendicular to the axis of rotation, lateral displacement couplings accommodate both radial and angular misalignment. By virtue of its orbital motion the coupling aligns one hub, then the other: in effect, straddling the misalignment. The use of a mechanical sliding contact means that quite large radial errors — which are proportionate to the coupling's diameter — can be overcome without the penalty of a long coupling.

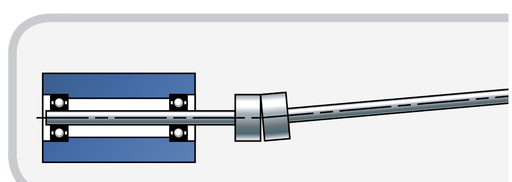
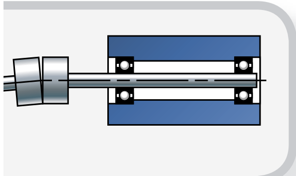


Fig.17 *Two single-stage couplers locate a fully-floating shaft on a stable axis of rotation.*

II

THE OLDHAM

This three-part coupling transmits rotation through a central plastic disc that slides over the tenons on the hubs under controlled pre-load to eliminate backlash. The disc can be manufactured from a variety of engineering polymers to suit many different applications. These range from the incremental control of fluid valves to positional systems in machine tools, robots and slide tables. They can also be applied to microstepper and closed loop servo systems and, to a lesser extent, half and full step motor drives. They are available with keywayed or 'D' bores in through bore types, and also with radiation and heat resistant torque discs and free running discs (no pre-load)



THE UNI-LAT

To combat angular and radial misalignment this coupling type combines the sliding mechanism of the Oldham (see above) with the pivotal action of the universal joint. The process uses a series of integral pins engage a pair of injection moulded annular rings that feature controlled pre-load to eliminate backlash. The main features of the UNI-LAT are the generous angular and radial misalignment capacity, along with the fact that they are electrically isolating.

The application area for these couplings is found in general purpose, light-duty stepper (half and full step) encoder, resolver and tachogenerator drives, and light pull/push duties. Normally supplied with 'D' bores, the UNI-LAT can also have other features machined into the hubs.

UNIVERSAL JOINTS

Universal joint couplings use a mechanical pivotal action controlled by radial bearings. In the case of the Huco-Pol range, these couplings are injection moulded in Acetal and benefit from controlled pre-load to eliminate backlash. This type of coupling has a large offset capacity, along with good torsional damping, water resistance and the added benefit of being lubrication-free. Universal couplings are ideal general-purpose units used typically in light-duty drives in the food, textile, paper handling and packaging environments. Connection options include gears, pulleys, square, keyways and hexagonal bores, and 'D' bores with spring clips and with non-pre-loaded bearings.

Cardan couplings also use a two-stage pivotal action with defined fulcrums to handle radial and angular misalignment.

II

Selecting for true angular misalignment

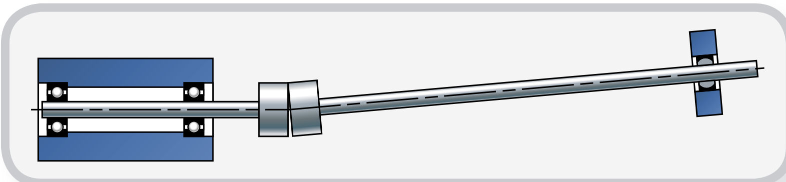


Fig.18 *A semi-floating shaft located by a self -aligning bearing at one end should be supported with a single-stage coupling at the other.*

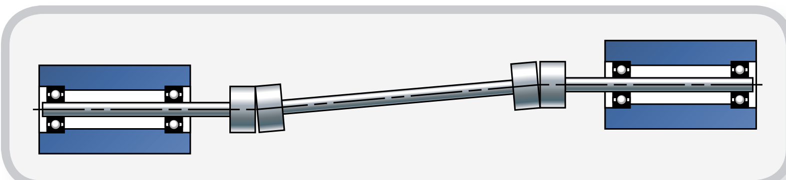


Fig.19 *Two single-stage couplings locate a fully-floating shaft on a stable axis of rotation.*

The common causes of true angular misalignment are when one of the connected shafts is compliantly mounted; for example, when it is located by a self-aligning bearing (See figure 18). Alternatively, it could be that an unsupported intermediate shaft is placed between the driver and the load (See figure 19).

Because the shafts are not mounted conventionally, they will self-align to intersect at the centre of the coupling, which acts as a hinge and, to a degree, a radial bearing. As the coupling is locating the shafts on a stable axis of rotation, it should be of the single-stage type due to the fact that any radial compliance in the coupling is counter-productive.

While couplings based on the flexible shaft can be used in these circumstances, there is a possibility that the coupled system may go into lateral oscillation. This is best described by visualising the effect of a belt and pulley drive mounted on the compliant shaft. Having a lateral compliance capacity, the coupling responds to fluctuating tension in the belt by allowing lateral oscillation of the shaft (See figure 20).

The shafts in figure 18 are described as semi-floating, while those in figure 19 are fully-floating. This is an important point as under no circumstances should a coupling with lateral displacement be used with floating shafts. The reason is that this type of coupling has no self-centring action and its use would allow the shafts to orbit in an uncontrolled way.

Couplings capable of overcoming true angular misalignment include the **single universal joint** with its capacity to handle large offsets, torsional damping, water resistance and lubrication-free operation. **Single-stage disc couplings** are also ideal, thanks to their near-infinite life and built-in dynamically balanced properties. Similarly, **single-stage bellows** with their high torsional stiffness are a good choice in this application.

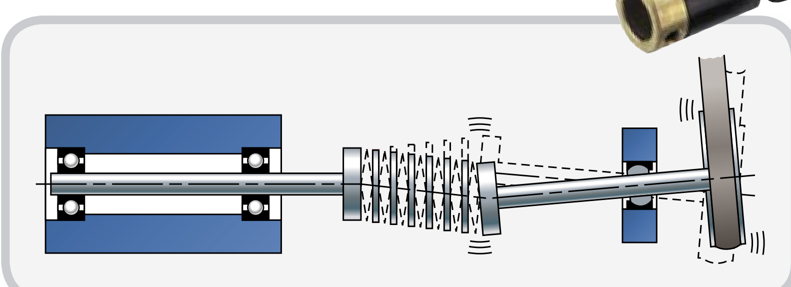


Fig.20 *A semi-floating shaft located by a self-aligning bearing at one end and a multi-stage bellows at the other. The coupling reacts to fluctuating tension in the belt by allowing lateral oscillations in the shaft.*

Selecting for zero misalignment

Zero misalignment can be achieved by assembling both shafts in self-aligning bearings (See figure 21). In this way both shafts can float into concentric relationships, allowing the use of a solid coupling which simply supports the shaft in perfect alignment.

Difficulties arise when attempting to connect fixed axis shafts in this way, as the level of alignment is difficult to both achieve and maintain, due to settlement, creep, thermal expansion and contraction. The influence of these factors results in relative movement between the shafts and the alignment achieved in the factory may not be achievable 'in the field'. Therefore, a flexible coupling is always the preferred option.

Before installing a solid coupling an interesting test is to try a flexible coupling first. With the machine at normal operating temperature measure the speed and/or the current drawn by the motor. The difference between these readings and those with the solid coupling indicate the losses generated by the additional friction at the bearings.

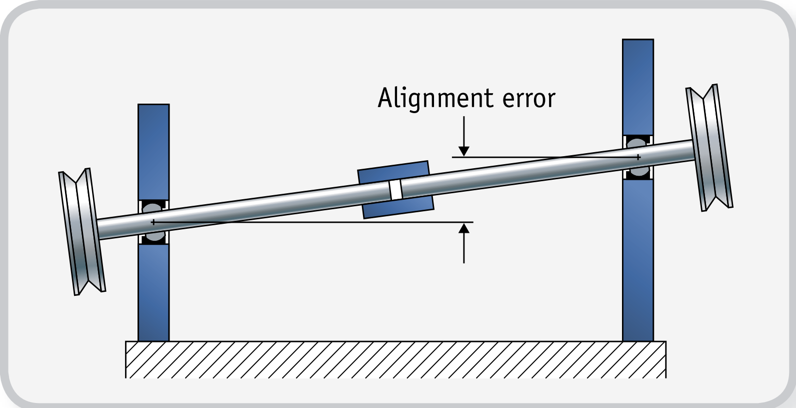


Fig.21 *Shafts located by self-aligning bearings can float into perfect alignment for connection with a solid coupling.*

II

Selecting for axial compliance

Longitudinal, or axial, shaft displacement can either be intentional or unintentional. The movements due to tolerances, settlement and thermal expansion or contraction cause the latter. While these movements maybe small, they can contribute to substantial thrust loads and result in bearing damage. In these cases a coupling with axial compliance capacity should be selected, predominantly bellows, sliding disc or even helical beam. Multi-stage bellows create the greatest amount of axial compliance while the single-stage disc or bellows provides the smallest amount of axial compliance.

For intentional shaft displacement, such as push/pull systems or those with extensible drives where the distance between actuator and load is variable, a teleshaft coupling such as the **HUCO-POL** should be used. This embodies precision-drawn nesting tubes manufactured from square-section brass that can be cut to the appropriate length to provide a wide range of axial movement. This ability to customise the length of the coupling means they can be tailored to required length of stroke.

In push/pull situations the coupling should be capable of resisting the corresponding forces. These values are listed under 'end loading' for mechanical couplings and 'axial spring rate' for flexural couplings. Another option is the **Oldham** de-mountable, three-part coupling. By mounting the hubs slightly out of full engagement, a limited amount of axial compliance is created.

II

Selecting for torque capacity

TORQUE

Torque is the angular force needed to overcome the resistance of a load. Rotating loads have both a frictional and an inertial component, and are classified according to whichever dominates. For example, the resistance encountered by a pump delivering fluid is a frictional load as the inertial part is secondary, assuming that the pump runs continuously at a steady speed. The total application torque comprises the frictional plus inertial elements. If the pump runs at a constant speed, it produces a uniform load and the required power would be given in kW or HP. The kW rating is related to torque by the following formula: torque Nm = kW x 9550 divided by revs per min.

Conversely, a ball-mounted slide table, typified by short cycles of rapid acceleration and deceleration in both directions of rotation, will have inertial loads as the predominant factor. These will determine the reversing torque factor of the coupling.

To be more precise, the maximum torque experienced by the coupling may be dictated by whether braking is applied by the load or the motor. In the following diagrams (fig 22, 23, 24) the arrows indicate the direction of the angular forces due to acceleration, deceleration or braking.

Once the maximum torque in the system is known, the selection of the correct coupling can be made by relating it to the **Peak Torque Rating**. This can be found in the relevant table in the Huco catalogue. The coupling should be selected using the following formula: peak torque ≥ application torque x service factor.

Note: The service factor for a non-uniform load is 2. A lower or higher service factor can be incorporated, depending on the service life required.

In the case of Huco couplings, the peak torque rating relates to the static reversing torque load sustained for a minimum of one million cycles under test conditions (zero misalignment).

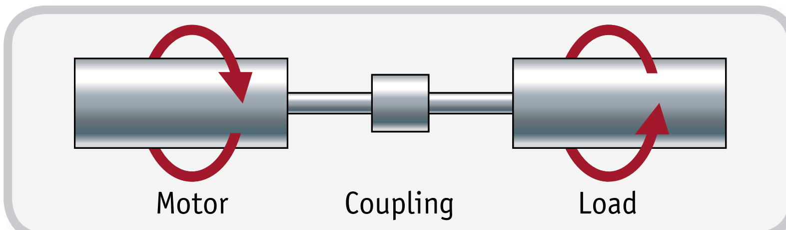


Fig.22 *Mode: motor accelerates load. Torque 'seen' by coupling = load inertia + frictional resistance of load.*

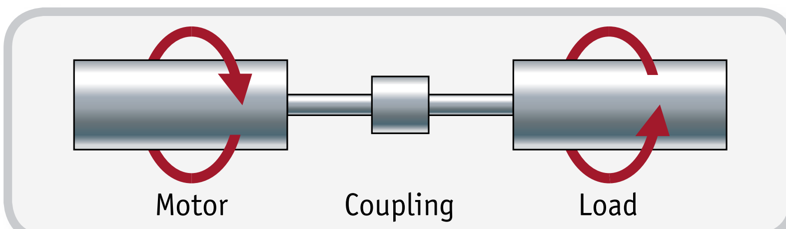


Fig.23 *Mode: Supply to motor discontinued, braking applied to load. Torque 'seen' by coupling = motor inertia – motor drag.*

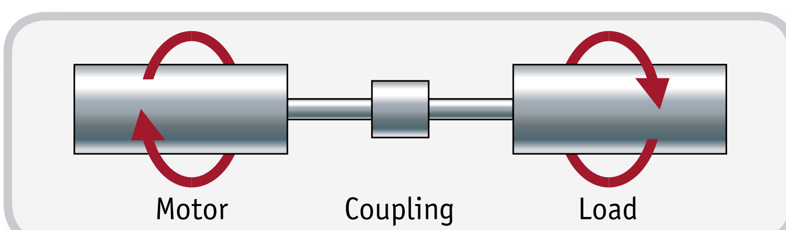


Fig.24 *Mode: Motor decelerates load. Torque 'seen' by coupling (in opposite direction) = load inertia – frictional resistance of load. The coupling 'sees' this as a torque reversal although the direction of rotation is unchanged.*

Selecting for torsional stiffness

Torsional stiffness may be expressed in several different units, but the most common and easiest to work with is Nm/rad. Often described as torque per unit deflection, torsional stiffness is significant in positional systems and describes a coupling's resistance to torsional deflection.

Torsional deflection is the inverse of torsional stiffness and is defined by deflection per unit torque. This also has many denominations but is best expressed in degrees/ Nm. The conversion tables at the back of this booklet allow conversion from other denominations.

When used in a closed loop or velocity control system a coupling's torsional stiffness becomes more critical and forms a contributory factor in calculating the upper limit of dynamic performance and stability. Therefore, the stiffness of a coupling should be such that its torsional resonance frequency exceeds 300 – 600 Hz, depending on dynamics. Stiffness is at its most critical when load inertia is dominant and becomes less so when that dominance swings in the motor's favour. (See figure 34)

Fig.34 *The formulae for torsional stiffness and resonant frequency are:*

$$C_T \geq \frac{(F_R \times 2\pi)^2}{\left(\frac{1}{J_M} + \frac{1}{J_L}\right)}$$

$$F_R \geq \frac{1}{2\pi} \times \sqrt{\left(\frac{1}{J_M} + \frac{1}{J_L}\right) \times C_T}$$

Where

C_T = *torsional stiffness (Nm/rad);*

J_M = *motor inertia (kgm²);*

F_R = *resonant frequency (Hz);*

J_L = *load inertia (kgm²)*

II

II

Selecting for cost, duty and life expectancy

The couplings manufactured by Huco fall into two classifications: mechanical and flexural. The former work through sliding contact, while the latter rely on the flex of the constituent material. The pertinent issues relating to cost, duty and life expectancy are given below.

Oldham or Uni-Lat couplings should be considered when:

1. *Cost is the paramount consideration*
2. *The backlash-free life requirement is within the coupling's backlash-free life expectancy or backlash can be tolerated*
3. *The coupling is expected to transmit only incremental or periodic rotation*
4. *The duty is 50% or less, i.e. the coupling is stationary for half of the time or more*
5. *Radial misalignment is severe and the available space is limited*
6. *Radial misalignment is difficult to predict or maintain*
7. *Slight torsional damping is beneficial*
8. *A three-piece coupling is advantageous. With the Oldham coupling the drive can be connected/ disconnected with the hubs in place. The wear element is renewable*
9. *Electrical isolation of shafts is required*
10. *The coupling is required to transmit longitudinal motion (push/pull)*

Bear in mind that Uni-Lats have a more pronounced damping characteristic, lower torque capacity and generally run more quietly than Oldham couplings. They also have a greater angular misalignment capacity, though this is only useable at low speeds.

Oldham-type couplings, though, are more robust and the replaceable wear element can be supplied in both heat and radiation resistant plastics. The hubs on the standard series are blind bored to a controlled depth, while the X-Y series couplings have through bores and have two-three times the backlash-free life. Torque discs are solid but can be specified with a bore to allow the passage of a shaft, although this will reduce the torsional stiffness of the coupling.

Both Uni-Lat and Oldham general-purpose couplings are suitable for position control. Specifically Uni-Lats are better suited to full and half step motor drives and Oldhams are suited to micro-stepper and closed loop systems.

Huco-Flex disc, bellows or multi beam couplings should be considered when:

1. *Torsional stiffness is a critical parameter*
2. *The backlash-free life requirement is beyond the capacity of the Oldham or Uni-Lat*
3. *Speeds are typically higher than 3000 revs/min*
4. *Rotation is continuous or the duty-cycle exceeds 50%*
5. *A coupling with axial compliance is required to protect fragile bearings from thrust load*
6. *There is little risk of the alignment errors exceeding prescribed limits during initial installation or on subsequent replacement of the motor, encoder, etc.*
7. *The environmental conditions favour an all metal coupling*

Although the life expectancy of the Huco-Flex bellows coupling is not as high as the comparable Huco-Flex disc coupling, size for size it offers the highest torsional stiffness

ratio and provides a high level of translation accuracy. This makes the bellows-type coupling ideally suited to intermittent applications.

Huco-Flex disc couplings have a greater reliability and near infinite life when used within their torque and misalignment ratings. They also provide a high level of translational accuracy and their spring rates can be modified through varying the number and thickness of the stainless steel membranes.

However users must be aware that couplings designed around a flexural system can fail with little or no warning, causing immediate loss of drive. The cause of these failures is due mainly to metal fatigue caused by sustained flexure above the coupling's recommended torque and compliance factors.

Failure in mechanical couplings is more subjective and useful life can vary depending on individual applications.
For instance, in zero backlash applications the coupling is deemed to have failed as soon as backlash is in evidence. In other applications the failure threshold may be 2 degrees of backlash.

II

II

Fig.35 *Apply the following tables to equate operating hours with total revolutions at varying speeds of rotation*

r.p.m.	10^6	10^7	10^8	10^9
60	278 hrs	2,778 hrs	27,778 hrs	277,778 hrs
100	167	1,667	16,667	166.667
250	67	667	6,667	66,667
500	33	333	3,333	33,333
1000	17	167	1667	16,667
1500	11	111	1,111	11,111
2000	8.3	83	833	8,333
3000	5.6	56	556	5,556
4000	4.2	42	417	4,167
5000	3.3	33	333	3,333
7500	2.2	22	222	2,222
10000	1.7	17	167	1,667

Fig.36 *Revolutions per number of operating hours*

r.p.m.	1000 hrs	2000 hrs	4000 hrs	6000 hrs
60	3.6×10^6	7.2×10^6	1.4×10^7	2.2×10^7
100	6×10^6	1.2×10^7	2.4×10^7	$3.6\ 10^7$
250	1.5×10^7	3×10^7	6×10^7	9×10^7
500	3×10^7	6×10^7	1.2×10^8	1.8×10^8
1000	6×10^7	1.2×10^8	2.4×10^8	3.6×10^8
1500	9×10^7	1.8×10^8	3.6×10^8	5.4×10^8
2000	1.2×10^8	2.4×10^8	4.8×10^8	7.2×10^8
3000	1.8×10^8	3.6×10^8	7.2×10^8	1.1×10^9
4000	2.4×10^8	4.8×10^8	9.6×10^8	1.4×10^9
5000	3×10^8	6×10^8	1.2×10^9	1.8×10^9
7500	4.5×10^8	9×10^8	1.8×10^9	2.7×10^9
10000	6×10^8	1.2×10^9	2.4×10^9	3.6×10^9

Tubing Data

Contents

Tubing Selection

Proper selection, handling, and installation of tubing, when combined with proper selection of Swagelok® tube fittings, are essential to reliable tubing systems.

The following variables should be considered when ordering tubing for use with Swagelok tube fittings:

- Surface finish
- Material
- Hardness
- Wall thickness.

Tubing Surface Finish

Many ASTM specifications cover the above requirements, but they often are not very detailed on surface finish. For example, ASTM A450, a general tubing specification, reads:

> 11. Straightness and Finish
>
> 11.1 Finished tubes shall be reasonably straight and have smooth ends free of burrs. They shall have a workmanlike finish. Surface imperfections (Note) may be removed by grinding, provided that a smooth curved surface is maintained, and the wall thickness is not decreased to less than that permitted by this or the product specification. The outside diameter at the point of grinding may be reduced by the amount so removed.
>
> **Note:** An imperfection is any discontinuity or irregularity found in the tube.

Tubing Material

Our suggested ordering instructions for each type of tubing are shown under the respective tables.

Tubing Outside Diameter Hardness

The key to selecting proper tubing for use with metal Swagelok tube fittings is that the tubing must be softer than the fitting material. Swagelok tube fittings are designed to work properly with the tubing that is suggested in the ordering instructions.

Most misunderstandings about tubing hardness are in the area of stainless steel tubing. Swagelok stainless steel tube fittings have been repeatedly tested successfully with tubing with hardness up to Vickers 200 (HV) and 90 HRB.

Although such tubing hardness is permissible and Swagelok tube fittings will perform satisfactorily on such tubing, we suggest that, whenever possible, you specify

- 180 HV (metric) and
- 80 HRB (fractional)

maximum when ordering tubing. Such tubing lowers installed cost because it is more easily bent and installed. Tubing installers should be particularly careful when installing harder tubing, ensuring that the fitting is installed according to the installation and gaugeability instructions in the Swagelok *Gaugeable Tube Fittings and Adapter Fittings* catalog.

Tubing Wall Thickness

The accompanying tables show working pressure ratings of tubing in a wide range of wall thicknesses. Except as noted, allowable pressure ratings are calculated from *S* values as specified by ASME B31.3, Process Piping.

Swagelok tube fittings have been repeatedly tested in both the minimum and maximum wall thicknesses shown.

Swagelok tube fittings are not recommended for tube wall thicknesses outside the ranges shown in the accompanying tables for each size.

Tubing Handling

Good handling practices can greatly reduce scratches on tubing and protect the good surface finish that reliable tube manufacturers supply.

- Tubing should never be dragged out of a tubing rack or across a rough surface.
- Tube cutters or hacksaws should be sharp. Do not take deep cuts with each turn of the cutter or stroke of the saw.
- Tube ends should be deburred. This helps to ensure that the tubing will go all the way through the ferrules without damaging the ferrule sealing edge.

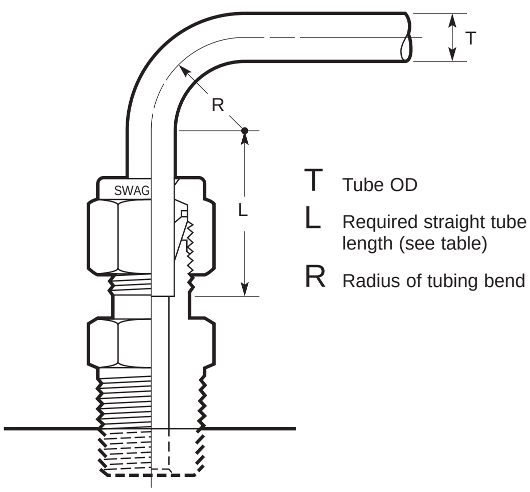
Gas Service

Gases (air, hydrogen, helium, nitrogen, etc.) have very small molecules that can escape through even the most minute leak path. Some surface defects on the tubing can provide such a leak path. As tube outside diameter (OD) increases, so does the likelihood of a scratch or other surface defect interfering with proper sealing.

The most successful connection for gas service will occur if all installation instructions are carefully followed and the heavier wall thicknesses of tubing on the accompanying tables are selected.

A heavy wall tube resists ferrule action more than a thin wall tube, allowing the ferrules to coin out minor surface imperfections. A thin wall tube offers less resistance to ferrule action during installation, reducing the chance of coining out surface defects, such as scratches. Within the applicable suggested allowable working pressure table, select a tube wall thickness whose working pressure is *outside* of the shaded areas.

Tubing Installation



Fractional, in.	
T Tube OD	**L**①
1/16	1/2
1/8	23/32
3/16	3/4
1/4	13/16
5/16	7/8
3/8	15/16
1/2	1 3/16
5/8	1 1/4
3/4	
7/8	1 5/16
1	1 1/2
1 1/4	2
1 1/2	2 13/32
2	3 1/4

① Required straight tube length.

Metric, mm	
T Tube OD	**L**①
3	19
6	21
8	23
10	25
12	31
14	32
15	
16	
18	
20	34
22	
25	40
28	46
30	50
32	54
38	63
50	80

Tubing properly selected and handled, when combined with the quality of Swagelok fittings, will give you leak-tight systems. Properly installed on such tubing, Swagelok fittings provide reliable service under a wide variety of fluid applications.

When installing fittings near tube bends, there must be a sufficient straight length of tubing to allow the tube to be bottomed in the Swagelok fitting (see tables).

For maximum assurance of reliable performance, use Swagelok tube fittings assembled in accordance with catalog instructions, and use properly selected and handled high-quality tubing—such as provided by Swagelok.

Hydraulic Swaging Unit

When installing carbon steel or stainless steel Swagelok tube fittings over 1 in. (25 mm), a Swagelok hydraulic swaging unit must be used. This unit provides sufficient pre-swaging of the ferrules onto the tubing for 1 1/4, 1 1/2 and 2 in. and 28, 30, 32, 38, and 50 mm Swagelok tube fittings. Ask your authorized Swagelok sales and service representative for a demonstration.

Suggested Allowable Pressure Tables

Figure and tables are for reference only. No implication is made that these values can be used for design work. Applicable codes and practices in industry should be considered. ASME Codes are the successor to and replacement of ASA Piping Codes.

- All pressures are calculated from equations in ASME B31.3, Process Piping. See factors for calculating working pressures in accordance with ASME B31.1, Power Piping.
- Calculations are based on maximum OD and minimum wall thickness, except as noted in individual tables.

  Example: 1/2 in. OD × 0.035 in. wall stainless steel tubing purchased to ASTM A269:

  OD Tolerance ± 0.005 in. / Wall Thickness ±10 %

  Calculations are based on 0.505 in. OD × 0.0315 in. wall tubing.
- No allowance is made for corrosion or erosion.

Suggested Allowable Working Pressure for *Carbon Steel Tubing*

Table 1—Fractional Carbon Steel Tubing

Allowable working pressures are calculated from an *S* value of 15 700 psi (108 200 kPa) for ASTM A179 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3. Multiply carbon steel rating by 0.75 for working pressure in accordance with ASME B31.1.

Tube OD in.	Tube Wall Thickness, in.													Swagelok Fitting Series
	0.028	**0.035**	**0.049**	**0.065**	**0.083**	**0.095**	**0.109**	**0.120**	**0.134**	**0.148**	**0.165**	**0.180**	**0.220**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)													
1/8	8000	10 200												200
3/16	5100	6 600	9600											300
1/4	3700	4 800	7000	9600										400
5/16		3 700	5500	7500										500
3/8		3 100	4500	6200										600
1/2		2 300	3200	4500	5900									810
5/8		1 800	2600	3500	4600	5300								1010
3/4			2100	2900	3700	4300	5100							1210
7/8			1800	2400	3200	3700	4300							1410
1			1500	2100	2700	3200	3700	4100						1610
1 1/4				1 600	2100	2500	2900	3200	3600	4000	4600	5000		2000
1 1/2					1800	2000	2400	2600	2900	3300	3700	4100	5100	2400
2						1500	1700	1900	2100	2400	2700	3000	3700	3200

Suggested Ordering Information

High-quality, soft annealed seamless carbon steel hydraulic tubing ASTM A179 or equivalent. Hardness 72 HRB (130 HV) or less. Tubing to be free of scratches, suitable for bending and flaring.

Table 2—Metric Carbon Steel Tubing

Allowable working pressures are based on equations from ASME B31.3 for DIN 2391 tubing, using a stress value of 1130 bar (16 400 psi) and tensile strength of 3400 bar (49 300 psi).

Tube OD mm	Tube Wall Thickness, mm													Swagelok Fitting Series
	0.8	**1.0**	**1.2**	**1.5**	**1.8**	**2.0**	**2.2**	**2.5**	**2.8**	**3.0**	**3.5**	**4.0**	**4.5**	
	Working Pressure, bar **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)													
3	630	790												3M0
6	290	370	460	590										6M0
8		270	330	430										8M0
10		210	260	330										10M0
12		170	210	270	330	380	420							12M0
14		150	180	230	280	320	350							14M0
15		140	170	210	260	290	330							15M0
16		130	150	200	240	270	300	350						16M0
18			140	170	210	240	270	310						18M0
20			120	160	190	210	240	270	310					20M0
22			110	140	170	190	210	240	280					22M0
25			100	120	150	170	180	210	240	260				25M0
28						150	160	190	210	230	270			28M0
30						140	150	170	200	210	250			30M0
32						130	140	160	180	200	230	270		32M0
38							120	130	150	160	190	230	260	38M0

Suggested Ordering Information

High-quality, soft annealed carbon steel tubing to DIN 2391 or equivalent. Hardness 130 HV (72 HRB) or less. Tubing to be free of scratches, suitable for bending or flaring.

II

Suggested Allowable Working Pressure for *Stainless Steel Tubing*

Table 3—Fractional Stainless Steel Seamless Tubing

Allowable working pressures are calculated from an *S* value of 20 000 psi (137 800 kPa) for ASTM A269 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3, except as noted. Multiply stainless steel rating by 0.94 for working pressure in accordance with ASME B31.1.

For Welded Tubing

For welded and drawn tubing, a derating factor must be applied for weld integrity:

- for double-welded tubing, multiply pressure rating by 0.85
- for single-welded tubing, multiply pressure rating by 0.80.

Tube OD in.	Tube Wall Thickness, in.																Swagelok Fitting Series
	0.010	0.012	0.014	0.016	0.020	0.028	0.035	0.049	0.065	0.083	0.095	0.109	0.120	0.134	0.156	0.188	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)																
1/16	5600	6800	8100	9400	12 000												100
1/8						8500	10 900										200
3/16						5400	7 000	10 200									300
1/4						4000	5 100	7 500	10 200②								400
5/16							4 000	5 800	8 000								500
3/8							3 300	4 800	6 500	7500①②							600
1/2							2 600	3 700	5 100	6700							810
5/8								2 900	4 000	5200	6000						1010
3/4								2 400	3 300	4200	4900	5800					1210
7/8								2 000	2 800	3600	4200	4800					1410
1									2 400	3100	3600	4200	4700				1610
1 1/4										2400	2800	3300	3600	4100	4900		2000
1 1/2											2300	2700	3000	3400	4000	4900	2400
2												2000	2200	2500	2900	3600	3200

① Rating based on repeated pressure testing of the Swagelok tube fitting with a 4:1 design factor based upon hydraulic fluid leakage.

② For higher pressures and tubing with heavier wall thicknesses, see the Swagelok *High-Pressure Fittings* catalog.

Suggested Ordering Information

Fully annealed, high-quality (Type 304, 316, etc.) (seamless or welded and drawn) stainless steel hydraulic tubing ASTM A269 or A213, or equivalent. Hardness 80 HRB (180 HV) or less. Tubing to be free of scratches, suitable for bending and flaring.

Note: Certain austenitic stainless tubing has an allowable ovality tolerance double the OD tolerance and may not fit into Swagelok precision tube fittings.

Table 4—Metric Stainless Steel Seamless Tubing

Allowable working pressures are based on equations from ASME B31.3 for EN ISO 1127 tubing (D4, T4 tolerance for 3 to 12 mm; D4, T3 tolerance 14 to 50 mm), using a stress value of 1370 bar (20 000 psi) and tensile strength of 5170 bar (75 000 psi), except as noted. Multiply stainless steel rating by 0.94 for working pressure in accordance with ASME B31.1.

For Welded Tubing

For welded and drawn tubing, a derating factor must be applied for weld integrity:

- for double-welded tubing, multiply pressure rating by 0.85
- for single-welded tubing, multiply pressure rating by 0.80.

Tube OD mm	Tube Wall Thickness, mm														Swagelok Fitting Series
	0.8	1.0	1.2	1.5	1.8	2.0	2.2	2.5	2.8	3.0	3.5	4.0	4.5	5.0	
	Working Pressure, bar **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)														
3	670														3M0
6	310	420	540	710											6M0
8		310	390	520											8M0
10		240	300	400	510	580									10M0
12		200	250	330	410	470									12M0
14		160	200	270	340	380	430								14M0
15		150	190	250	310	360	400								15M0
16			170	230	290	330	370	400①							16M0
18			150	200	260	290	320	370							18M0
20			140	180	230	260	290	330	380						20M0
22			120	160	200	230	260	300	340						22M0
25					180	200	230	260	290	320					25M0
28						180	200	230	260	280	330				28M0
30						170	180	210	240	260	310				30M0
32						160	170	200	220	240	290	330			32M0
38							140	160	190	200	240	270	310		38M0
50										150	180	210	240	270	50M0

① Rating based on repeated pressure testing of the Swagelok tube fitting with a 4:1 design factor based upon hydraulic fluid leakage.

Suggested Ordering Information

Fully annealed, high-quality (Type 304, 316, etc.) stainless steel tubing to EN ISO 1127 or equivalent. Hardness 180 HV (80 HRB) or less. Tubing to be free of scratches, suitable for bending or flaring.

Suggested Allowable Working Pressure for *Copper Tubing*

Table 5—Fractional Copper Tubing

Allowable working pressures are calculated from an *S* value of 6000 psi (41 300 kPa) for ASTM B75 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3.

Tube OD in.	Tube Wall Thickness, in.									Swagelok Fitting Series
	0.028	0.035	0.049	0.065	0.083	0.095	0.109	0.120	0.134	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)									
1/8	2700	3600								200
3/16	1800	2300	3400							300
1/4	1300	1600	2500	3500						400
5/16		1300	1900	2700						500
3/8		1000	1600	2200						600
1/2		800	1100	1600	2100					810
5/8			900	1200	1600	1900				1010
3/4			700	1000	1300	1500	1800			1210
7/8			600	800	1100	1300	1500			1410
1			500	700	900	1100	1300	1500		1610
1 1/8				600	800	1000	1100	1300	1400	1810

II

Suggested Ordering Information

Fractional—high-quality, soft annealed seamless copper tubing ASTM B75 or equivalent. Also soft annealed (Temper O) copper water tube, type K or type L to ASTM B88.

Suggested Allowable Working Pressure for *Aluminum Tubing*

Table 6—Fractional Aluminum Tubing

Allowable working pressures are calculated from an *S* value of 14 000 psi (96 500 kPa) for ASTM B210, Type 6061-T6 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3. Multiply aluminum rating by 0.75 for working pressure in accordance with ASME B31.1.

Tube OD in.	Tube Wall Thickness, in.					Swagelok Fitting Series
	0.035	0.049	0.065	0.083	0.095	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)					
1/8	8600					200
3/16	5600	8000				300
1/4	4000	5900				400
5/16	3100	4600				500
3/8	2600	3700				600
1/2	1900	2700	3700			810
5/8	1500	2100	2900			1010
3/4		1700	2400	3100		1210
7/8		1500	2000			1410
1		1300	1700	2300	2700	1610

Suggested Ordering Information

High-quality aluminum alloy drawn seamless tubing ASTM B210 (Type 6061-T6) or equivalent.

Suggested Allowable Working Pressure for *Additional Alloys*

A limited amount of test data is available on Swagelok tube fittings used with special alloy tubing. For sizes not listed in the following tables, we recommend that a sample of the tubing be provided for evaluation before installation. Please include all pertinent information relating to system parameters. Give tubing sample to your authorized Swagelok representative to forward to the factory.

Table 7—Fractional Alloy 400 Tubing

Allowable working pressures are calculated from an *S* value of 18 700 psi (128 800 kPa) for ASTM B165 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3. Multiply alloy 400 rating by 0.93 for working pressure in accordance with ASME B31.1.

II

Tube OD in.	**Tube Wall Thickness**, in.								Swagelok Fitting Series
	0.028	**0.035**	**0.049**	**0.065**	**0.083**	**0.095**	**0.109**	**0.120**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)								
1/8	7900	10 100							200
1/4	3700	4 800	7000	9500					400
3/8		3 100	4400	6100					600
1/2		2 300	3200	4400					810
3/4			2200	3000	4000	4600			1210
1				2200	2900	3400	3900	4300	1610

Suggested Ordering Information

Fractional—Fully annealed, quality seamless alloy 400 hydraulic tubing ASTM B165 or equivalent. Hardness 75 HRB maximum. Tubing to be free of scratches, suitable for bending and flaring.

Table 8—Fractional Alloy C-276 Tubing

Allowable working pressures are based on equations from ASME B31.3 and a maximum *S* value of 20 000 psi (137 800 kPa).

Tube OD in.	**Tube Wall Thickness**, in.				Swagelok Fitting Series
	0.028	**0.035**	**0.049**	**0.065**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)				
1/4	4000	5100	7500	10 200	400
3/8		3300	4800	6 500	600
1/2		2600	3700	5 100	810

Suggested Ordering Information

Fully annealed quality alloy C-276 tubing ASTM B622 or equivalent. Hardness 100 HRB maximum. Tubing to be free of scratches, suitable for bending and flaring. OD tolerances not to exceed ± 0.005 in.

Table 9—Fractional Alloy 20 Tubing

Allowable working pressures are based on equations from ASME B31.3 and a maximum *S* value of 20 000 psi (137 800 kPa).

Tube OD in.	**Tube Wall Thickness**, in.				**Swagelok Fitting Series**
	0.028	**0.035**	**0.049**	**0.065**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)				
1/4	4000	5100	7500	10 200	400
3/8		3300	4800	6 500	600
1/2		2600	3700	5 100	810

Suggested Ordering Information

Fully annealed, seamless or welded and drawn alloy 20 alloy tubing, ASTM B729, B468 or equivalent. Hardness 95 HRB or less. Tubing to be free of scratches, suitable for bending and flaring. OD tolerances not to exceed ± 0.005 in.

II

Table 10—Fractional Alloy 600 Tubing

Allowable working pressures are based on equations from ASME B31.3 and a maximum *S* value of 20 000 psi (137 800 kPa).

Tube OD in.	**Tube Wall Thickness**, in.				**Swagelok Fitting Series**
	0.028	**0.035**	**0.049**	**0.065**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)				
1/4	4000	5100	7500	10 200	400
3/8		3300	4800	6 500	600
1/2		2600	3700	5 100	810

Suggested Ordering Information

Cold drawn, fully annealed, #1 temper alloy 600 seamless alloy tubing, ASTM B167 or equivalent. Hardness 92 HRB or less. Tubing to be free of scratches, suitable for bending and flaring. Order to outside diameter and wall thickness only, not to inside diameter, average wall specification. OD tolerances not to exceed ± 0.005 in.

Table 11—Fractional Grade 2 Titanium Tubing

Allowable working pressures are calculated from an *S* value of 16 700 psi (115 000 kPa) for ASTM B338 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3. Multiply grade 2 titanium rating by 0.75 for working pressure in accordance with ASME B31.1.

Tube OD in.	**Tube Wall Thickness**, in.				**Swagelok Fitting Series**
	0.028	**0.035**	**0.049**	**0.065**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)				
1/4	3500	4500	6700	9100	400
3/8		2900	4200	5800	600
1/2		2100	3100	4200	810

Suggested Ordering Information

Fully annealed seamless or welded and drawn grade 2 titanium tubing, ASTM B338 or equivalent. Tubing to be free of scratches, suitable for bending. OD tolerances not to exceed ± 0.005 in.

Table 12—Fractional SAF 2507 Super Duplex Tubing

Allowable working pressures are calculated from an *S* value of 38 700 psi (266 000 kPa) for ASTM A789 tubing at –20 to 100°F (–28 to 37°C), as listed in ASME B31.3. For tubing suitable for SAF 2507 super duplex weld fittings with working pressures calculated based on ASME B31.3 Chapter IX, see the Swagelok *SAF 2507 Super Duplex Weld Fittings* catalog.

Tube OD in.	Tube Wall Thickness, in.						Swagelok Fitting Series
	0.028	**0.035**	**0.049**	**0.065**	**0.083**	**0.095**	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)						
1/4	7800	10 000	15 000①				400
3/8		6 500	10 100①	12 700			600
1/2		5 000	7 200	10 100①	12 900		810
5/8			5 800	7 600	10 100		1010
3/4			4 700	6 300	8 500①	10 000①	1210

① Pressure ratings based on special wall thickness tolerance for Swagelok SAF 2507 tubing.

Suggested Ordering Information

Fully annealed SAF 2507 super duplex tubing, ASTM A789 or equivalent. Hardness 32 HRC or less. Tubing to be free of scratches, suitable for bending and flaring.

Pressure Ratings at Elevated Temperatures

Table 13—Elevated Temperature Factors

Temperature		Tubing Materials										
°F	**°C**	**Aluminum**	**Copper**	**Carbon Steel①**	**304 SS**	**316 SS**	**Alloy 400**	**Alloy 20②**	**Alloy C-276②**	**Alloy 600②**	**Titanium**	**SAF 2507**
200	93	1.00	0.80	0.95	1.00	1.00	0.87	1.00	1.00	1.00	0.86	0.90
400	204	0.40	0.50	0.87①	0.93	0.96	0.79	0.96	0.96	0.96	0.61	0.82
600	315				0.82	0.85	0.79	0.85	0.85	0.85	0.45	0.80
800	426				0.76	0.79	0.75	0.79	0.79	0.79		
1000	537				0.69	0.76			0.76	0.35		

① Based on 375°F (190°C) max.
② Based on the lower derating factor for stainless steel, in accordance with ASME B31.3.

To determine allowable working pressure at elevated temperatures, multiply allowable working pressures from Tables 1 through 12 by a factor shown in Table 13.

Example: Type 316 stainless steel 1/2 in. OD × 0.035 in. wall at 1000°F

1. The allowable working pressure at –20 to 100°F (–28 to 37°C) is 2600 psig (Table 3, page 4).
2. The elevated temperature factor for 1000°F (537°C) is 0.76 (Table 13, above):

   2600 psig × 0.76 = 1976 psig

The allowable working pressure for 316 SS 1/2 in. OD × 0.035 in. wall tubing at 1000°F (537°C) is 1976 psig.

Safe Product Selection
When selecting a product, the total system design must be considered to ensure safe, trouble-free performance. Function, material compatibility, adequate ratings, proper installation, operation, and maintenance are the responsibilities of the system designer and user.

Unbrako®

Engineering Guide

II

Inch & Metric

A comprehensive catalog of
UNBRAKO® socket screws and related products

In this catalog you will find complete information about UNBRAKO socket screws and such related products as shoulder screws, dowel pins, pressure plugs and hex keys. Everything you need to select, specify and order these precision products is at your finger tips except actual prices. Furthermore, all data has been organized to let you find the facts you want with the greatest speed and the least effort. Wherever possible, all data for a particular product is presented in a two-page spread for your convenience.

Included in this catalog are:

- UNBRAKO fastener product descriptions
- Features and technical data about each product
- Technical discussions for application and use

II For prices of stock items, see current UNBRAKO fastener price lists or call your local UNBRAKO fastener distributor.
For non-stock items, consult your UNBRAKO fastener distributor, or contact the UNBRAKO Engineered Fastener Group by phone at 216-581-3000 or by fax on 800-225-5777,or Internet at http://www.spstech.com.

Commercial and Government Entity (CAGE) Code 71838

IMPORTANT

Referenced consensus standards can change over time. UNBRAKO products are manufactured in accordance with revisions valid at time of manufacture.

This guide refers to products and sizes that may not be manufactured to stock. Please consult an UNBRAKO distributor or UNBRAKO to determine stock status.

The technical discussions represent typical applications only.
The use of the information is at the sole discretion of the reader. Because applications vary enormously, UNBRAKO does not warrant the scenarios described are appropriate for any specific application. The reader must consider all variables prior to using this information.

Products modified other than by UNBRAKO are not guaranteed and not subject to return.

LIMITED WARRANTY AND EXCLUSIVE REMEDY

SPS Technologies, Inc., through its Unbrako Division warrants that these product conform to industry standards specified herein and will be free from defects in materials and workmanship. THIS WARRANTY IS EXPRESSLY GIVEN IN LIEU OF ANY AND ALL OTHER EXPRESS OR IMPLIED WARRANTIES, INCLUDING ANY IMPLIED WARRANTY OF MERCHANTABILITY OR FITNESS FOR A PARTICULAR PURPOSE, AND IN LIEU OF ANY OTHER OBLIGATION ON THE PART OF SPS TECHNOLOGIES, INC. SPS Technologies, Inc. will, at its option, repair or replace free of charge (excluding all shipping all shipping and handling costs) any products which have not been subject to misuse, abuse, or modification and which in its sole determination were not manufactured in compliance with the warranty given above.

THE REMEDY PROVIDED FOR HEREIN SHALL BE THE EXCLUSIVE REMEDY FOR ANY BREACH OF WARRANTY OR ANY CLAIM ARISING IN ANY WAY OUT OF THE MANUFACTURE, SALE, OR USE OF THESE PRODUCTS. In no event shall SPS Technologies, Inc. be liable for consequential, incidental or any other damages of any nature whatsoever except those specifically provided herein for any breach of warranty or any claim arising in any way out of the manufacture, sale, or use of these products. No other person is authorized by SPS Technologies, Inc. to give any other warranty, written or oral, pertaining to the products.

TABLE OF CONTENTS

UNBRAKO® Socket Screw Products

II

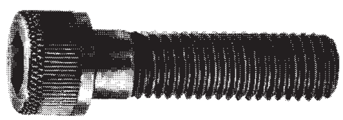
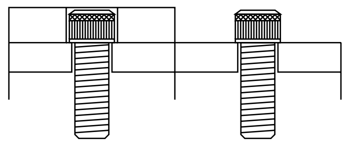
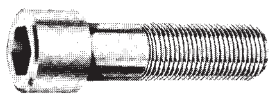
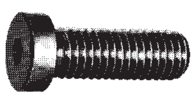
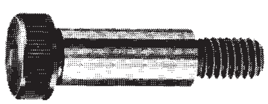
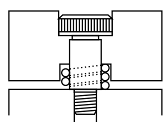
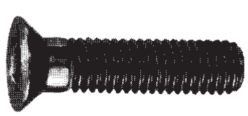
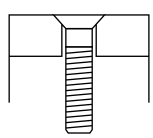
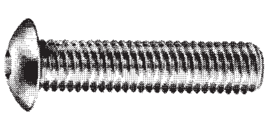
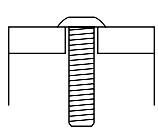
NOTE: The proper tightening of threaded fasteners can have a significant effect on their performance.

Many application problems such as self-loosening and fatigue can be minimized by adequate tightening.

The recommended seating torques listed in the catalog tables serve as guidelines only.

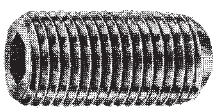
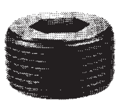
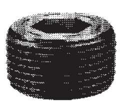
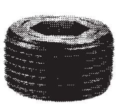
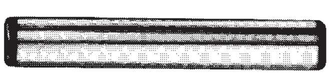
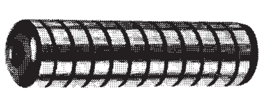
Even when using the recommended seating torques, the induced loads obtained may vary as much as ±25% depending upon the uncontrolled variables such as mating material, lubrication, surface finish, hardness, bolt/joint compliance, etc.

II

TYPES		APPLICATIONS/FEATURES	PERFORMANCE (See Note 1) tensile psi (room temp.)	PERFORMANCE (See Note 1) 10^7 cycle dynamic fatigue (psi)	PERFORMANCE (See Note 1) operating temperatures (unplated)	page
Socket Head Cap Screws 1960 Series Alloy Steel		COUNTERBORED PROTRUDING  Use alloy for maximum tensiles; up to 190,000 psi, highest of any socket cap screw	190,000 180,000	20,000	550°F	4–10
Socket Head Cap Screws 1960 Series Stainless Steel		Use stainless for corrosive, cryogenic or elevated temperature environments, hygienic cleanliness.	95,000	30,000	800°F	4–10
Socket Head Cap Screws Low Head Series		Use in parts too thin for standard height heads and where clearance is limited	170,000	20,000	550°F	11
Shoulder Screws		 Tool and die industry standards; also replace costly special parts-shafts, pivots, clevis pins, guides, trunnion mountings, linkages, etc.	**heat treat level psi** 160,000	**shear strength in psi** 96,000	550°F	12–13
Flat Head Socket Screws Alloy/ Stainless		 Uniform, controlled 82° underhead angle for maximum flushness and side wall contact; non-slip hex socket prevents marring of material	160,000	96,000	550°F 800°F	14, 16
Button Head Cap Screws Alloy/ Stainless		 Low heads streamline design, use in materials too thin to countersink; also for non-critical loading requiring heat treated screws	160,000	96,000	550°F 800°F	15–16

NOTE 1: Performance data listed are for standard production items only. Non-stock items may vary due to variables in methods of manufacture. It is suggested that the user verify performance on any non-standard parts for critical applications.

INCH QUICK SELECTOR GUIDE

TYPES		APPLICATIONS/FEATURES	PERFORMANCE (See Note 1) hardness	operating temperatures (unplated)		page
Square Head Set Screws		Half-dog or self-locking cup points only. Use where maximum tightening torques are required	Rc 45 (min.)	450° F		17
Socket Set Screws Alloy Steel		Fasten collars, sheaves, gears, knobs on shafts. Locate machine parts. Cone, half-dog, flat, oval, cup and self-locking cup points standard	Rc 45-53	450°F		18–23
Socket Set Screws Stainless Steel		Use stainless for corrosive, cryogenic or elevated temperatures environments. Plain cup point standard. Other styles on special order	Rb96-Rc33	800°F		18–23
Pressure Plugs 3/4″ Taper Dryseal		Features common to 3/4″ and 7/8″ tapers: Dryseal threads for positive seal without sealing compound; controlled chamfer for faster starting	Rc 34-40 Rb 82 Typical	550°F 800°F 400°F Brass		24, 26
7/8″ Taper LEVL-SEAL® Pressure Plug		LEVL-SEAL® plug features: controlled 7/8″ tape in 3/4″ taper hole seats plug level, flush with surface within 1/2 pitch. LEVL-SEAL plug is an UNBRAKO original	Rc 35-40 Rb 82 Typical	550°F 400°F Brass		25–27
PTFE/ TEFLON Coated**		PTFE/TEFLON coated plugs seal at 60% lower seating torques without tape or compound; install faster at lower cost; smaller sizes can be power installed; LEVL-SEAL plug type for 100% flush seating	Rc 35-40	450°F (uncoated)		26–27
Hex Keys		Tough, ductile, for high torquing; accurate fit in all types socket screws; size marked for quick identity	Rc 47-57	torsional shear in-lb. min. 1.2 to 276.000		32–33
				calculated shear psi	**surface**	
Dowel Pins (Standard)		Formed ends, controlled heat treat; close tolerances; standard for die work; also used as bearings, gages, precision parts, etc.	core: Rc 50-58	150,000	8 micro-inch (max)	28–29
Dowel Pins Pull-Out Type		For use in blind holes. Easily removed without special tools. Reusable, Save money. No need for knock-out holes. Same physicals, finish, accuracy and tolerances as standard UNBRAKO dowel pins.	surface: Rc 60 (min.)	150,000	8 micro-inch (max)	30–31

NOTE 1: Performance data listed are for standard production items only.
Non-stock items may vary due to variables in methods of manufacture.
It is suggested that the user verify performance on any non-standard parts for critical applications.

SOCKET HEAD CAP SCREWS. . . Why Socket Screws? Why UNBRAKO?

II

The most important reasons for the increasing use of socket head cap screws in industry are safety, reliability and economy. All three reasons are directly traceable to the superior performance of socket screws vs. other fasteners, and that is due to their superior strength and advanced design.

- Reliability, higher pressures, stresses and speeds in todays machines and equipment demand stronger, more reliable joints and stronger, more reliable fasteners to hold them together.
- Rising costs make failure and downtime intolerable. Bigger, more complex units break down more frequently despite every effort to prevent it.
- This is why the reliability of every component has become critical. Components must stay together to function properly, and to keep them together joints must stay tight.
- Joint reliability and safety with maximum strength and fatigue resistance. UNBRAKO socket cap screws offer this to a greater degree than any other threaded fastener you can purchase "off-the-self."
- UNBRAKO socket cap screws offer resistance to a greater degree than any other threaded fasteners you can purchase "off-the-shelf."

TENSILE STRENGTH

- U.S. standard alloy steel socket head cap screws are made to strength levels of 180,000 and 170,000 psi to current industry standards. However, UNBRAKO socket cap screws are consistently maintained at 190,000 and 180,000 psi (depending on screw diameter).
- The higher tensile strength of UNBRAKO socket screws can be translated into savings. Using fewer socket screws of the same size can achieve the same clamping force in the joint. A joint requiring twelve 1-3/8" Grade 5 hex heads would need only 7 UNBRAKO socket head cap screws. Use them size for size and there are fewer holes to drill and tap and fewer screws to buy and handle. Smaller diameter socket head cap screws vs. larger hex screws cost less to drill and tap, take less energy to drive, and there is also weight saving.
- The size of the component parts can be reduced since the cylindrical heads of socket screws need less space than hex heads and require no additional wrench space.

FATIGUE STRENGTH

- Joints that are subject to external stress loading are susceptible to fatigue failure. UNBRAKO socket screws have distinct advantages that give you an extra bonus of protection against this hazard.
- Three major factors account for the greater fatigue resistance of UNBRAKO socket screws – design improvements, mechanical properties and closely controlled manufacturing processes.

AUSTENITIC STAINLESS STEEL STANDARD SERIES

UNBRAKO stainless socket screws are made from austenitic stainless steel. UNBRAKO stainless screws offer excellent resistance to rust and corrosion from acids, organic substances, salt solutions and atmospheres. Superior properties attained with stainless steel include retention of a high percentage of tensile strength and good creep resistance up to 800°F. without scaling or oxidation, and good shock and impact resistance to temperatures as low as –300°F.

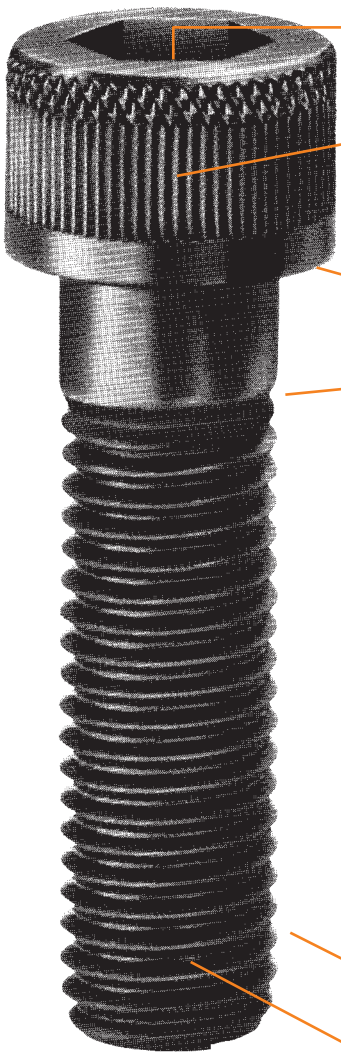
non-magnetic – Valuable in certain electrical applications. Maximum permeability is 1.2 Can be reduced to 1.02 by bright annealing.

cleanliness – Corrosion resistant characteristics of UNBRAKO screws are useful in chemical, food processing, appliance, paper, textile, packaging and pharmaceutical industries, as well as laboratories, hospitals, etc.

eye-appeal – Bright, non-tarnishing qualities add to appearance and salability of many products; are valuable assets to designers.

Standard processing of UNBRAKO stainless steel socket screws includes a passivation surface treatment which removes any surface contaminations.

PROFILE OF EXTRA STRENGTH



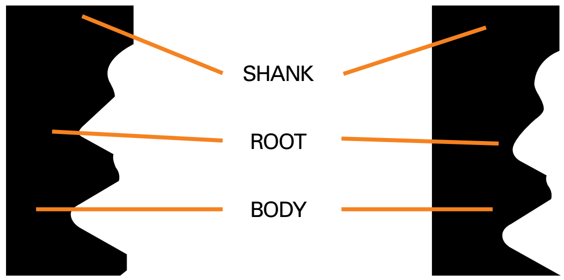
Deep, accurate socket for high torque wrenching. Knurls for easier handling. Marked for easier identification.

Head with increased bearing area for greater loading carrying capacity. Precision forged for symmetrical grain flow, maximum strength.

Elliptical fillet doubles fatigue life at critical head-shank juncture.

II

"3-R" (radiused-root runout) increases fatigue life in this critical head-shank juncture.

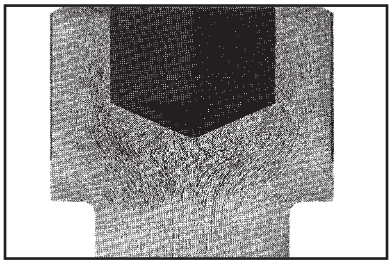


CONVENTIONAL THREAD RUNOUT – Note sharp angle at root where high stress concentration soon develops crack which penetrates into body of the screw.

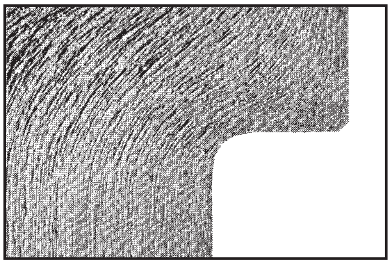
UNBRAKO "3-R" (RADIUSED ROOT RUNOUT) THREAD – Controlled radius of runout root provides a smooth form that distributes stress and increases fatigue life of thread run-out as a much as 300% in certain sizes.

Fully formed radiused thread increases fatigue life 100% over flat root thread forms.

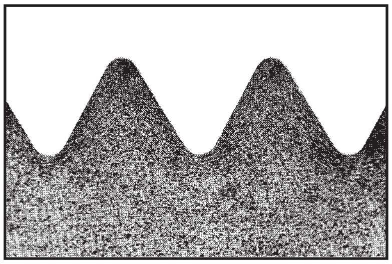
Controlled heat treatment produces maximum strength without brittleness.



Accurate control of socket depth gives more wrench engagement than other screws, permits full tightening without cracking or reaming the socket, yet provides ample metal in the crucial fillet area for maximum head strength.



Controlled head forging, uniform grain flow, unbroken flow lines; makes heads stronger; minimizes failure in vital fillet area; adds to fatigue strength.



Contour-following flow lines provide extra shear strength in threads, resist stripping and provide high fatigue resistance. The large root radius UNBRAKO socket screw development doubles fatigue life compared to flat root thread forms.

II

UNBRAKO Metric Fasteners

UNBRAKO Metric Fasteners are the strongest off-the-shelf threaded fasteners you can buy. Their exclusive design features and closely controlled manufacturing processes insure the dimensional accuracy, strength and fatigue resistance needed for reliability in today's advanced technology. They are manufactured with the same methods and features as their inch-series counterpart.

Strength

UNBRAKO metric socket head cap screws are made into property class 12.9 with a minimum ultimate tensile strength of 1300 or 1250 MPa depending on screw diameter. Precision in manufacturing and careful control in stress areas insure strength in such critical areas as heads, sockets, threads, fillets, and bearing areas.

When you purchase UNBRAKO metric socket screw products, you can be sure that they meet or exceed the strength levels of all current standards, including the three most common-ANSI, ISO and DIN. Unbrako is represented on several ASME, ANSI, ASTM and ISO committees.

- ANSI (American National Standards Institute) documents are published by ASME (The American Society of Mechanical Engineers) and are familiar to almost all users of socket screw products in the U.S.A.
- ASTM (American Society for Testing and Materials). Many ANSI documents list dimensional information but refer to ASTM specifications for materials, mechanical properties, and test criteria.
- ISO (International Standards Organization) is a standards group comprising 70 member nations. Its objective is to provide standards that will be completely universal and common to all countries subscribing.
- DIN (Deutsche Industries Normen) is the German standards group.

NOTE: The proper tightening of threaded fasteners can have a significant effect on their performance.

A WARNING TO METRIC FASTENER USERS

Metric socket cap screws are NOT sold in a single strength level like U.S. inch socket screws.

II

Property Class	General Material	Strength Level, UTS min. MPa (KSI)
	International Standards Organization, ISO	
Property Class 8.8	Carbon Steel	800 (116) < M16 830 (120) ≥ M16
Property Class 10.9	Alloy Steel	1040 (151)
Property Class 12.9	Alloy Steel	1220 (177)
USA Standards ASTM A574M	Alloy Steel	1220 (177)
Unbrako Standards ASTM A574M	Alloy Steel	1300 (189) ≤ M16 1250 (181) > M16

STANDARDS

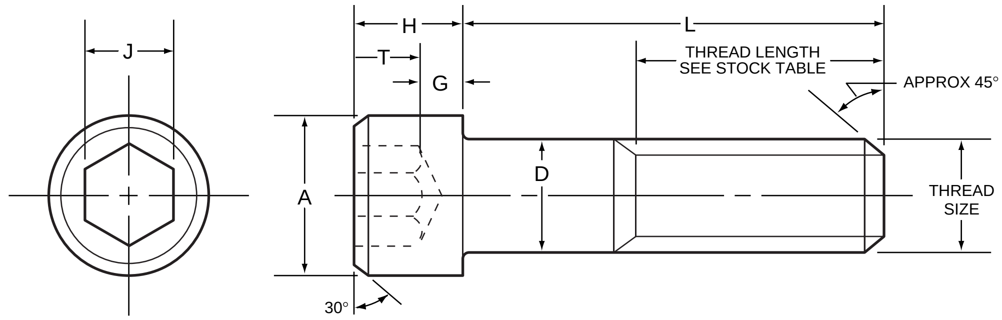
The use of metric fasteners in the worldwide market has led to the creation of many standards. These standards specify the fastener requirements: dimensions, material, strength levels, inspection, etc. Different standards are the responsibility of various organizations and are not always identical. Unbrako supplies metric fasteners for maximum interchangeability with all standards. This Engineering Guide was published with the most current values, which are however subject to change by any standards organization at any time.

METRIC SOCKET HEAD CAP SCREWS

Dimensions

Threads: ANSI B1.13M, ISO 261, ISO 262 (coarse series only)

Property Class: 12.9-ISO 898/1



II

NOTES

1. **Material:** ASTM A574M, DIN912-alloy steel
2. **Hardness:** Rc 38-43
3. **Tensile Stress:** 1300 MPa thru M16 size.
   1250 MPa over M16 size.
4. **Yield Stress:** 1170 MPa thru M16 size.
   1125 MPa over M16 size.
5. **Thread Class:** 4g 6g

LENGTH TOLERANCE

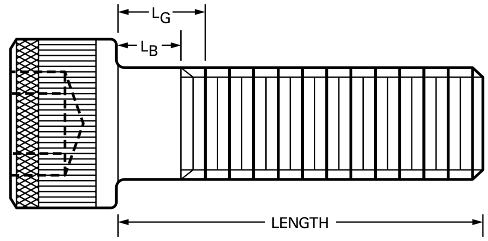
nominal screw length	nominal screw diameter		
	M1.6 thru M10	M12 thru M20	over 20
	tolerance on lgth., mm		
Up to 16 mm, incl.	±0.3	±0.3	–
Over 16 to 50 mm, incl.	±0.4	±0.4	±0.7
Over 50 to 120 mm, incl.	±0.7	±1.0	±1.5
Over 120 to 200 mm, incl.	±1.0	±1.5	±2.0
Over 200 mm	±2.0	±2.5	±3.0

DIMENSIONS / MECHANICAL PROPERTIES / APPLICATION DATA

thread size nom.	pitch	A max.	D max.	H max.	J nom.	G min.	T min.	UTS min. MPa	tensile strength min. kN	tensile strength min. lbs.	single shear strength of body min. kN	single shear strength of body min. lbs.	recommended ** seating torque plain finish N-m	recommended ** seating torque plain finish in-lbs.
M1.6	0.35	3.0	1.6	1.6	1.5	0.54	0.80	1300	1.65	370	1.57	352.5	0.29	2.6
M2	0.40	3.8	2.0	2.0	1.5	0.68	1.0	1300	2.69	605	2.45	550	0.60	5.3
M2.5	0.45	4.5	2.5	2.5	2.0	0.85	1.25	1300	4.41	990	3.83	860	1.21	11
M3	0.5	5.5	3.0	3.0	2.5	1.02	1.5	1300	6.54	1,470	5.5	1240	2.1	19
M4	0.7	7.0	4.0	4.0	3.0	1.52	2.0	1300	11.4	2,560	9.8	2,205	4.6	41
M5	0.8	8.5	5.0	5.0	4.0	1.90	2.5	1300	18.5	4,160	15.3	3,445	9.5	85
M6	1.0	10.0	6.0	6.0	5.0	2.28	3.0	1300	26.1	5,870	22.05	4,960	16	140
M8	1.25	13.0	8.0	8.0	6.0	3.2	4.0	1300	47.6	10,700	39.2	8,800	39	350
M10	1.5	16.0	10.0	10.0	8.0	4.0	5.0	1300	75.4	17,000	61	13,750	77	680
M12	1.75	18.0	12.0	12.0	10.0	4.8	6.0	1300	110	24,700	88	19,850	135	1,200
*(M14)	2.0	21.0	14.0	14.0	12.0	5.6	7.0	1300	150	33,700	120	27,000	215	1,900
M16	2.0	24.0	16.0	16.0	14.0	6.4	8.0	1300	204	45,900	157	35,250	330	2,900
M20	2.5	30.0	20.0	20.0	17.0	8.0	10.0	1250	306	68,800	235.5	53,000	650	5,750
M24	3.0	36.0	24.0	24.0	19.0	9.6	12.0	1250	441	99,100	339	76,500	1100	9,700
*M30	3.5	45.0	30.0	30.0	22.0	12.0	15.0	1250	701	158,000	530	119,000	2250	19,900
*M36	4.0	54.0	36.0	36.0	27.0	14.4	18.0	1250	1020	229,000	756	171,500	3850	34,100
*M42	4.5	63.0	42.0	42.0	32.0	16.8	21.0	1250	1400	315,000	1040	233,500	6270	55,580
*M48	5.0	72.0	48.0	48.0	36.0	19.2	24.0	1250	1840	413,000	1355	305,000	8560	75,800

All dimensions in millimeters.
Sizes in brackets not preferred for new designs.
*Non-stock diameter.
**Torque calculated in accordance with VDI 2230, "Systematic Calculation of High Duty Bolted Joints," to induce approximately 800 MPa stress in screw threads. Torque values listed are for plain screws. (See Note, page 1.)

SOCKET HEAD CAP SCREWS ■ Metric ■ Body and Grip Lengths



BODY and GRIP LENGTHS

L_G is the maximum grip length and is the distance from the bearing surface to the first complete thread.

L_B is the minimum body length and is the length of the unthreaded cylindrical portion of the shank.

II

BODY AND GRIP LENGTH DIMENSIONS FOR METRIC SOCKET HEAD CAP SCREWS

Nominal Size	M1.6		M2		M2.5		M3		M4		M5		M6		M8		M10		M12		M14		M16		M20		M24	
Nominal Length	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B	L_G	L_B
20	4.8	3.0	4.0	2.0																								
25	9.8	8.0	9.0	7.0	8.0	5.7	7.0	4.5																				
30	14.8	13.0	14.0	12.0	13.0	10.7	12.0	9.5	10.0	6.5																		
35	...	...	19.0	17.0	18.0	15.7	17.0	14.5	15.0	11.5	13.0	9.0	11.0	6.0														
40	...	...	24.0	22.0	23.0	20.7	22.0	19.5	20.0	16.5	18.0	14.0	16.0	11.0														
45	...	...	...	...	28.0	25.7	27.0	24.5	25.0	21.5	23.0	19.0	21.0	16.0	17.0	10.7												
50	...	...	...	...	33.0	30.7	32.0	29.5	30.0	26.5	28.0	24.0	26.0	21.0	22.0	15.7	18.0	10.5										
55	...	...	...	...	...	...	37.0	34.5	35.0	31.5	33.0	29.0	31.0	26.0	27.0	20.7	23.0	15.5										
60	...	...	...	...	...	...	42.0	39.5	40.0	36.5	38.0	34.0	36.0	31.0	32.0	25.7	28.0	20.5	24.0	15.2								
65	...	...	...	...	...	...	47.0	44.5	45.0	41.5	43.0	39.0	41.0	36.0	37.0	30.7	33.0	25.5	29.0	20.2	25.0	15.0						
70	...	...	...	...	...	...	...	...	50.0	46.5	48.0	44.0	46.0	41.0	42.0	35.7	38.0	30.5	34.0	25.2	30.0	20.0	26.0	16.0				
80	...	...	...	...	...	...	...	...	60.0	56.5	58.0	54.0	56.0	51.0	52.0	45.7	48.0	40.5	44.0	35.2	40.0	30.0	36.0	26.0				
90	...	...	...	...	...	...	...	...	...	...	68.0	64.0	66.0	61.0	62.0	55.7	58.0	50.5	54.0	45.2	50.0	40.0	46.0	36.0	38.0	25.5		
100	...	...	...	...	...	...	...	...	...	...	78.0	74.0	76.0	71.0	72.0	65.7	68.0	60.5	64.0	55.2	60.0	50.0	56.0	46.0	48.0	35.5	40.0	25.0
110	...	...	...	...	...	...	...	...	...	...	...	...	86.0	81.0	82.0	75.7	78.0	70.5	74.0	65.2	70.0	60.0	66.0	56.0	58.0	45.5	50.0	35.0
120	...	...	...	...	...	...	...	...	...	...	...	...	96.0	91.0	92.0	85.7	88.0	80.5	84.0	75.2	80.0	70.0	76.0	66.0	68.0	55.5	60.0	45.0
130	...	...	...	...	...	...	...	...	...	...	...	...	...	...	102.0	95.7	98.0	90.5	94.0	85.2	90.0	80.0	86.0	76.0	78.0	65.5	70.0	55.0
140	...	...	...	...	...	...	...	...	...	...	...	...	...	...	112.0	105.7	108.0	100.5	104.0	95.2	100.0	90.0	96.0	86.0	88.0	75.5	80.0	65.0
150	...	...	...	...	...	...	...	...	...	...	...	...	...	...	122.0	115.7	118.0	110.5	114.0	105.2	110.0	100.0	106.0	96.0	98.0	85.5	90.0	75.0
160	...	...	...	...	...	...	...	...	...	...	...	...	...	...	132.0	125.7	128.0	120.5	124.0	115.2	120.0	110.0	116.0	106.0	108.0	95.5	100.0	85.0
180	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	148.0	140.5	144.0	135.2	140.0	130.0	136.0	126.0	128.0	115.5	120.0	105.0
200	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	168.0	160.5	164.0	155.2	160.0	150.0	156.0	146.0	148.0	135.5	140.0	125.0
220	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	184.0	175.2	180.0	170.0	176.0	166.0	168.0	155.5	160.0	145.0
240	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	204.0	195.2	200.0	190.0	196.0	186.0	188.0	175.5	180.0	165.0
260	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	220.0	210.0	216.0	206.0	208.0	195.5	200.0	185.0
300	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	256.0	246.0	248.0	235.5	240.0	225.0

SOCKET HEAD CAP SCREWS (METRIC SERIES)
PER ASME/ANSI B18.3.1M-1986

METRIC SOCKET FLAT HEAD CAP SCREWS

Dimensions

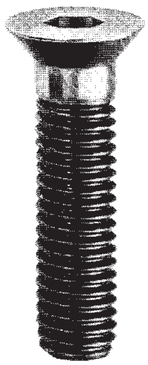
Threads: ANSI B1.13M, ISO 262 (coarse series only)

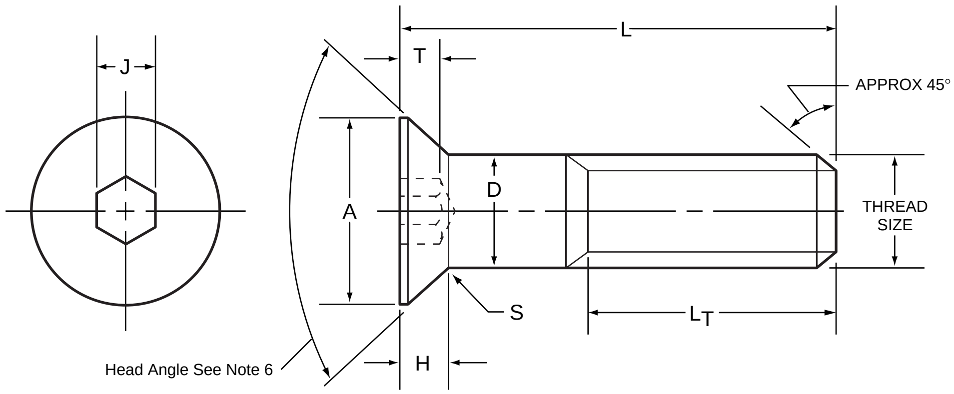
Applicable or Similar Specification: DIN 9427

General Note: Flat, countersunk head cap screws and button head cap screws are designed and recommended for moderate fastening applications: machine guards, hinges, covers, etc. They are not suggested for use in critical high strength applications where socket head cap screws should be used.

NOTES

1. **Material:** ASTM F835M
2. **Dimensions:** B18.3.5M
3. **Property Class:** 12.9
4. **Hardness:** Rc 38-43 (alloy steel)
5. **Tensile Stress:** 1040MPa
6. **Shear Stress:** 630 MPa
7. **Yield Stress:** 945 MPa
8. **Sizes:** For sizes up to and including M20, head angle shall be 92°/90°. For larger sizes head angle shall be 62°/60°.
9. **Thread Class:** 4g 6g





LENGTH TOLERANCE

nominal screw length	nominal screw diameter M3 thru M24 tolerance on lgth., mm
Up to 16 mm, incl.	±0.3
Over 16 to 60 mm, incl.	±0.5
Over 60 mm	±0.8

DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	A max.	D max.	H ref.	T min.	S ref.	L_T min.	J nom.	recommended seating torque** plain N-m	recommended seating torque** plain in-lbs.
M3	0.5	6.72	3	1.7	1.10	0.50	18	2	1.2	11
M4	0.7	8.96	4	2.3	1.55	0.70	20	2.5	2.8	25
M5	0.8	11.20	5	2.8	2.05	0.70	22	3	5.5	50
M6	1.0	13.44	6	3.3	2.25	0.85	24	4	9.5	85
M8	1.25	17.92	8	4.4	3.20	1.20	28	5	24	210
M10	1.50	22.40	10	5.5	3.80	1.50	32	6	47	415
M12	1.75	26.88	12	6.5	4.35	1.85	36	8	82	725
M16	2.00	33.60	16	7.5	4.89	1.85	44	10	205	1800
M20	2.50	40.32	20	8.5	5.45	1.85	52	12	400	3550
*M24	3.00	40.42	24	14.0	10.15	2.20	60	14	640	5650

All dimensions in millimeters.
*Non-stock Diameter
**Torque calculated to induce 420 MPa in the screw threads.
Torque values are for plain screws. (See Note, page 1.)

METRIC SOCKET BUTTON HEAD CAP SCREWS

Dimensions

II

Threads: ANSI B1.13M, ISO 262(coarse series only)

Similar Specifications: DIN 9427, ISO 7380

General Note: Flat, countersunk head cap screws and button head cap screws are designed and recommended for moderate fastening applications: machine guards, hinges, covers, etc. They are not suggested for use in critical high strength applications where socket head cap screws should be used.

NOTES

1. Material: ASTM F835M

2. Dimensions: ANSI B18.3.4M

3. Property Class: 12.9

4. Hardness: Rc 38-43

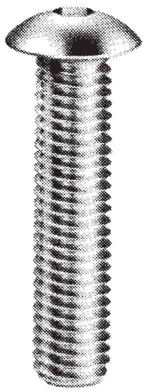
5. Tensile Stress: 1040 MPa

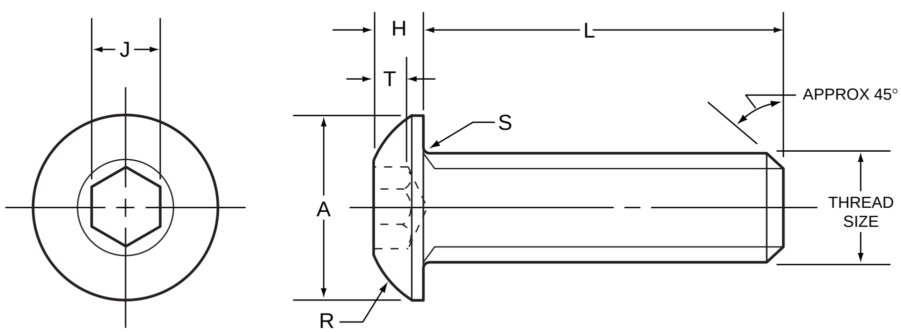
6. Shear Stress: 630 MPa

7. Yield Stress: 945 MPa

8. Bearing surface of head square with body within 2°.

9. Thread Class: 4g 6g





LENGTH TOLERANCE

nominal screw length	nominal screw diameter M3 thru M16 tolerance on lgth., mm
Up to 16 mm, incl.	±0.3
Over 16 to 60 mm, incl.	±0.5
Over 60 mm	±0.8

DIMENSIONS

APPLICATION DATA

nom. thread size	pitch	A	H	T	R	S	J	recommended seating torque** plain	
		max.	max.	min.	ref.	ref.	nom.	N-m	in-lbs.
M3	0.5	5.70	1.65	1.05	2.95	.35	2.0	1.2	11
M4	0.7	7.60	2.20	1.35	4.10	.35	2.5	2.8	25
M5	0.8	9.50	2.75	1.92	5.20	.45	3.0	5.5	50
M6	1.0	10.50	3.30	2.08	5.60	.45	4.0	9.5	85
M8	1.285	14.00	4.40	2.75	7.50	.45	5.0	24.0	210
M10	1.50	18.00	5.50	3.35	10.00	.60	6.0	47.0	415
M12	1.75	21.00	6.60	4.16	11.00	.60	8.0	82.0	725
*M16	2.0	28.00	8.60	5.20	15.00	.60	10.0	205.0	1800

All dimensions in millimeters.
*Non-stock Diameter
**Torque calculated to induce 420 MPa in the screw threads.
Torque values are for plain screws. (See Note, page 1.)

METRIC SOCKET HEAD SHOULDER SCREWS

Threads: ANSI B 1.13 M, ISO 262

Similar Specifications: ANSI B18.3.3M, ISO 7379, DIN 9841

NOTES

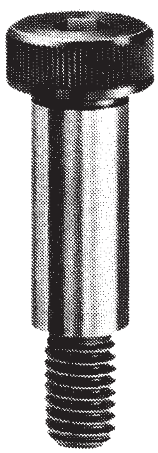
1. **Material:** ASTM A574M alloy steel
2. **Hardness:** Rc 36-43
3. **Tensile Stress:** 1100 MPa based on minimum thread neck area (J min.).
4. **Shear Stress:** 660 MPa
5. **Concentricity:** Body to head O.D. within 0.15 TIR when checked in a "V" block.

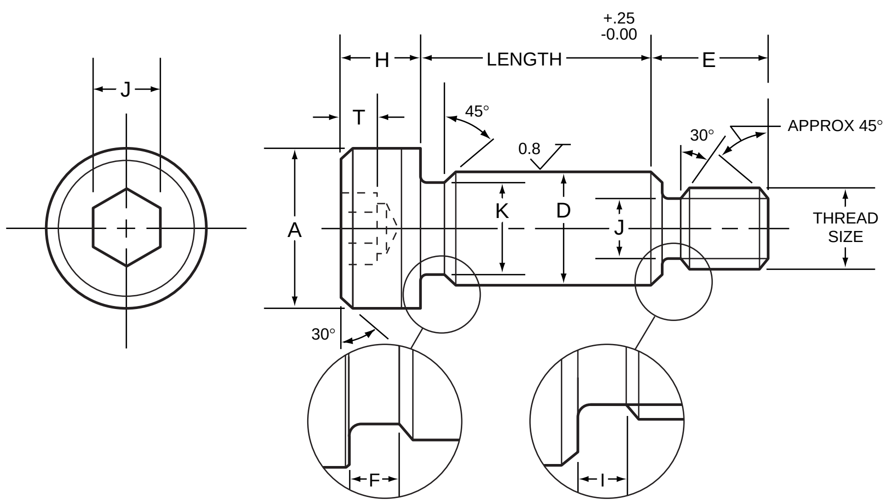
Body to thread pitch diameter within 0.1 TIR when checked at a distance of 5.0 C from the shoulder at the threaded end.

Squareness, concentricity, parallelism, and bow of body to thread pitch diameter shall be within 0.05 TIR per centimeter of body length with a maximum of 0.5 when seated against the shoulder in a threaded bushing and checked on the body at a distance of 2.5 "B" from the underside of the head.

6. **Squareness:** The bearing surface of the head shall be perpendicular to the axis of the body within a maximum deviation of 2°.
7. **Thread Class:** 4g 6g

II





DIMENSIONS **APPLICATION DATA**

nom. size	thread size	pitch	A max.	T min.	D* max.	D* min.	K min.	H max.	G min.	F max.	I max.	E max.	J nom.	recommended seating torque** N-m	recommended seating torque** in-lbs.
6	M5	0.8	10.00	2.4	6.0	5.982	5.42	4.50	3.68	2.5	2.40	9.75	3	7	60
8	M6	1.0	13.00	3.3	8.0	7.978	7.42	5.50	4.40	2.5	2.60	11.25	4	12	105
10	M8	1.25	16.00	4.2	10.0	9.978	9.42	7.00	6.03	2.5	2.80	13.25	5	29	255
12	M10	1.5	18.00	4.9	12.0	11.973	11.42	8.00	7.69	2.5	3.00	16.40	6	57	500
16	M12	1.75	24.00	6.6	16.0	15.973	15.42	10.00	9.35	2.5	4.00	18.40	8	100	885
20	M16	2.0	30.00	8.8	20.0	19.967	19.42	14.00	12.96	2.5	4.80	22.40	10	240	2125
24	M20	2.5	36.00	10.0	24.0	23.967	23.42	16.00	16.30	3.0	5.60	27.40	12	470	4160

All dimensions in millimeters.
*Shoulder diameter tolerance h8 (ISO R 286)

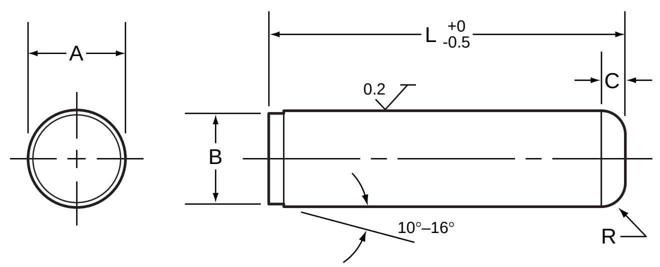
**See Note, page 1.

Applicable or Similar Specifications: ANSI B18.8.5M, ISO 8734 or DIN 6325.

Installation warning: Dowel pins should not be installed by striking or hammering. Wear safety glasses or shield when pressing chamfered point end first.

NOTES

1. **Material:** ANSI B18.85-alloy steel
2. **Hardness:** Rockwell C60 minimum (surface)
   Rockwell C 50-58 (core)
3. **Shear Stress:** Calculated values based on 1050 MPa.
4. **Surface Finish:** 0.2 micrometer maximum



DIMENSIONS **APPLICATION DATA**

nominal size	A pin diameter		B point diameter		C crown height	R crown radius	calculated single shear strength		recommended hole size	
	max.	min.	max.	min.	max.	min.	kN	pounds	max.	min.
3	3.008	3.003	2.9	2.6	0.8	0.3	7.4	1,670	3.000	2.987
4	4.009	4.004	3.9	3.6	0.9	0.4	13.2	2,965	4.000	3.987
5	5.009	5.004	4.9	4.6	1.0	0.4	20.6	4,635	5.000	4.987
6	6.010	6.004	5.8	5.4	1.1	0.4	29.7	6,650	6.000	5.987
8	8.012	8.006	7.8	7.4	1.3	0.5	52.5	11,850	8.000	7.987
10	10.012	10.006	9.8	9.4	1.4	0.6	82.5	18,550	10.000	9.987
12	12.013	12.007	11.8	11.4	1.6	0.6	119.0	26,700	12.000	11.985
16	16.013	16.007	15.8	15.3	1.8	0.8	211.0	47,450	16.000	15.985
20	20.014	20.008	19.8	19.3	2.0	0.8	330.0	74,000	20.000	19.983
25	25.014	25.008	24.8	24.3	2.3	1.0	515.0	116,000	25.000	24.983

All dimensions in millimeters.

METRIC SOCKET SET SCREWS ■ Knurled Cup Point and Plain Cup Point ■ Dimensions

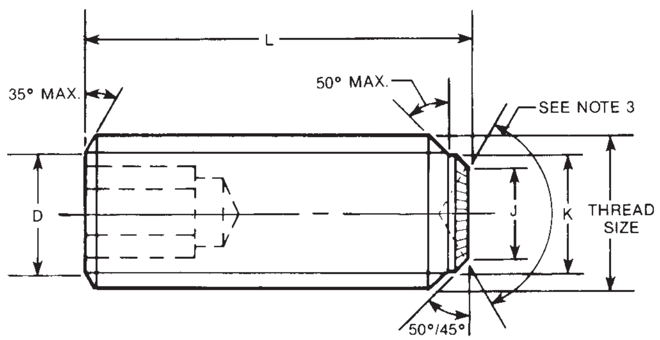
Threads: ANSI B 1.13M, ISO 261, ISO 262 (coarse series only)

Grade: 45H

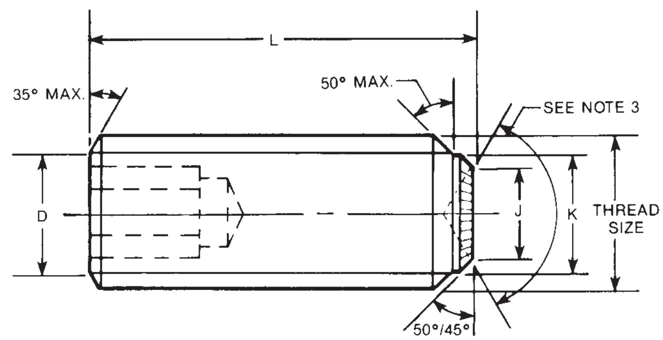
Applicable or Similar Specifications: ANSI B 18.3.6M, ISO 4029, DIN 916, DIN 915, DIN 914, DIN 913

NOTES

1. **Material:** ASTM F912M
2. **Hardness:** Rockwell C45-53
3. **Angle:** The cup angle is 135 maximum for screw lengths equal to or smaller than screw diameter. For longer lengths, the cup angle will be 124 maximum
4. **Thread Class:** 4g 6g



KNURLED CUP POINT



PLAIN CUP POINT

LENGTH TOLERANCE

nominal screw length	nominal screw diameter M1.6 thru M24 tolerance on lgth., mm
Up to 12 mm, incl.	±0.3
Over 12 to 50 mm, incl.	±0.5
Over 50 mm	±0.8

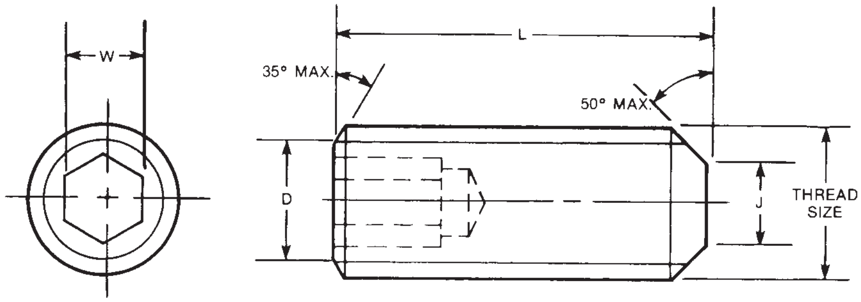
DIMENSIONS

APPLICATION DATA

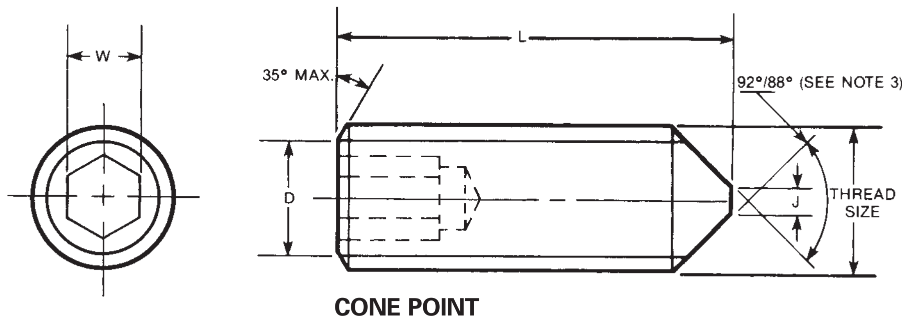
nom. thread size	pitch	D max.	J max. plain cup	J max. knurled cup	K max.	L min. preferred plain cup	L min. preferred knurled cup	W nom.	recommended* seating torque N-m	recommended* seating torque in-lbs.
MICROSIZE – Plain Cup Only										
M1.6	0.35	1.0	0.80	–	–	2.0	–	0.7	0.09	0.8
M2	0.40	1.32	1.00	–	–	2.5	–	0.9	0.21	1.8
M2.5	0.45	1.75	1.25	–	–	3.0	–	1.3	0.57	5.0
STANDARD SIZE – Knurled Cup Point Supplied Unless Plain Cup Point Is Specified										
M3	0.5	2.10	1.50	1.40	2.06	3.0	3.0	1.5	0.92	8.0
M4	0.7	2.75	2.00	2.10	2.74	3.0	3.0	2.0	2.2	19.0
M5	0.8	3.70	2.50	2.50	3.48	4.0	4.0	2.5	4.0	35.0
M6	1.0	4.35	3.00	3.30	4.14	4.0	5.0	3.0	7.2	64
M8	1.25	6.00	5.00	5.00	5.62	5.0	6.0	4.0	17.0	150.0
M10	1.5	7.40	6.00	6.00	7.12	6.0	8.0	5.0	33.0	290
M12	1.75	8.60	8.00	8.00	8.58	8.0	10.0	6.0	54.0	480
M16	2.0	12.35	10.00	10.00	11.86	12.0	14.0	8.0	134	1190
M20	2.5	16.00	14.00	14.00	14.83	16.0	18.0	10.0	237	2100
M24	3.0	18.95	16.00	16.00	17.80	20.0	20.0	12.0	440	3860

All dimensions in millimeters.
*Not applicable to screws with a length equal to or less than the diameter. See Note, page 1.

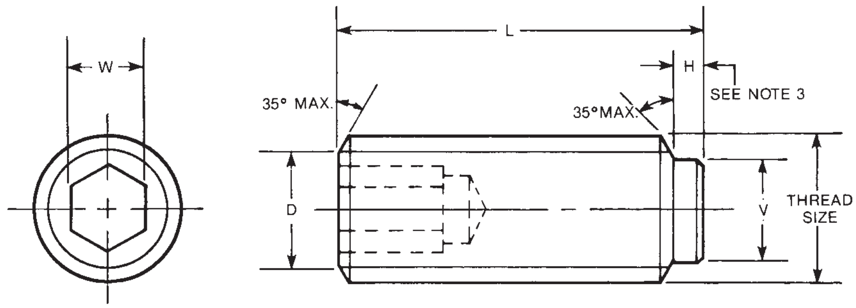
II



FLAT POINT



CONE POINT



DOG POINT

DIMENSIONS

nom. thread size	pitch	D	flat point		cone point		dog point			
			J	L	J	L	H nom.		L	V
		max.	max.	min. preferred	max.	min. preferred	short lgth.	long lgth.	min. preferred	max.
M3	0.5	2.10	2.0	3.0	0.3	4.0	0.75	1.5	5.0	2.00
M4	0.7	2.75	2.5	3.0	0.4	4.0	1.00	2.0	5.0	2.50
M5	0.8	3.70	3.5	4.0	0.5	5.0	1.25	2.5	6.0	3.50
M6	1.00	4.25	4.0	4.0	1.5	6.0	1.50	3.0	6.0	4.00
M8	1.25	6.00	5.5	5.0	2.0	6.0	2.00	4.0	8.0	5.50
M10	1.50	7.40	7.0	6.0	2.5	8.0	2.50	5.0	8.0	7.00
M12	1.75	8.60	8.5	8.0	3.0	10.0	3.00	6.0	12.0	8.50
M16	2.00	12.35	12.0	12.0	4.0	14.0	4.00	8.0	16.0	12.00
M20	2.50	16.00	15.0	14.0	6.0	18.0	5.00	10.0	20.0	15.00
M24	3.00	18.95	18.0	20.0	8.0	20.0	6.00	12.0	22.0	18.00

METRIC LOW HEAD CAP SCREWS

Threads: ANSI B 1.13M, ISO 262 (coarse series only)

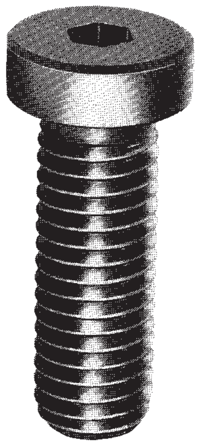
Property Class: 10.9

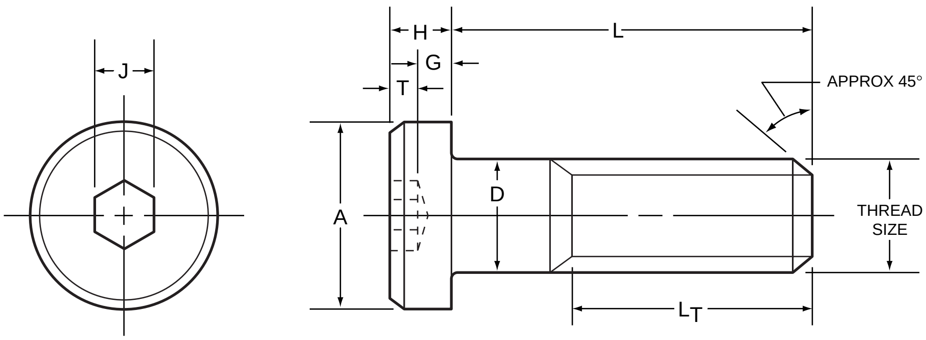
Similar Specifications: DIN 7984, DIN 6912

NOTES

1. **Material:** ASTM A574M-alloy steel
2. **Hardness:** Rc 33-39
3. **Tensile Stress:** 1040 MPa
4. **Yield Stress:** 940 MPa
5. **Thread Class:** 4g 6g

II





DIMENSIONS **APPLICATION DATA**

nom. thread size	pitch	A max.	D max.	G min.	T min.	H max.	L_T min.	J nom.	recommended* seating torque, plain: N-m	recommended* seating torque, plain: in-lbs.
M4	0.7	7	4	1.06	1.48	2.8	20	3	4.5	40
M5	0.8	8.5	5	1.39	1.85	3.5	22	4	8.5	75
M6	1.0	10	6	1.65	2.09	4.0	24	5	14.5	130
M8	1.25	13	8	2.24	2.48	5.0	28	6	35	310
M10	1.5	16	10	2.86	3.36	6.5	32	8	70	620
M12	1.75	18	12	3.46	4.26	8.0	36	10	120	1060
M16	2.0	24	16	4.91	4.76	10.0	44	12	300	2650
M20	2.5	30	20	6.10	6.07	12.5	52	14	575	5100

All dimensions in millimeters.
*Torque calculated to induce 620 MPa in the screw threads.
Torque values are for plain screws. (See Note, page 1.)

METRIC HEXAGON KEYS

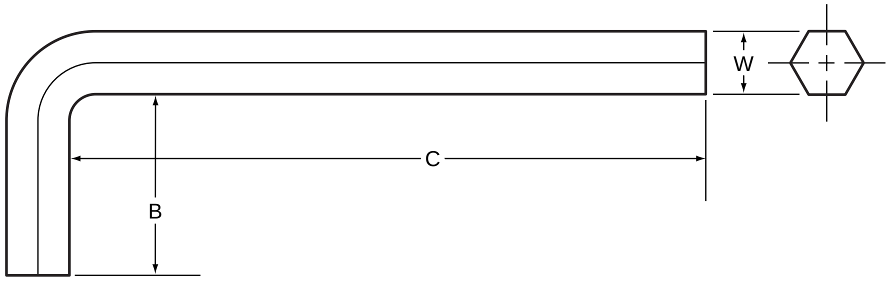
Dimensions ■ Mechanical Properties ■ Socket Applications

These UNBRAKO keys are made to higher requirements than ISO or DIN keys, which may not properly torque Class 12.9 cap screws. The strength and dimensional requirements are necessary to properly install the products in this catalog.

Material: ANSI B18.3.2.M alloy steel

Dimensions: ANSI B18.3.2.M

Similar Specifications: DIN 911, ISO 2936



II

METRIC KEY APPLICATION CHART

size W	socket cap screws std. head height	socket cap screws low head	socket cap screws	flat head socket cap screws	button head shoulder screws	socket set screws
0.7						M1.6
0.9						M2
1.3						M2.5
1.5	M1.6/M2					M3
2.0	M2.5		M3	M3		M4
2.5	M3		M4	M4		M5
3.0	M4	M4	M5	M5	M6	M6
4.0	M5	M5	M6	M6	M8	M8
5.0	M6	M6	M8	M8	M10	M10
6.0	M8	M8	M10	M10	M12	M12
8.0	M10	M10	M12	M12	M16	M16
10.0	M12	M12	M16	M16	M20	M20
12.0	M14	M16	M20		M24	M24
14.0	M16	M20	M24			
17.0	M20	M24				
19.0	M24					
22.0	M30					
27.0	M36					
32.0	M42					
36.0	M48					

DIMENSIONS / MECHANICAL PROPERTIES

key size W max.	key size W min.	B nominal	C nominal short arm	C nominal long arm	torsional shear strength minimum N-m	torsional shear strength minimum In-lbs.	torsional yield strength minimum N-m	torsional yield strength minimum In-lbs.
0.711	0.698	5.5	31	*69	0.12	1.1	0.1	0.9
0.889	0.876	9	31	71	0.26	2.3	0.23	2.
1.270	1.244	13.5	42	75	0.73	6.5	.63	5.6
1.500	1.470	14	45	78	1.19	10.5	1.02	9.
2.000	1.970	16	50	83	2.9	26	2.4	21
2.500	2.470	18	56	90	5.4	48	4.4	39
3.000	2.960	20	63	100	9.3	82	8.	71
4.000	3.960	25	70	106	22.2	196	18.8	166
5.000	4.960	28	80	118	42.7	378	36.8	326
6.000	5.960	32	90	140	74	655	64	566
8.000	7.950	36	100	160	183	1,620	158	1,400
10.000	9.950	40	112	170	345	3,050	296	2,620
12.000	11.950	45	125	212	634	5,610	546	4,830
14.000	13.930	55	140	236	945	8,360	813	7,200
17.000	16.930	60	160	250	1,690	15,000	1,450	12,800
19.000	18.930	70	180	280	2,360	20,900	2,030	18,000
22.000	21.930	80	*200	*335	3,670	32,500	3,160	28,000
24.000	23.930	90	*224	*375	4,140	36,600	3,560	31,500
27.000	26.820	100	*250	*500	5,870	51,900	5,050	44,700
32.000	31.820	125	*315	*630	8,320	73,600	7,150	63,300
36.000	35.820	140	*355	*710	11,800	104,000	10,200	90,300

All dimensions in millimeters.
*Non-stock sizes

ISO TOLERANCES FOR METRIC FASTENERS

nominal dimension		tolerance zone in mm (external measurements)													tolerance zone in mm					
over	to	h6	h8	h10	h11	h13	h14	h15	h16	js14	js15	js16	js17	m6	H7	H8	H9	H11	H13	H14
0	1	0 –0.006	0 –0.014	0 –0.040	0 –0.060	0 –0.14								+0.002 +0.008	+0.010 0	+0.0014 0	+0.025 0	+0.060 0	+0.14 0	
1	3	0 –0.006	0 –0.014	0 –0.040	0 –0.060	0 –0.14	0 –0.25	0 –0.40	0 –0.60	±0.125	±0.20	±0.30	±0.50	+0.002 +0.008	+0.010 0	+0.014 0	+0.025 0	+0.060 0	+0.14 0	+0.25 0
3	6	0 –0.008	0 –0.018	0 –0.048	0 –0.075	0 –0.18	0 –0.30	0 –0.48	0 –0.75	±0.15	±0.24	±0.375	±0.60	+0.004 +0.012	+0.012 0	+0.018 0	+0.030 0	+0.075 0	+0.18 0	+0.30 0
6	10	0 –0.009	0 –0.022	0 –0.058	0 –0.090	0 –0.22	0 –0.36	0 –0.58	0 –0.90	±0.18	±0.29	±0.45	±.075	+0.006 +0.0015	+0.015 0	+0.022 0	+0.036 0	+0.090 0	+0.22 0	+0.36 0
10	18	0 –0.011	0 –0.027	0 –0.070	0 –0.110	0 –0.27	0 –0.43	0 –0.70	0 –1.10	±0.215	±0.35	±0.55	±0.90	+0.007 +0.018	+0.018 0	+0.027 0	+0.043 0	+0.110 0	+0.27 0	+0.43 0
18	30	0 –0.030	0 –0.033	0 –0.084	0 –0.130	0 –0.33	0 –0.52	0 –0.84	0 –1.30	± 0.26	±0.42	±0.65	±1.05	+0.008 +0.021	+0.021 0	+0.033 0	+0.052 0	+0.130 0	+0.33 0	+0.52 0
30	50					0 –0.39	0 –0.62	0 –1.00	0 –1.60	±0.31	±0.50	±0.80	±1.25						+0.39 0	+0.62 0
50	80					0 –0.46	0 –0.74	0 –1.20	0 –1.90	±0.37	±0.60	±0.95	±1.50						+0.46 0	+0.74 0
80	120					0 –0.54	0 –0.87	0 –1.40	0 –2.20	±0.435	±0.70	±1.10	±1.75						+0.54 0	+0.87 0
120	180									±0.50	±0.80	±1.25	±2.00							
180	250									±0.575	±0.925	±1.45	±2.30							
250	315									±0.65	±1.05	±1.60	±2.60							
315	400									±0.70	±1.15	±1.80	±2.85							
400	500									±0.775	±1.25	±2.00	±3.15							

ISO TOLERANCES FOR SOCKET SCREWS

nominal dimension		tolerance zone in mm										
over	to	C13	C14	D9	D10	D11	D12	EF8	E11	E12	Js9	K9
	3	+0.20 +0.06	+0.31 +0.06	+0.045 +0.020	+0.060 +0.020	+0.080 +0.020	+0.12 +0.02	+0.024 +0.010	+0.074 +0.014	+0.100 +0.014	±0.0125	0 -0.025
3	6	+0.24 +0.06	+0.37 +0.07	+0.060 +0.030	+0.078 +0.030	+0.115 +0.030	+0.15 +0.03	+0.028 +0.014	+0.095 +0.020	+0.140 +0.020	±0.015	0 -0.030
6	10					+0.130 +0.040	+0.19 +0.40	+0.040 +0.018	+0.115 +0.025	+0.115 +0.025	±0.018	0 -0.036
10	18						+0.2 +0.05		+0.142 +0.032	+0.212 +0.032		
18	30						+0.275 +0.065					
30	50						+0.33 +0.08					
50	80						+0.40 +0.10					

References
ISO R 286
ISO 4759/I
ISO 4759/II
ISO 4759/III

Notes

ANSI standards allow slightly wider tolerances for screw lengths than ISO and DIN.

The table is intended to assist in the design with metric fasteners. For tolerances not listed here refer to the complete standards.

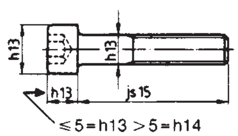
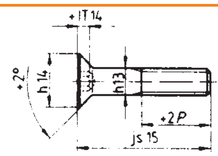
II

II

Tolerances for Metric Fasteners

The tolerances in the tables below are derived from ISO standard: ISO 4759

The tables show tolerances on the most common metric fasteners. However, occasionally some slight modifications are made.

Item	DIN
	912
	7991

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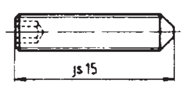
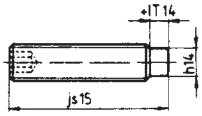
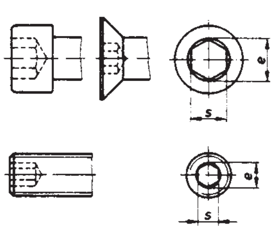
	913 914 916
	915 966

Notes

Product grade A applies to sizes up to M24 and length not exceeding 10 x diameter or 150 mm, whatever is shorter.

Product grade B applies to the sizes above M24 and all sizes with lengths, greater than 10 x diameter or 150 mm, whichever is shorter.

<table>
<tr><th>Feature</th><th colspan="3"></th></tr>
<tr><td rowspan="16">Hexagon Sockets
</td><td rowspan="2">s</td><td colspan="2">Tolerance</td></tr>
<tr><td>*</td><td>**</td></tr>
<tr><td>0.7</td><td colspan="2">EF8</td></tr>
<tr><td>0.9</td><td colspan="2">JS9</td></tr>
<tr><td>1.3</td><td colspan="2">K9</td></tr>
<tr><td>1.5</td><td rowspan="2">D9</td><td rowspan="2">D10</td></tr>
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</table>

*Tolerance zones for socket set screws

**Tolerance zones for socket head cap screws

Note: For S 0.7 to 1.3 the actual allowance in the product standards has been slightly modified for technical reasons.

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IMPORTANT

The technical discussions represent typical applications only. The use of the information is at the sole discretion of the reader. Because applications vary enormously, UNBRAKO does not warrant the scenarios described are appropriate for any specific application. The reader must consider all variables prior to using this information.

II

INSTALLATION CONTROL

Several factors should be considered in designing a joint or selecting a fastener for a particular application.

JOINT DESIGN AND FASTENER SELECTION.

Joint Length

The longer the joint length, the greater the total elongation will occur in the bolt to produce the desired clamp load or preload. In design, if the joint length is increased, the potential loss of preload is decreased.

Joint Material

If the joint material is relatively stiff compared to the bolt material, it will compress less and therefore provide a less sensitive joint, less sensitive to loss of preload as a result of brinelling, relaxation and even loosening.

Thread Stripping Strength

Considering the material in which the threads will be tapped or the nut used, there must be sufficient engagement length to carry the load. Ideally, the length of thread engagement should be sufficient to break the fastener in tension. When a nut is used, the wall thickness of the nut as well as its length must be considered.

An estimate, a calculation or joint evaluation will be required to determine the tension loads to which the bolt and joint will be exposed. The size bolt and the number necessary to carry the load expected, along with the safety factor, must also be selected.

The safety factor selected will have to take into consideration the consequence of failure as well as the additional holes and fasteners. Safety factors, therefore, have to be determined by the designer.

SHEAR APPLICATIONS

Shear Strength of Material

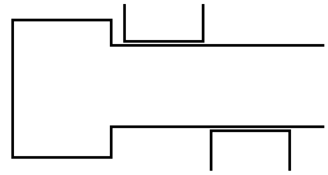
Not all applications apply a tensile load to the fastener. In many cases, the load is perpendicular to the fastener in shear. Shear loading may be single, double or multiple loading.

There is a relationship between the tensile strength of a material and its shear strength. For alloy steel, the shear strength is 60% of its tensile strength. Corrosion resistant steels (e.g. 300-Series stainless steels) have a lower tensile/shear relationship and it is usually 50-55%

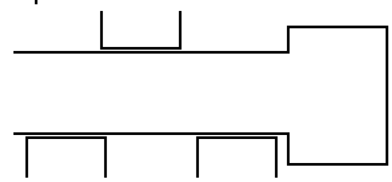
Single/Double Shear

Single shear strength is exactly one-half the double shear value. Shear strength listed in pounds per square inch (psi) is the shear load in pounds divided by the cross sectional area in square inches.

Single Shear



Double Shear



OTHER DESIGN CONSIDERATIONS

Application Temperature

For elevated temperature, standard alloy steels are useful to about 550°F–600°F. However, if plating is used, the maximum temperature may be less (eg. cadmium should not be used over 450°F.

Austenitic stainless steels (300 Series) may be useful to 800°F. They can maintain strength above 800°F but will begin to oxidize on the surface.

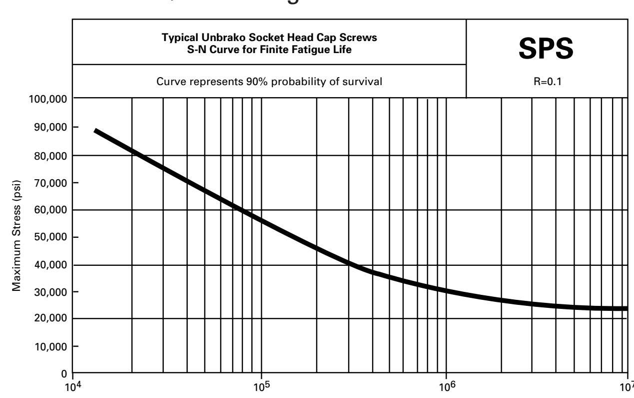
Corrosion Environment

A plating may be selected for mild atmospheres or salts. If plating is unsatisfactory, a corrosion resistant fastener may be specified. The proper selection will be based upon the severity of the corrosive environment.

FATIGUE STRENGTH

S/N Curve

Most comparative fatigue testing and specification fatigue test requirements are plotted on an S/N curve. In this curve, the test stress is shown on the ordinate (y-axis) and the number of cycles is shown on the abscissa (x-axis) in a lograthmic scale. On this type curve, the high load to low load ratio must be shown. This is usually R =.1, which means the low load in all tests will be 10% of the high load.



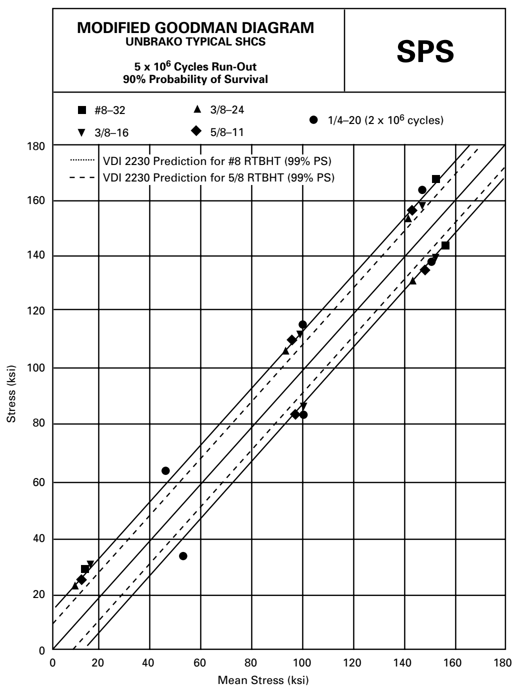
Effect of Preload

Increasing the R to .2, .3 or higher will change the curve shape. At some point in this curve, the number of cycles will reach 10 million cycles. This is considered the endurance limit or the stress at which infinite life might be expected.

Modified Goodman/ Haigh Soderberg Curve

The S/N curve and the information it supplies will not provide the information needed to determine how an individual fastener will perform in an actual application. In application, the preload should be higher than any of the preloads on the S/N curve.

Therefore, for application information, the modified Goodman Diagram and/or the Haigh Soderberg Curve are more useful. These curves will show what fatigue performance can be expected when the parts are properly preloaded.



METHODS OF PRELOADING

Elongation

The modulus for steel of 30,000,000 (thirty million) psi means that a fastener will elongate .001 in/in of length for every 30,000 psi in applied stress. Therefore, if 90,000 psi is the desired preload, the bolt must be stretched .003 inches for every inch of length in the joint.

This method of preloading is very accurate but it requires that the ends of the bolts be properly prepared and also that all measurements be very carefully made. In addition, direct measurements are only possible where both ends of the fastener are available for measurement after installation. Other methods of measuring lengths changes are ultrasonic, strain gages and turn of the nut.

Torque

By far, the most popular method of preloading is by torque. Fastener manufacturers usually have recommended seating torques for each size and material fastener. The only requirement is the proper size torque wrench, a conscientious operator and the proper torque requirement.

Strain

Since stress/strain is a constant relationship for any given material, we can use that relationship just as the elongation change measurements were used previously.

Now, however, the strain can be detected from strain gages applied directly to the outside surface of the bolt or by having a hole drilled in the center of the bolt and the strain gage installed internally. The output from these gages need instrumentation to convert the gage electrical measurement method. It is, however, an expensive method and not always practical.

Turn of the Nut

The nut turn method also utilizes change in bolt length. In theory, one bolt revolution (360° rotation) should increase the bolt length by the thread pitch. There are at least two variables, however, which influence this relationship. First, until a snug joint is obtained, no bolt elongation can be measured. The snugging produces a large variation in preload. Second, joint compression is also taking place so the relative stiffnesses of the joint and bolt influences the load obtained.

VARIABLES IN TORQUE

Coefficient of Friction

Since the torque applied to a fastener must overcome all friction before any loading takes place, the amount of friction present is important.

In a standard unlubricated assembly, the friction to be overcome is the head bearing area and the thread-to-thread friction. Approximately 50% of the torque applied will be used to overcome this head-bearing friction and approximately 35% to overcome the thread friction. So 85% of the torque is overcoming friction and only 15% is available to produce bolt load.

If these interfaces are lubricated (cadmium plate, molybdenum disulfide, anti-seize compounds, etc.), the friction is reduced and thus greater preload is produced with the same torque.

The change in the coefficient of friction for different conditions can have a very significant effect on the slope of the torque tension curve. If this is not taken into consideration, the proper torque specified for a plain unlubricated bolt may be sufficient to yield or break a lubricated fastener.

Thread Pitch

The thread pitch must be considered when a given stress is to be applied, since the cross-sectional area used for stress calculations is the thread tensile stress area and is different for coarse and fine threads. The torque recommendations, therefore, are slightly higher for fine threads than for coarse threads to achieve the same stress.

Differences between coarse and fine threads.

Coarse Threads are…
- more readily available in industrial fasteners.
- easier to assemble because of larger helix angle.
- require fewer turns and reduce cross threading.
- higher thread stripping strength per given length.
- less critical of tap drill size.
- not as easily damaged in handling.

Their disadvantages are…
- lower tensile strength.
- reduced vibrational resistance.
- coarse adjustment.

Fine Threads provide...
- higher tensile strength.
- greater vibrational resistance.
- finer adjustment.

Their disadvantages are…
- easier cross threaded.
- threads damaged more easily by handling.
- tap drill size slightly more critical.
- slightly lower thread stripping strength.

II

Other Design Guidelines

In addition to the joint design factors discussed, the following considerations are important to the proper use of high-strength fasteners.

- Adequate thread engagement should be guaranteed by use of the proper mating nut height for the system. Minimum length of engagement recommended in a tapped hole depends on the strength of the material, but in all cases should be adequate to prevent stripping.
- Specify nut of proper strength level. The bolt and nut should be selected as a system.
- Specify compatible mating female threads. 2B tapped holes or 3B nuts are possibilities.
- Corrosion, in general, is a problem of the joint, and not just of the bolt alone. This can be a matter of galvanic action between dissimilar metals. Corrosion of the fastener material surrounding the bolt head or nut can be critical with high-strength bolting. Care must be exercised in the compatibility of joint materials and/or coatings to protect dissimilar metals.

PROCESSING CONTROL

The quality of the raw material and the processing control will largely affect the mechanical properties of the finished parts.

MATERIAL SELECTION

The selection of the type of material will depend on its end use. However, the control of the analysis and quality is a critical factor in fastener performance. The material must yield reliable parts with few hidden defects such as cracks, seams, decarburization and internal flaws.

FABRICATION METHOD

Head

There are two general methods of making bolt heads, forging and machining. The economy and grain flow resulting from forging make it the preferred method.

The temperature of forging can vary from room temperature to 2000°F. By far, the greatest number of parts are cold upset on forging machines known as headers or boltmakers. For materials that do not have enough formability for cold forging, hot forging is used. Hot forging is also used for bolts too large for cold upsetting due to machine capacity. The largest cold forging machines can make bolts up to 1-1/2 inch diameter. For large quantities of bolts, hot forging is more expensive then cold forging.

Some materials, such as stainless steel, are warm forged at temperatures up to 1000°F. The heating results in two benefits, lower forging pressures due to lower yield strength and reduced work hardening rates.

Machining is the oldest method and is used for very large diameters or small production runs.

The disadvantage is that machining cuts the metal grain flow, thus creating planes of weakness at the critical head-to-shank fillet area. This can reduce tension fatigue performance by providing fracture planes.

Fillets

The head-to-shank transition (fillet) represents a sizable change in cross section at a critical area of bolt performance. It is important that this notch effect be minimized. A generous radius in the fillet reduces the notch effect. However, a compromise is necessary because too large a radius will reduce load-bearing area under the head.

Composite radii such as elliptical fillets, maximize curvature on the shank side of the fillet and minimize it on the head side to reduce loss of bearing area on the load-bearing surface.

Critical Fastener Features

Head-Shank-Fillet: This area on the bolt must not be restricted or bound by the joint hole. A sufficient chamfer or radius on the edge of the hole will prevent interference that could seriously reduce fatigue life. Also, if the bolt should seat on an unchamfered edge, there might be serious loss of preload if the edge breaks under load.

Threads

Threads can be produced by grinding, cutting or rolling.

In a rolled thread, the material is caused to flow into the thread die contour, which is ground into the surface during the manufacture of the die. Machines with two or three circular dies or two flat dies are most common.

Thread cutting requires the least tooling costs and is by far the most popular for producing internal threads. It is the most practical method for producing thin wall parts and the only technique available for producing large diameter parts (over 3 inches in diameter).

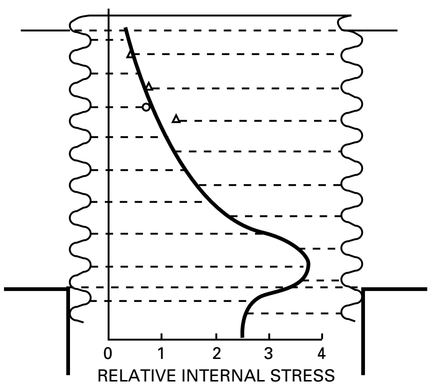
Thread grinding yields high dimensional precision and affords good control of form and finish. It is the only practical method for producing thread plug gages.

Both machining and grinding have the disadvantage of cutting material fibers at the most critical point of performance.

The shape or contour of the thread has a great effect on the resulting fatigue life. The thread root should be large and well rounded without sharp corners or stress risers. Threads with larger roots should always be used for harder materials.

In addition to the benefits of grain flow and controlled shape in thread rolling, added fatigue life can result when the rolling is performed after heat treatment.

This is the accepted practice for high fatigue performance bolts such as those used in aircraft and space applications.



EVALUATING PERFORMANCE

Mechanical Testing

In the fastener industy, a system of tests and examinations has evolved which yields reliable parts with proven performance.

Some tests are conducted on the raw material; some on the finished product.

There always seems to be some confusion regarding mechanical versus metallurgical properties. Mechanical properties are those associated with elastic or inelastic reaction when force is applied, or that involve the relationship between stress and strain. Tensile testing stresses the fastener in the axial direction. The force at which the fastener breaks is called the breaking load or ultimate tensile strength. Load is designated in pounds, stress in pounds per square inch and strain in inches per inch.

When a smooth tensile specimen is tested, the chart obtained is called a Stress-Strain Curve. From this curve, we can obtain other useful data such as yield strength. The method of determining yield is known as the offset method and consists of drawing a straight line parallel to the stress strain curve but offset from the zero point by a specified amount. This value is usually 0.2% on the strain ordinate. The yield point is the intersection of the stress-strain curve and the straight line. This method is not applicable to fasteners because of the variables introduced by their geometry.

When a fastener tensile test is plotted, a load/elongation curve can be obtained. From this curve, a yield determination known as Johnson's 2/3 approximate method for determination of yield strength is used to establish fastener yield, which will be acceptable for design purposes. It is not recommended for quality control or specification requirements.

Torque-tension testing is conducted to correlate the required torque necessary to induce a given load in a mechanically fastened joint. It can be performed by hand or machine. The load may be measured by a tensile machine, a load cell, a hydraulic tensile indicator or by a strain gage.

Fatigue tests on threaded fasteners are usually alternating tension-tension loading. Most testing is done at more severe strain than its designed service load but ususally below the material yield strength.

Shear testing, as previously mentioned, consists of loading a fastener perpendicular to its axis. All shear testing should be accomplished on the unthreaded portion of the fastener.

Checking hardness of parts is an indirect method for testing tensile strength. Over the years, a correlation of tensile strength to hardness has been obtained for most materials. See page 83 for more detailed information. Since hardness is a relatively easy and inexpensive test, it makes a good inspection check. In hardness checking, it is very important that the specimen be properly prepared and the proper test applied.

Stress durability is used to test parts which have been subjected to any processing which may have an embrittling effect. It requires loading the parts to a value higher than the expected service load and maintaining that load for a specified time after which the load is removed and the fastener examined for the presence of cracks.

Impact testing has been useful in determining the ductile brittle transformation point for many materials. However, because the impact loading direction is transverse to a fastener's normal longitude loading, its usefulness for fastener testing is minimal. It has been shown that many fastener tension impact strengths do not follow the same pattern or relationship of Charpy or Izod impact strength.

Metallurgical Testing

Metallurgical testing includes chemical composition, microstructure, grain size, carburization and decarburization, and heat treat response.

The chemical composition is established when the material is melted. Nothing subsequent to that process will influence the basic composition.

The microstructure and grain size can be influenced by heat treatment. Carburization is the addition of carbon to the surface which increases hardness. It can occur if heat treat furnace atmospheres are not adequately controlled. Decarburization is the loss of carbon from the surface, making it softer. Partial decarburization is preferable to carburization, and most industrial standards allow it within limits.

In summary, in order to prevent service failures, many things must be considered:

The Application Requirements

Strength Needed – Safety Factors
- Tension/Shear/Fatigue
- Temperature
- Corrosion
- Proper Preload

The Fastener Requirements

- Material
- Fabrication Controls
- Performance Evaluations

II

AN EXPLANATION OF JOINT DIAGRAMS

When bolted joints are subjected to external tensile loads, what forces and elastic deformation really exist? The majority of engineers in both the fastener manufacturing and user industries still are uncertain. Several papers, articles, and books, reflecting various stages of research into the problem have been published and the volume of this material is one reason for confusion. The purpose of this article is to clarify the various explanations that have been offered and to state the fundamental concepts which apply to forces and elastic deformations in concentrically loaded joints. The article concludes with general design formulae that take into account variations in tightening, preload loss during service, and the relation between preloads, external loads and bolt loads.

The Joint Diagram

Forces less than proof load cause elastic strains. Conversely, changes in elastic strains produce force variations. For bolted joints this concept is usually demonstrated by joint diagrams.

The most important deformations within a joint are elastic bolt elongation and elastic joint compression in the axial direction. If the bolted joint in Fig. 1 is subjected to the preload F_i the bolt elongates as shown by the line OB in Fig. 2A and the joint compresses as shown by the line OJ. These two lines, representing the spring characteristics of the bolt and joint, are combined into one diagram in Fig. 2B to show total elastic deformation.

If a concentric external load F_e is applied under the bolt head and nut in Fig. 1, the bolt elongates an additional amount while the compressed joint members partially relax. These changes in deformation with external loading are the key to the interaction of forces in bolted joints.

In Fig. 3A the external load F_e is added to the joint diagram Fe is located on the diagram by applying the upper end to an extension of OB and moving it in until the lower end contacts OJ. Since the total amount of elastic deformation (bolt plus joint) remains constant for a given preload, the external load changes the total bolt elongation to $\Delta l_B + \lambda$ and the total joint compression to $\Delta l_J - \lambda$.

In Fig. 3B the external load F_e is divided into an additional bolt load F_{eB} and the joint load F_{eJ}, which unloads the compressed joint members. The maximum bolt load is the sum of the load preload and the additional bolt load:

$$F_{B\,max} = F_i + F_{eB}$$

If the external load Fe is an alternating load, F_{eB} is that part of F_e working as an alternating bolt load, as shown in Fig. 3B. This joint diagram also illustrates that the joint absorbs more of the external load than the bolt subjected to an alternating external load.

The importance of adequate preload is shown in Fig. 3C. Comparing Fig. 3B and Fig. 3C, it can be seen that F_{eB} will remain relatively small as long as the preload F_i is greater than F_{eJ}. Fig. 3C represents a joint with insufficient preload. Under this condition, the amount of external load that the joint can absorb is limited, and the excess load must then be applied to the bolt. If the external load is alternating, the increased stress levels on the bolt produce a greatly shortened fatigue life.

When seating requires a certain minimum force or when transverse loads are to be transformed by friction, the minimum clamping load $F_{J\,min}$ is important.

$$F_{J\,min} = F_{B\,max} - F_e$$

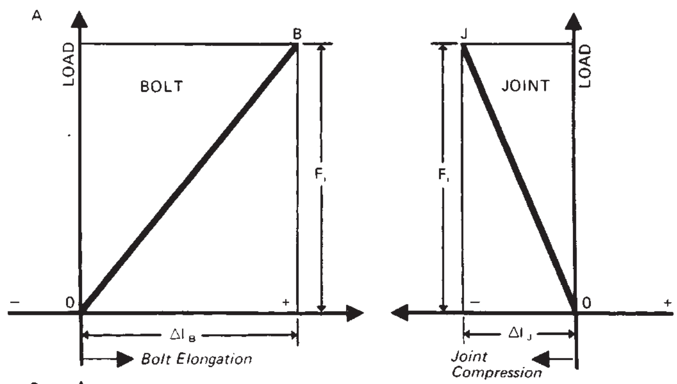


Fig. 1 (above) Joint components

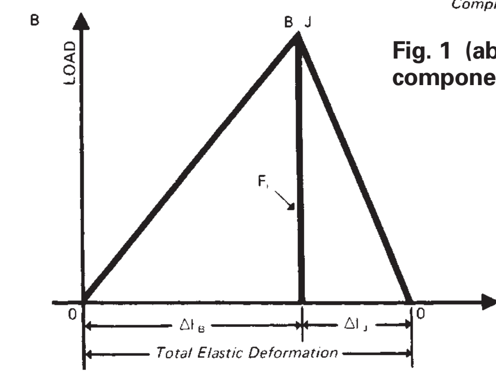


Fig. 2 Joint diagram is obtained by combining load vs. deformation diagrams of bolt and joints.

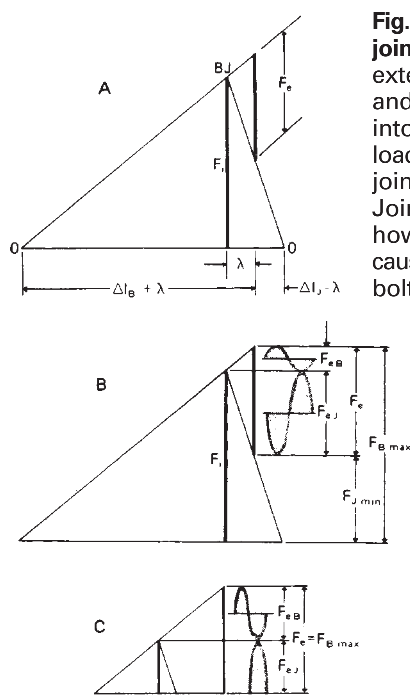


Fig. 3 The complete simple joint diagrams show external load F_e added (A), and external load divided into an additional bolt load F_{eB} and reduction in joint compression F_{eJ} (B). Joint diagram (C) shows how insufficient preload F_i causes excessive additional bolt load F_{eB}.

Spring Constants

To construct a joint diagram, it is necessary to determine the spring rates of both bolt and joint. In general, spring rate is defined as:

$$K = \frac{F}{\Delta l}$$

From Hook's law:

$$\Delta l = \frac{lF}{EA}$$

Therefore:

$$K = \frac{EA}{l}$$

To calculate the spring rate of bolts with different cross sections, the reciprocal spring rates, or compliances, of each section are added:

$$\frac{1}{K_B} = \frac{1}{K_1} + \frac{1}{K_2} + \ldots + \frac{1}{K_n}$$

Thus, for the bolt shown in Fig. 4:

$$\frac{1}{K_B} = \frac{1}{E}\left(\frac{0.4d}{A_1} + \frac{l_1}{A_1} + \frac{l_2}{A_2} + \frac{l_3}{A_m} + \frac{0.4d}{A_m}\right)$$

where

d = the minor thread diameter and

A_m = the area of the minor thread diameter

This formula considers the elastic deformation of the head and the engaged thread with a length of 0.4d each.

Calculation of the spring rate of the compressed joint members is more difficult because it is not always obvious which parts of the joint are deformed and which are not. In general, the spring rate of a clamped part is:

$$K_J = \frac{EA_S}{l_J}$$

where A_s is the area of a substitute cylinder to be determined.

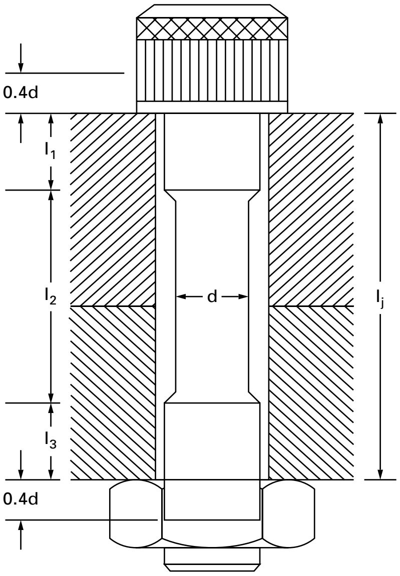


Fig. 4 Analysis of bolt lengths contributing to the bolt spring rate.

When the outside diameter of the joint is smaller than or equal to the bolt head diameter, i.e.,as in a thin bushing, the normal cross sectioned area is computed:

$$A_s = \frac{\pi}{4}\left(D_c^2 - D_h^2\right)$$

where

D_c = OD of cylinder or bushing and

D_h = hole diameter

When the outside diameter of the joint is larger than head or washer diameter D_H, the stress distribution is in the shape of a barrel, Fig 5. A series of investigations proved that the areas of the following substitute cylinders are close approximations for calculating the spring contents of concentrically loaded joints.

When the joint diameter D_J is greater than D_H but less than $3D_H$;

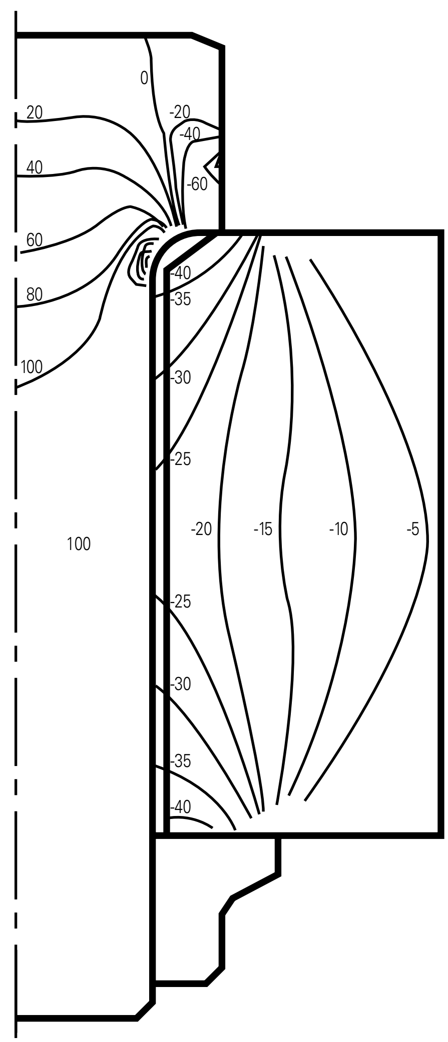


Fig. 5 Lines of equal axial stresses in a bolted joint obtained by the axisymmetric finite element method are shown for a 9/16–18 bolt preloaded to 100 KSI. Positive numbers are tensile stresses in KSI; negative numbers are compressive stresses in KSI.

$$A_s = \frac{\pi}{4}\left(D_H^2 - D_h^2\right) + \frac{\pi}{8}\left(\frac{D_J}{DH} - 1\right)\left(\frac{D_H l_J}{5} + \frac{l_J^2}{100}\right)$$

When the joint diameter D_J is equal to or greater than $3D_H$:

$$A_s = \frac{\pi}{4}\left[(D_H + 0.1\, l_J)^2 - D_h^2\right]$$

These formulate have been verified in laboratories by finite element method and by experiments.

Fig. 6 shows joint diagrams for springy bolt and stiff joint and for a stiff bolt and springy joint. These diagrams demonstrate the desirability of designing with springy bolt and a stiff joint to obtain a low additional bolt load F_{eB} and thus a low alternating stress.

The Force Ratio

Due to the geometry of the joint diagram, Fig. 7,

$$F_{eB} = \frac{K_e\, K_B}{K_B + K_J}$$

Defining $\Phi = \dfrac{K_B}{K_B + K_J}$

$F_{eB} = F_e\Phi$ and

Φ, called the Force Ratio, $= \dfrac{F_{eB}}{F_e}$

For complete derivation of Φ, see Fig. 7.

To assure adequate fatigue strength of the selected fastener the fatigue stress amplitude of the bolt resulting from an external load F_e is computed as follows:

$$\sigma_B = \pm\frac{F_{eB}/2}{A_m} \quad \text{or}$$

$$\sigma_B = \pm\frac{\Phi\, F_e}{2\, A_m}$$

Effect of Loading Planes

The joint diagram in Fig 3, 6 and 7 is applicable only when the external load F_e is applied at the same loading planes as the preloaded F_i, under the bolt head and the nut. However, this is a rare case, because the external load usually affects the joint somewhere between the center of the joint and the head and the nut.

When a preloaded joint is subjected to an external load F_e at loading planes 2 and 3 in Fig. 8, F_e relieves the compression load of the joint parts between planes 2 and 3. The remainder of the system, the bolt and the joint parts between planes 1-2 and 3-4, feel additional load due to F_e applied planes 2 and 3, the joint material between planes 2 and 3 is the clamped part and all other joint members, fastener and remaining joint material, are clamping parts. Because of the location of the loading planes, the joint diagram changes from black line to the blue line. Consequently, both the additional bolt load $F_{B\,max}$ decrease significantly when the loading planes of F_e shift from under the bolt head and nut toward the joint center.

Determination of the length of the clamped parts is, however, not that simple. First, it is assumed that the external load is applied at a plane perpendicular to the bolt axis. Second, the distance of the loading planes from each other has to be estimated. This distance may be expressed as the ratio of the length of clamped parts to the total joint length. Fig. 9 shows the effect of two different loading planes on the bolt load, both joints having the same preload F_i and the same external load F_e. The lengths of the clamped parts are estimated to be $0.75 l_J$ for joint A, and $0.25 l_J$ for joint B.

In general, the external bolt load is somewhere between $F_{eB} = 1\Phi F_e$ for loading planes under head and nut and $F_{eB} = 0\Phi F_e = 0$ when loading planes are in the joint center, as shown in Fig. 10. To consider the loading planes in calculation, the formula:

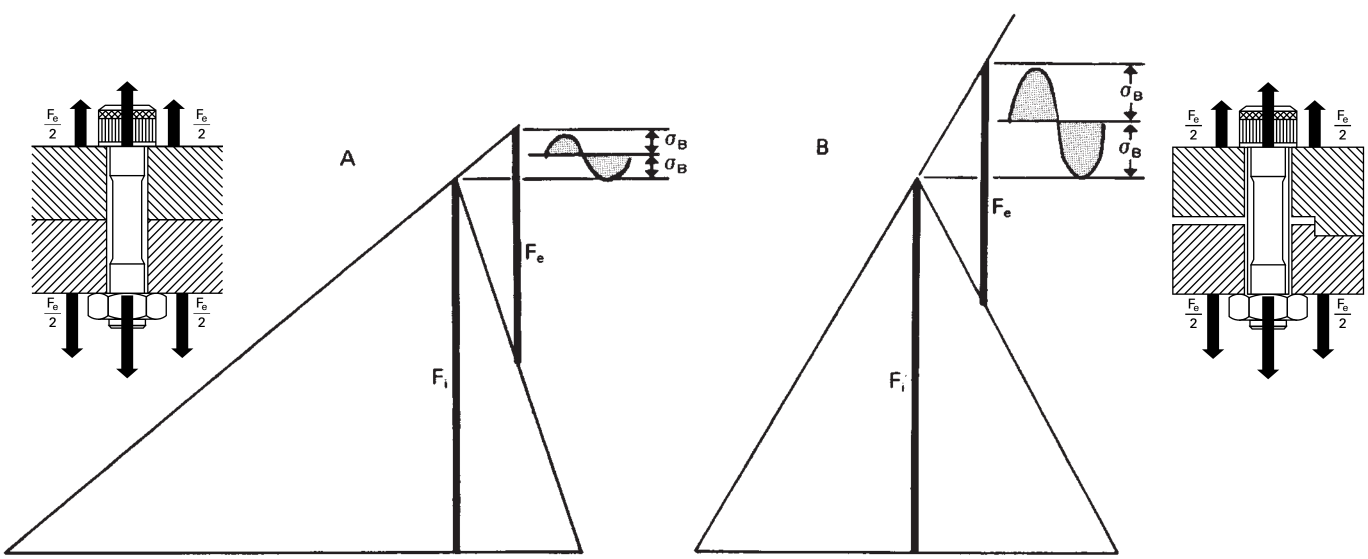


Fig. 6 Joint diagram of a springy bolt in a stiff joint (A), is compared to a diagram of a stiff bolt in a springy joint (B). Preload F_i and external load F_e are the same but diagrams show that alternating bolt stresses are significantly lower with a spring bolt in a stiff joint.

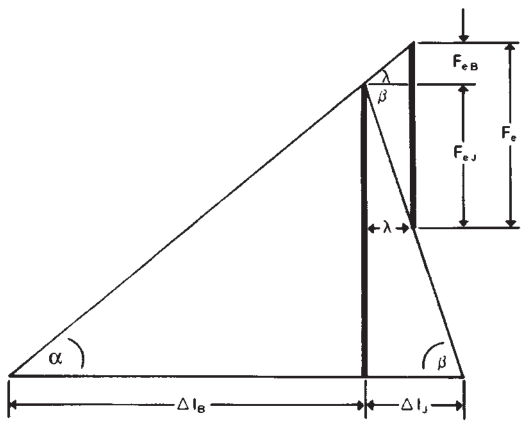


Fig. 7 Analysis of external load F_e and derivation of Force Ratio Φ.

$$\tan\alpha = \frac{F_i}{\Delta l_B} = K_B \text{ and } \tan\beta = \frac{Fi}{\Delta l_J} = K_J$$

$$\lambda = \frac{F_{eB}}{\tan\alpha} = \frac{F_{eJ}}{\tan\beta} = \frac{FeB}{K_B} = \frac{F_{eJ}}{K_J} \quad \text{or}$$

$$F_{eJ} = \lambda \tan\beta \text{ and } F_{eB} = \lambda \tan\alpha$$

Since $F_e = F_{eB} + F_{eJ}$

$$F_e = F_{eB} + \lambda \tan\beta$$

Substituting $\frac{F_{eB}}{\tan\alpha}$ for λ produces:

$$F_e = F_{eB} + \frac{F_{eB}\tan\beta}{\tan\alpha}$$

Multiplying both sides by tan α:

$$F_e \tan\alpha = F_{eB} (\tan\alpha + \tan\beta) \text{ and}$$

$$F_{eB} = \frac{F_e \tan\alpha}{\tan\alpha \tan\beta}$$

Substituting K_B for tan α and K_J for tan β

$$F_{eB} = F_e \frac{F_B}{K_B + K_J}$$

Defining $\Phi = \frac{K_B}{K_B + K_J}$

$$F_{eB} = \Phi F_e$$

$\Phi = \frac{F_{eB}}{F_e}$ and it becomes obvious why Φ is called force ratio.

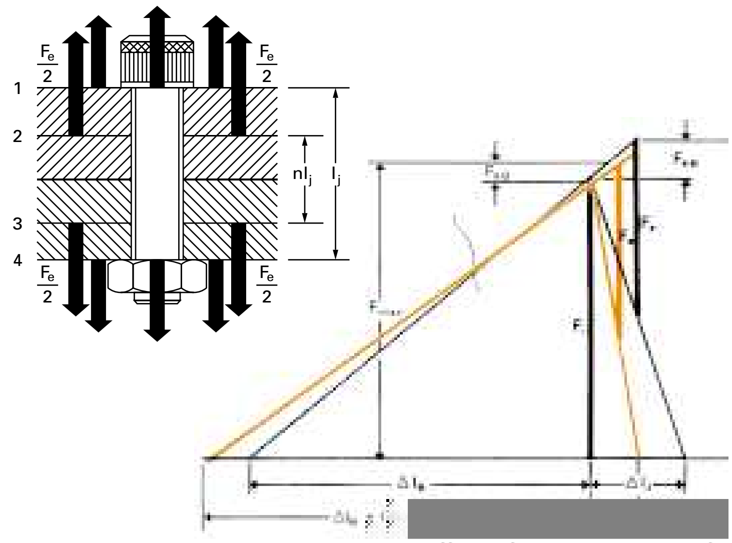


Fig. 8 Joint diagram shows effect of loading planes of F_e on bolt loads F_{eB} and $F_{B\ max}$. Black diagram shows F_{eB} and $F_{B\ max}$ resulting from F_e applied in planes 1 and 4. Orange diagram shows reduced bolt loads when Fe is applied in planes 2 and 3.

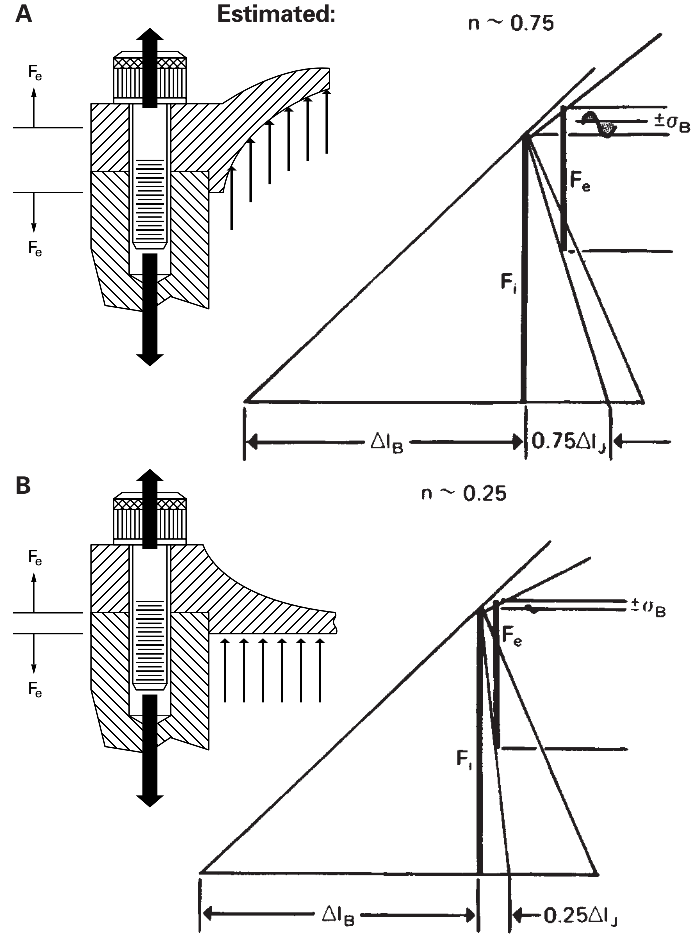


Fig. 9 When external load is applied relatively near bolt head, joint diagram shows resulting alternating stress α_B (A). When same value external load is applied relatively near joint center, lower alternating stress results (B).

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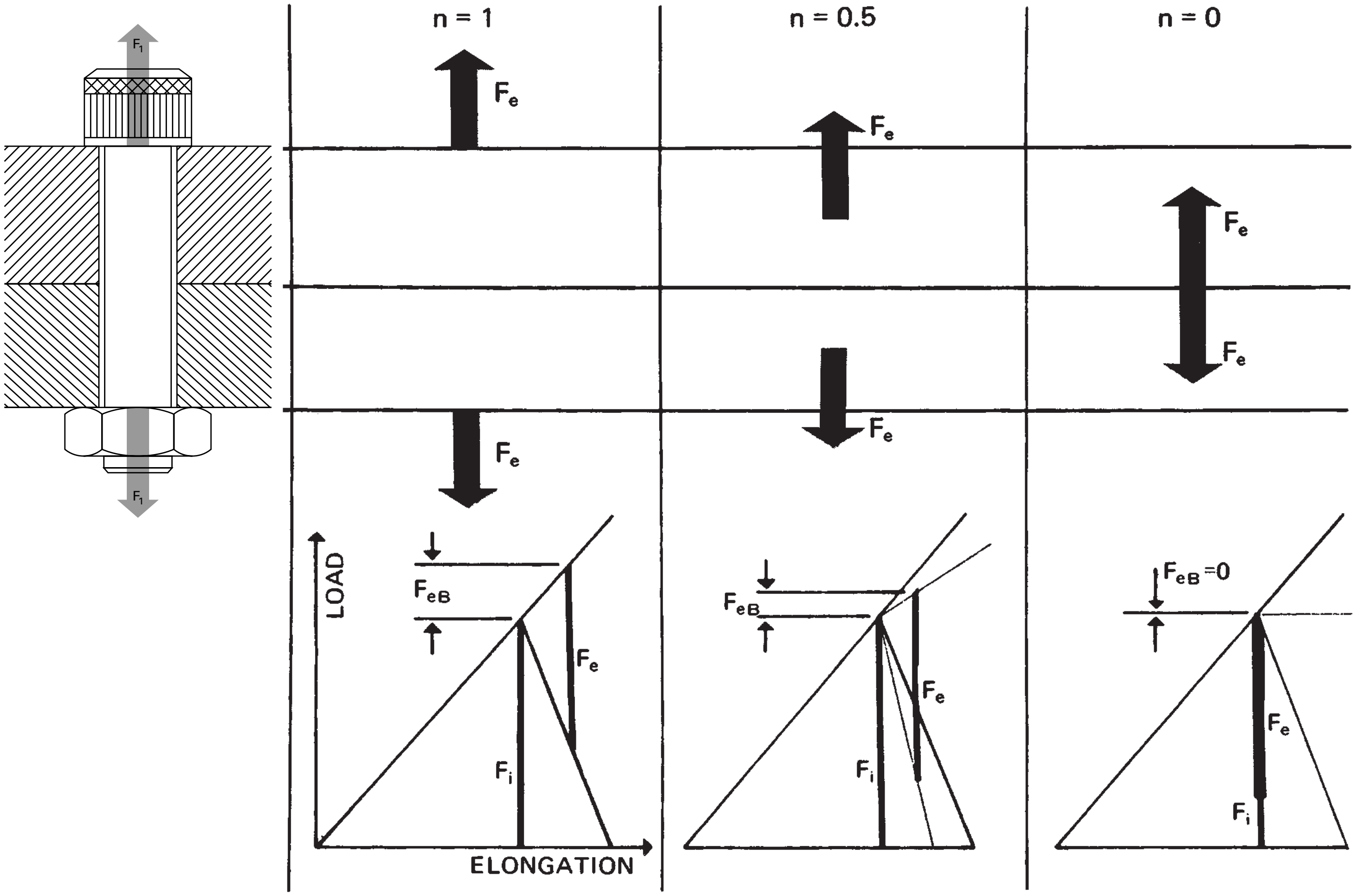

Fig. 10 Force diagrams show the effect of the loading planes of the external load on the bolt load.

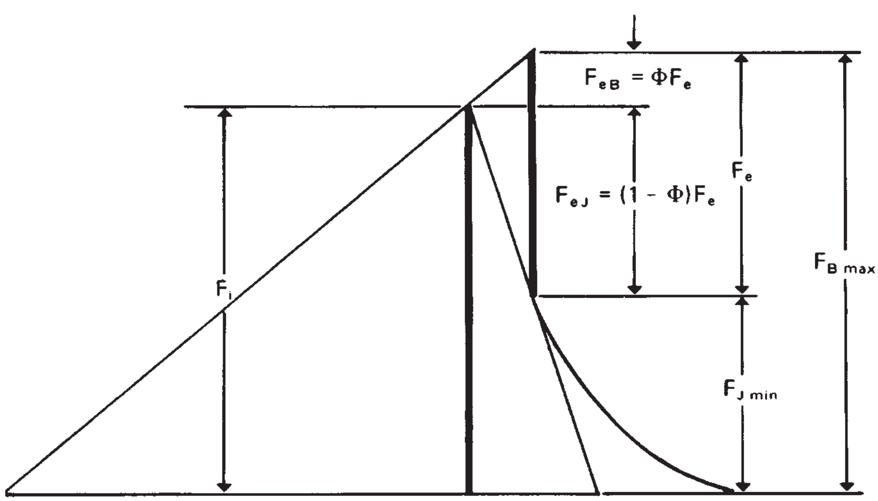

Fig. 11 Modified joint diagram shows nonlinear compression of joint at low preloads.

$F_{eB} = \Phi F_e$ must be modified to :

$F_{eB} = n \Phi F_e$

where n equals the ratio of the length of the clamped parts due to F_e to the joint length l_j. The value of n can range from 1, when F_e is applied under the head and nut, to O, when F_e is applies at the joint center. Consequently the stress amplitude:

$$\sigma_B = \pm \frac{\Phi F_e}{2 A_m} \quad \text{becomes}$$

$$\sigma_B = \pm \frac{n \Phi F_e}{2 A_m}$$

General Design Formulae

Hitherto, construction of the joint diagram has assumed linear resilience of both bolt and joint members. However, recent investigations have shown that this assumption is not quite true for compressed parts.

Taking these investigations into account, the joint diagram is modified to Fig. 11. The lower portion of the joint spring rate is nonlinear, and the length of the linear portion depends on the preload level F_i. The higher F_i the longer the linear portion. By choosing a sufficiently high minimum load, $F_{min}>2F_e$, the non-linear range of the joint spring rate is avoided and a linear relationship between F_{eB} and F_e is maintained.

Also from Fig. 11 this formula is derived:

$$F_{i\,min} = F_{J\,min} + (1 - \Phi) F_e + \Delta F_i$$

where ΔF_i is the amount of preload loss to be expected. For a properly designed joint, a preload loss $\Delta F_i = -(0.005 \text{ to } 0.10) F_i$ should be expected.

The fluctuation in bolt load that results from tightening is expressed by the ratio:

$$a = \frac{F_{i\,max}}{F_{i\,min}}$$

where a varies between 1.25 and 3.0 depending on the tightening method.

Considering a the general design formulae are:

$$F_{i\,nom} = F_{J\,min} = (1 - \Phi) F_e$$

$$F_{i\,max} = a\,[\,F_{j\,min} + (1 - \Phi) F_e + \Delta F_i\,]$$

$$F_{B\,max} = a\,[\,F_{j\,min} + (1 - \Phi) F_e + \Delta F_i\,] + \Phi F_e$$

Conclusion

The three requirements of concentrically loaded joints that must be met for an integral bolted joint are:

1. The maximum bolt load FB max must be less than the bolt yield strength.

2. If the external load is alternating, the alternating stress must be less than the bolt endurance limit to avoid fatigue failures.

3. The joint will not lose any preload due to permanent set or vibration greater than the value assumed for ΔF_i .

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SYMBOLS

A	Area (in.^{2})
Am	Area of minor thread diameter (in.^{2})
A_s	Area of substitute cyliner (in.^{2})
A_x	Area of bolt part 1_x (in.^{2})
d	Diameter of minor thread (in.)
D_c	Outside diameter of bushing (cylinder) (in.)
D_H	Diameter of Bolt head (in.)
D_h	Diameter of hole (in.)
D_J	Diameter of Joint
E	Modulus of Elasticity (psi)
F	Load (lb)
F_e	External load (lb.)
F_{eB}	Additinal Bolt Load due to external load (lb)
F_{eJ}	Reduced Joint load due to external load (lb)
F_i	Preload on Bolt and Joint (lb)
ΔF_i	Preload loss (–lb)
$F_{i\,min}$	Minimum preload (lb)
$F_{i\,max}$	Maximum preload (lb)
$F_{j\,nom}$	Nominal preload (lb)
$F_{B\,max}$	Maximum Bolt load (lb)
$F_{J\,min}$	Minimum Joint load (lb)
K	Spring rate (lb/in.)
K_B	Spring rate of Bolt (lb/in.)
K_J	Spring rate of Joint (lb/in.)
K_x	Spring rate of Bolt part l_x (lb/in.)
l	Length (in.)
Δl	Change in length (in.)
l_B	Length of Bolt (in.)
Δl_B	Bolt elongation due to F_i (in.)
l_J	Length of Joint (in.)
Δl_J	Joint compression to F_i (in.)
l_x	Length of Bolt part x (in.)
n	$\frac{\text{Length of clamped parts}}{\text{Total Joint Length}}$
α	Tightening factor
Φ	Force ratio
λ	Bolt and Joint elongation due to F_e (in.)
σ_B	Bolt stress amplitude (± psi)

TIGHTENING TORQUES AND THE TORQUE-TENSION RELATIONSHIP

All of the analysis and design work done in advance will have little meaning if the proper preload is not achieved. Several discussions in this technical section stress the importance of preload to maintaining joint integrity. There are many methods for measuring preload (see Table 12). However, one of the least expensive techniques that provides a reasonable level of accuracy versus cost is by measuring torque. The fundamental characteristic required is to know the relationship between torque and tension for any particular bolted joint. Once the desired design preload must be identified and specified first, *then* the torque required to induce that preload is determined.

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Within the elastic range, before permanent stretch is induced, the relationship between torque and tension is essentially linear (see figure 13). Some studies have found up to 75 variables have an effect on this relationship: materials, temperature, rate of installation, thread helix angle, coefficients of friction, etc. One way that has been developed to reduce the complexity is to depend on empirical test results. That is, to perform experiments under the application conditions by measuring the induced torque and recording the resulting tension. This can be done with relatively simple, calibrated hydraulic pressure sensors, electric strain gages, or piezoelectric load cells. Once the data is gathered and plotted on a chart, the slope of the curve can be used to calculate a correlation factor. This technique has created an accepted formula for relating torque to tension.

$T = K \times D \times P$

T = torque, lbf.-in.
D = fastener nominal diameter, inches
P = preload, lbf.
K = “nut factor,” “tightening factor,” or “k-value”

If the preload and fastener diameter are selected in the design process, and the K-value for the application conditions is known, then the necessary torque can be calculated. It is noted that even with a specified torque, actual conditions at the time of installation can result in variations in the actual preload achieved (see Table 12).

One of the most critical criteria is the selection of the K-value. Accepted nominal values for many industrial applications are:

K = 0.20 for as-received steel bolts into steel holes

K = 0.15 steel bolts with cadmium plating, which acts like a lubricant,

K = 0.28 steel bolts with zinc plating.

The K-value is not the coefficient of the friction (μ); it is an empirically derived correlation factor.

It is readily apparent that if the torque intended for a zinc plated fastener is used for cadmium plated fastener, the preload will be almost two times that intended; it may actually cause the bolt to break.

Another influence is where friction occurs. For steel bolts holes, approximately 50% of the installation torque is consumed by friction under the head, 35% by thread friction, and only the remaining 15% inducing preload tension. Therefore, if lubricant is applied just on the fastener underhead, full friction reduction will not be achieved. Similarly, if the material against which the fastener is bearing, e.g. aluminum, is different than the internal thread material, e.g. cast iron, the effective friction may be difficult to predict, These examples illustrate the importance and the value of identifying the torque-tension relationship. It is a recommend practice too contact the lubricant manufacturer for K-value information if a lubricant will be used.

The recommended seating torques for Unbrako headed socket screws are based on inducing preloads reasonably expected in practice for each type. The values for Unbrako metric fasteners are calculated using VDI2230, a complex method utilized extensively in Europe. All values assume use in the received condition in steel holes. It is understandable the designer may need preloads higher than those listed. The following discussion is presented for those cases.

TORSION-TENSION YIELD AND TENSION CAPABILITY AFTER TORQUING

Once a headed fastener has been seated against a bearing surface, the inducement of torque will be translated into both torsion and tension stresses. These stresses combine to induce twist. If torque continues to be induced, the stress along the angle of twist will be the largest stress *while the bolt is being torqued.* Consequently, the stress along the bolt axis (axial tension) will be something less. This is why a bolt can fail at a lower tensile stress *during installation* than when it is pulled in straight tension alone, eg . a tensile test. Research has indicated the axial tension can range from 135,000 to 145,000 PSI for industry socket head cap screws at torsion-tension yield, depending on diameter. Including the preload variation that can occur with various installation techniques, eg. up to 25%, it can be understood why some recommended torques induce preload reasonably lower than the yield point.

Figure 13 also illustrates the effect of straight tension applied after installation has stopped. Immediately after stopping the installation procedure there will be some relaxation, and the torsion component will drop toward zero. This leaves only the axial tension, which keeps the joint clamped together. Once the torsion is relieved, the axial tension yield value and ultimate value for the fastener will be appropriate.

Table 12
Industrial Fasteners Institute’s
Torque-Measuring Method

Preload Measuring Method	Accuracy Percent	Relative Cost
Feel (operator’s judgement)	±35	1
Torque wrench	±25	1.5
Turn of the nut	±15	3
Load-indicating washers	±10	7
Fastener elongation	±3 to 5	15
Strain gages	±1	20

Fig. 14

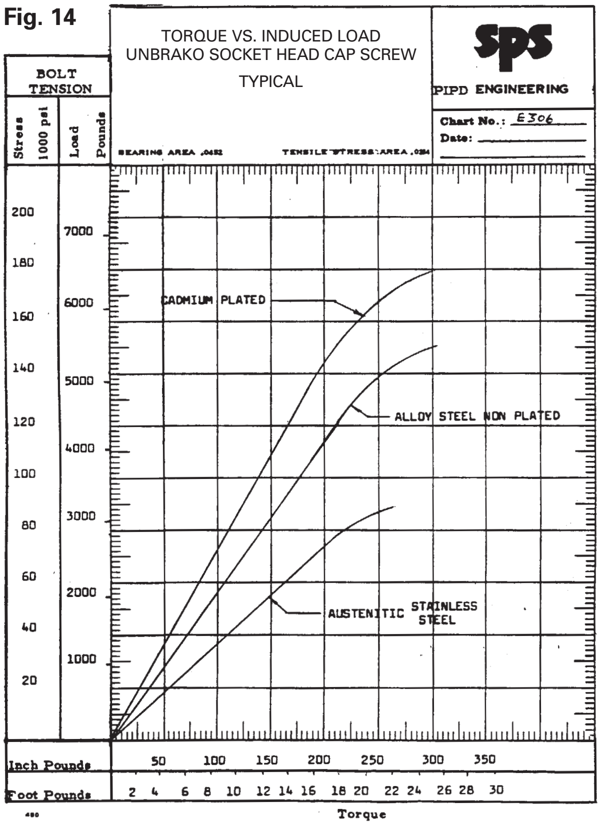

Fig. 13 Torque/Tension Relationship

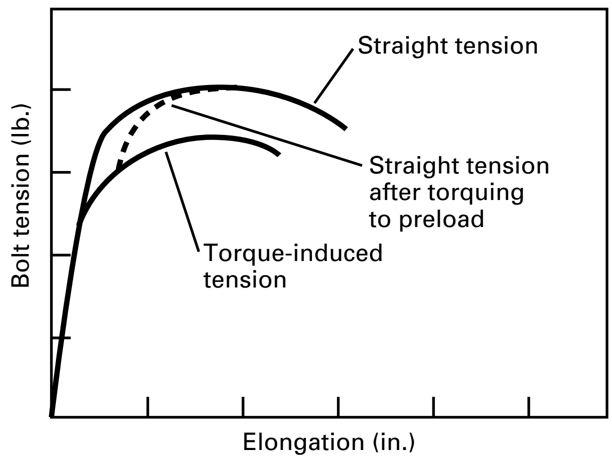

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Fig. 15 Recommended Seating Torques (Inch-Lb.) for Application in Various Materials UNBRAKO pHd (1960 Series) Socket Head Cap Screws

	mild steel Rb 87 cast iron Rb 83 note 1		brass Rb 72 note 2		aluminum Rb 72 (2024-T4) note 3	
	UNC	UNF	UNC	UNF	UNC	UNF
screw size	plain	plain	plain	plain	plain	plain
#0	–	*2.1	–	*2.1	–	*2.1
#1	*3.8	*4.1	*3.8	*4.1	*3.8	*4.1
#2	*6.3	*6.8	*6.3	*6.8	*6.3	*6.8
#3	*9.6	*10.3	*9.6	*10.3	*9.6	*10.3
#4	*13.5	*14.8	*13.5	*14.8	*13.5	*14.8
#5	*20	*21	*20	*21	*20	*21
#6	*25	*28	*25	*28	*25	*28
#8	*46	*48	*46	*48	*46	*48
#10	*67	*76	*67	*76	*67	*76
1/4	*158	*180	136	136	113	113
5/16	*326	*360	228	228	190	190
3/8	*580	635	476	476	397	397
7/16	*930	*1,040	680	680	570	570
1/2	*1,420	*1,590	1,230	1,230	1,030	1,030
9/16	*2,040	2,250	1,690	1,690	1,410	1,410
5/8	*2,820	3,120	2,340	2,340	1,950	1,950
3/4	*5,000	5,340	4,000	4,000	3,340	3,340
7/8	*8,060	8,370	6,280	6,280	5,230	5,230
1	*12,100	12,800	9,600	9,600	8,000	8,000
1 1/8	*13,800	*15,400	13,700	13,700	11,400	11,400
1 1/4	*19,200	*21,600	18,900	18,900	15,800	15,800
1 3/8	*25,200	*28,800	24,200	24,200	20,100	20,100
1 1/2	*33,600	*36,100	32,900	32,900	27,400	27,400

NOTES:
1. Torques based on 80,000 psi bearing stress under head of screw.
2. Torques based on 60,000 psi bearing stress under head of screw.
3. Torques based on 50,000 psi bearing stress under head of screw.

*Denotes torques based on 100,000 psi tensile stress in screw threads up to 1" dia., and 80,000 psi for sizes 1 1/8" dia. and larger.

To convert inch-pounds to inch-ounces – multiply by 16.

To convert inch-pounds to foot-pounds – divide by 12.

STRIPPING STRENGTH OF TAPPED HOLES

Charts and sample problems for obtaining minimum thread engagement based on applied load, material, type of thread and bolt diameter.

Knowledge of the thread stripping strength of tapped holes is necessary to develop full tensile strength of the bolt or, for that matter, the minimum engagement needed for any lesser load.

Conversely, if only limited length of engagement is available, the data help determine the maximum load that can be safely applied without stripping the threads of the tapped hole.

Attempts to compute lengths of engagement and related factors by formula have not been entirely satisfactory-mainly because of subtle differences between various materials. Therefore, strength data has been empirically developed from a series of tensile tests of tapped specimens for seven commonly used metals including steel, aluminum, brass and cast iron.

The design data is summarized in the six accompanying charts, (Charts E504-E509), and covers a range of screw thread sizes from #0 to one inch in diameter for both coarse and fine threads. Though developed from tests of Unbrako socket head cap screws having minimum ultimate tensile strengths (depending on the diameter) from 190,000 to 180,000 psi , these stripping strength values are valid for all other screws or bolts of equal or lower strength having a standard thread form. Data are based on static loading only.

In the test program, bolts threaded into tapped specimens of the metal under study were stressed in tension until the threads stripped. Load at which stripping occurred and the length of engagement of the specimen were noted. Conditions of the tests, all of which are met in a majority of industrial bolt applications, were:

- Tapped holes had a basic thread depth within the range of 65 to 80 per cent. Threads of tapped holes were Class 2B fit or better.
- Minimum amount of metal surrounding the tapped hole was 2 1/2 times the major diameter.
- Test loads were applied slowly in tension to screws having standard Class 3A threads. (Data, though, will be equally applicable to Class 2A external threads as well.)
- Study of the test results revealed certain factors that greatly simplified the compilation of thread stripping strength data:
- Stripping strengths are almost identical for loads applied either by pure tension or by screw torsion. Thus data are equally valid for either condition of application.
- Stripping strength values vary with diameter of screw. For a given load and material, larger diameter bolts required greater engagement.
- Minimum length of engagement (as a percent of screw diameter) is a straight line function of load. This permits easy interpolation of test data for any intermediate load condition.
- When engagement is plotted as a percentage of bolt diameter, it is apparent that stripping strengths for a wide range of screw sizes are close enough to be grouped in a single curve. Thus , in the accompanying charts, data for sizes #0 through #12 have been represented by a single set of curves.

With these curves, it becomes a simple matter to determine stripping strengths and lengths of engagement for any condition of application. A few examples are given below:

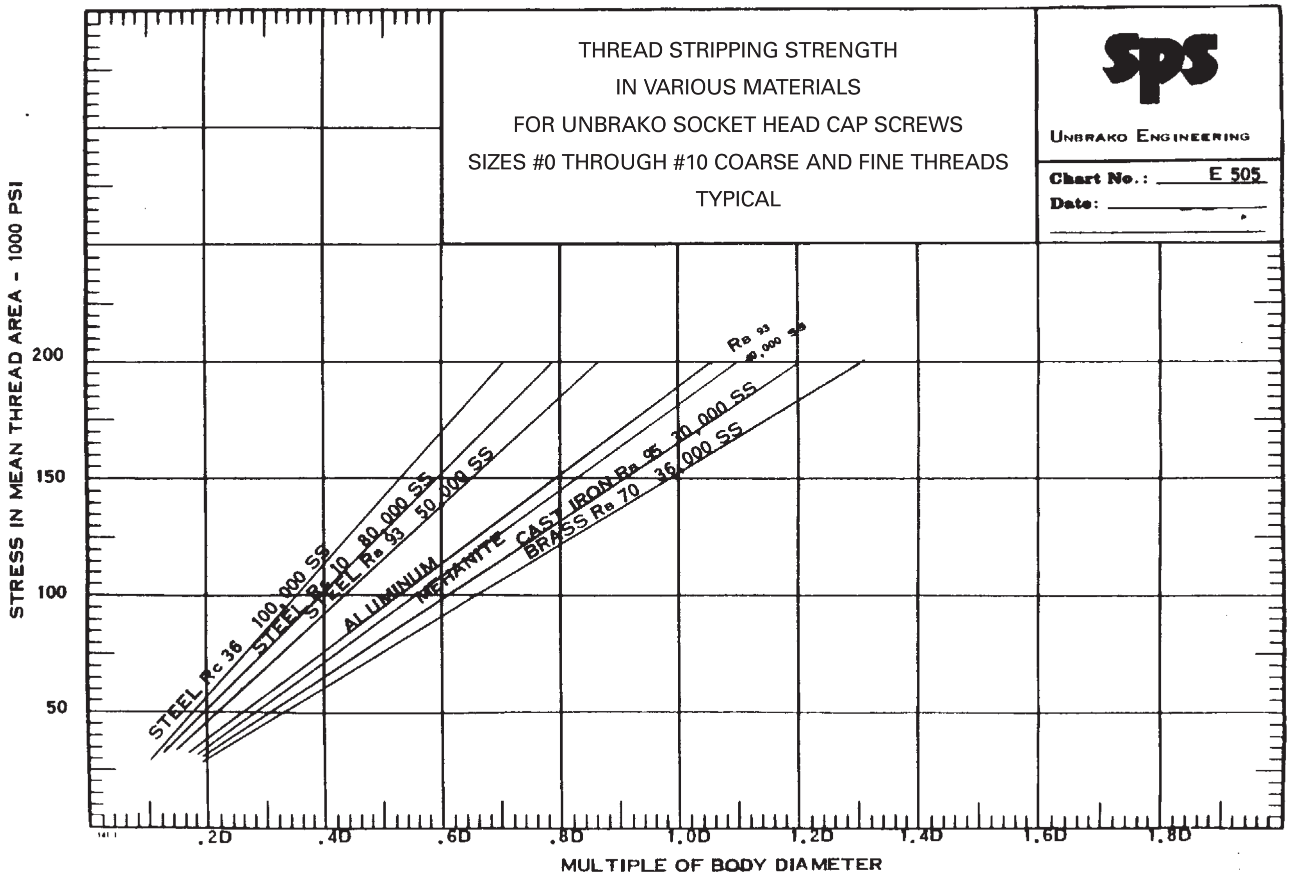
Example 1. Calculate length of thread engagement necessary to develop the minimum ultimate tensile strength (190,000 psi) of a 1/2–13 (National Coarse) Unbrako cap screw in cast iron having an ultimate shear strength of 30,000 psi. E505 is for screw sizes from #0 through #10; E506 and E507 for sizes from 1/4 in. through 5/8 in.; E508 and E509 for sizes from 3/4 in. through 1 in. Using E506 a value 1.40D is obtained. Multiplying nominal bolt diameter (0.500 in.) by 1.40 gives a minimum length of engagement of 0.700 in.

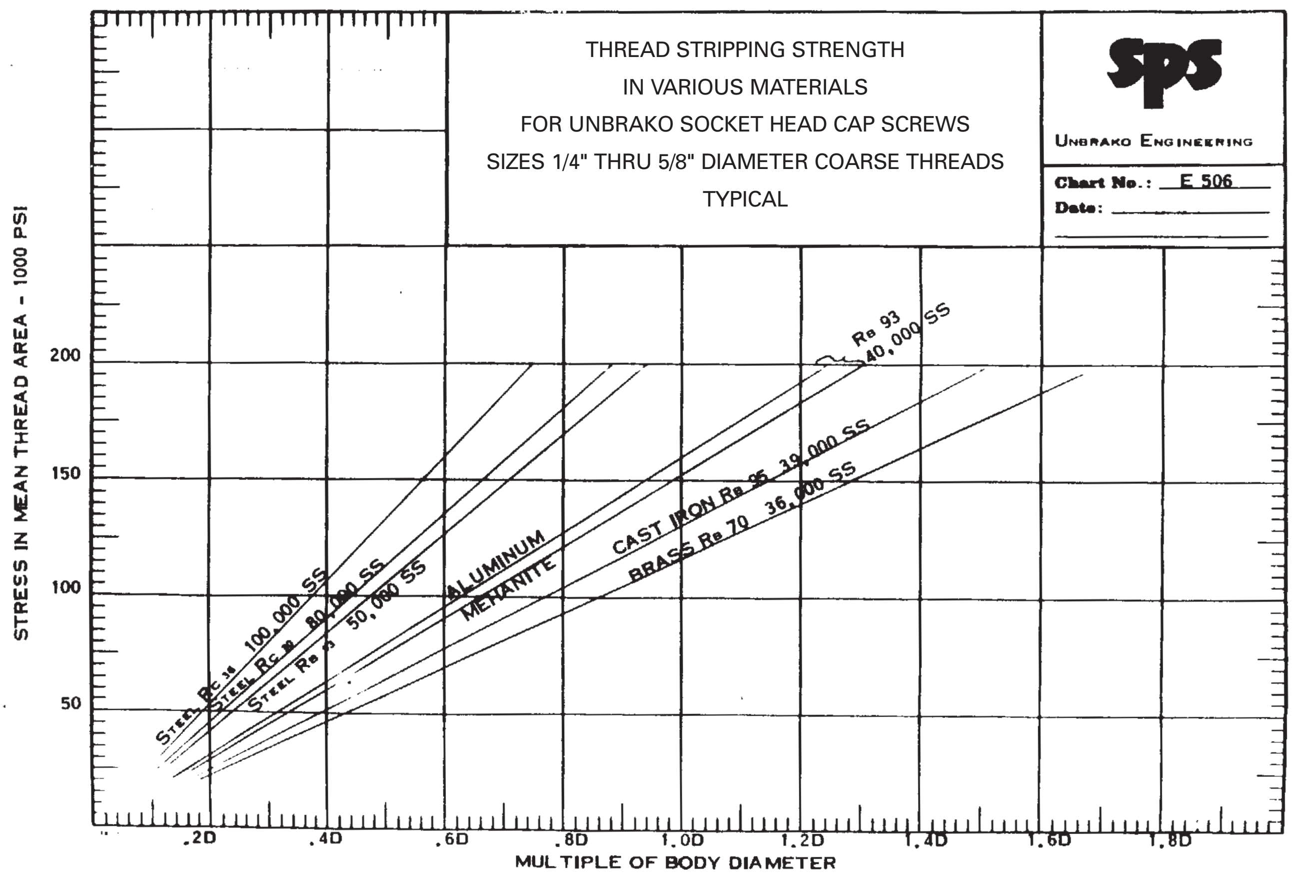
Example 2. Calculate the length of engagement for the above conditions if only 140,000 psi is to be applied. (This is the same as using a bolt with a maximum tensile strength of 140,000psi.) From E506 obtain value of 1.06D Minimum length of engagement = (0.500) (1.06) = 0.530.

Example 3. Suppose in Example 1 that minimum length of engagement to develop full tensile strength was not available because the thickness of metal allowed a tapped hole of only 0.600 in. Hole depth in terms of bolt dia. = 0.600/0.500 = 1.20D. By working backwards in Fig. 2, maximum load that can be carried is approximately 159,000 psi.

Example 4. Suppose that the hole in Example 1 is to be tapped in steel having an ultimate shear strength 65,000 psi. There is no curve for this steel in E506 but a design value can be obtained by taking a point midway between curves for the 80,000 psi and 50,000 psi steels that are listed. Under the conditions of the example, a length of engagement of 0.825D or 0.413 in. will be obtained.

STRIPPING STRENGTH OF TAPPED HOLES





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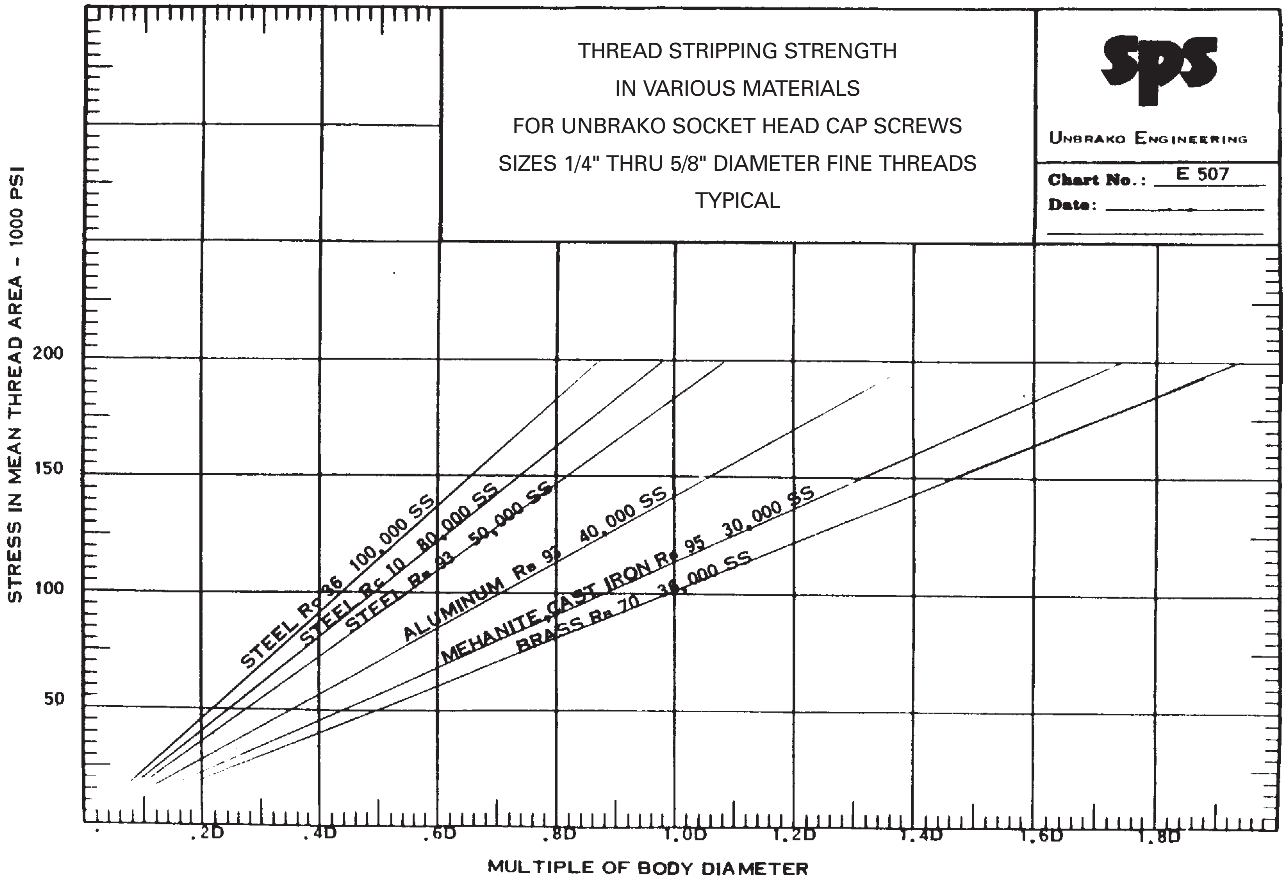
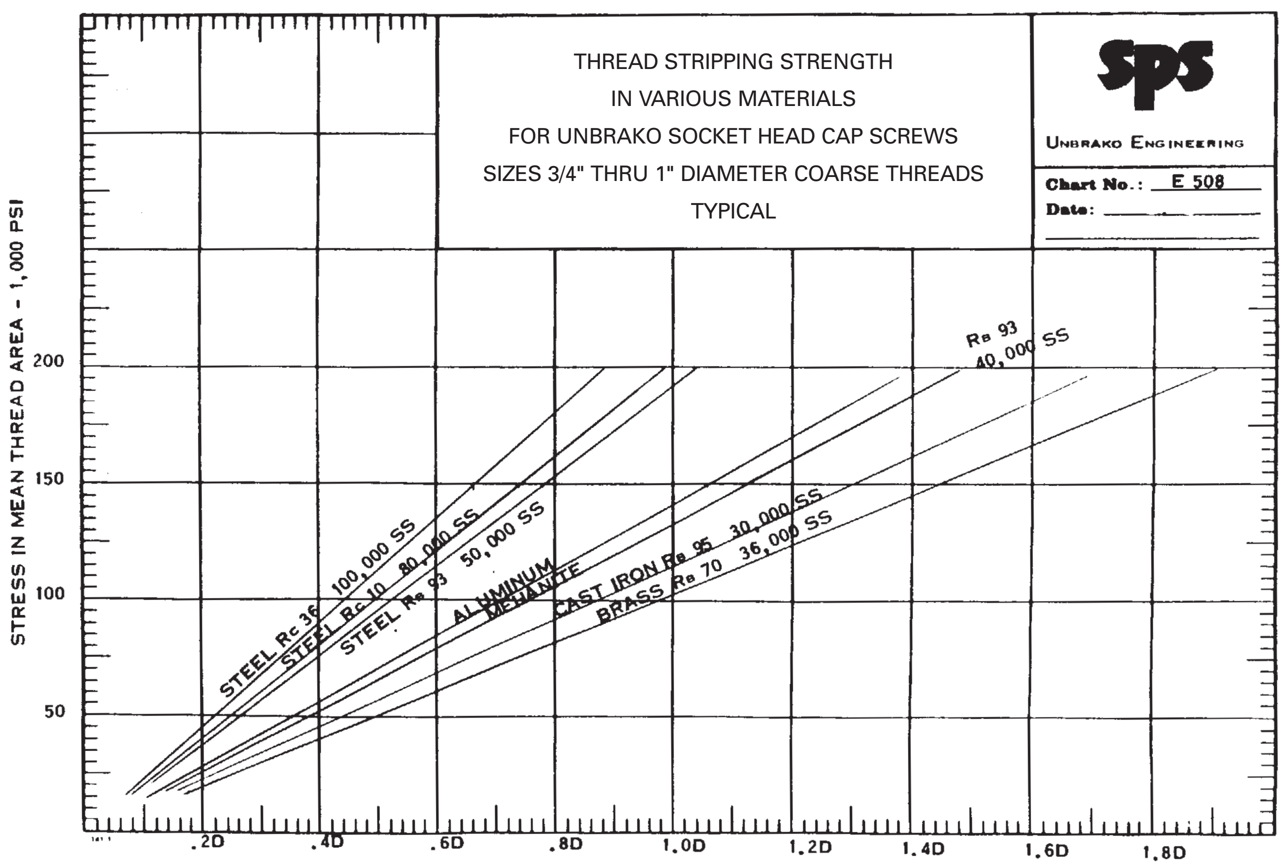
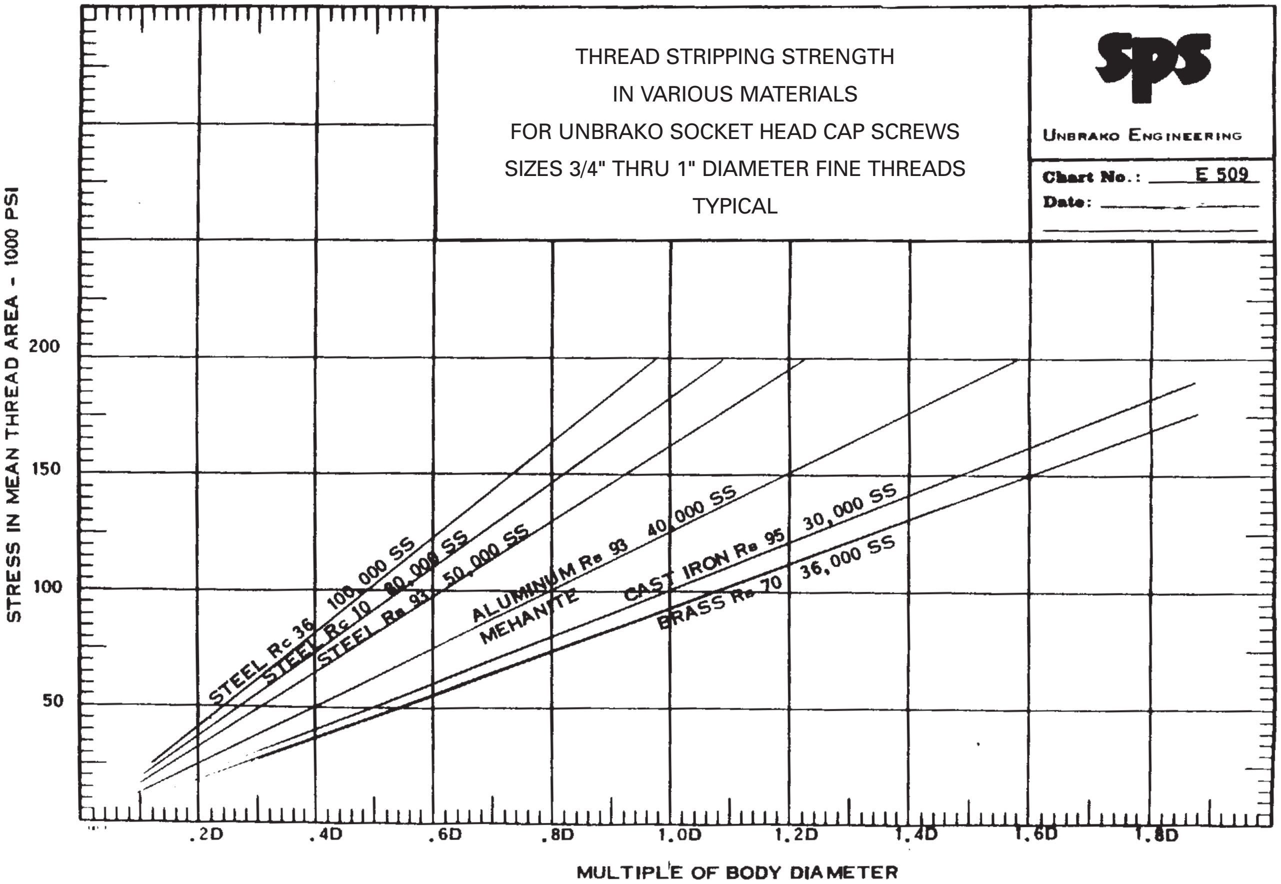


THREAD STRIPPING STRENGTH
IN VARIOUS MATERIALS
FOR UNBRAKO SOCKET HEAD CAP SCREWS
SIZES 1/4" THRU 5/8" DIAMETER FINE THREADS
TYPICAL
SPS
UNBRAKO ENGINEERING
Chart No.: E 507
Date:
STRESS IN MEAN THREAD AREA - 1000 PSI
200
150
100
50
STEEL Rc 36 100,000 SS
STEEL Rc 10 80,000 SS
STEEL Rв 93 50,000 SS
ALUMINUM Rв 93 40,000 SS
MEHANITE,CAST IRON Rв 95 30,000 SS
BRASS Rв 70 36,000 SS
.2D
.4D
.6D
.8D
1.0D
1.2D
1.4D
1.6D
1.8D
MULTIPLE OF BODY DIAMETER



THREAD STRIPPING STRENGTH
IN VARIOUS MATERIALS
FOR UNBRAKO SOCKET HEAD CAP SCREWS
SIZES 3/4" THRU 1" DIAMETER COARSE THREADS
TYPICAL
SPS
UNBRAKO ENGINEERING
Chart No.: E 508
Date:
STRESS IN MEAN THREAD AREA - 1,000 PSI
200
150
100
50
STEEL Rc 36 100,000 SS
STEEL Rc 10 80,000 SS
STEEL Rв 93 50,000 SS
ALUMINUM Rв 93 40,000 SS
MEHANITE
CAST IRON Rв 95 30,000 SS
BRASS Rв 70 36,000 SS
.2D
.4D
.6D
.8D
1.0D
1.2D
1.4D
1.6D
1.8D

STRIPPING STRENGTH OF TAPPED HOLES



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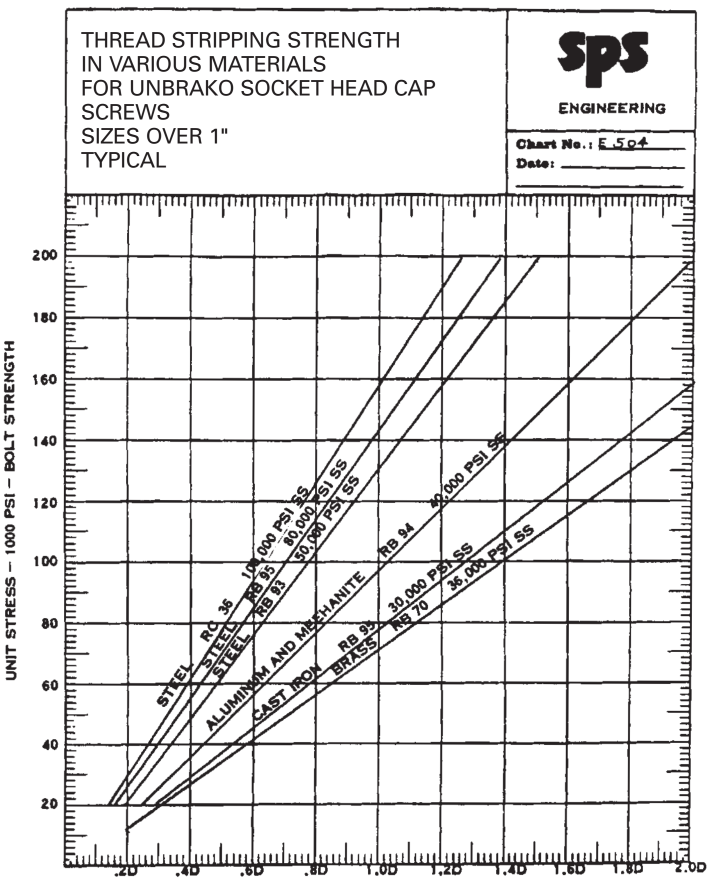


HIGH-TEMPERATURE JOINTS

Bolted joints subjected to cyclic loading perform best if an initial preload is applied. The induced stress minimizes the external load sensed by the bolt, and reduces the chance of fatigue failure. At high temperature, the induced load will change, and this can adeversely affect the fastener performance. It is therefore necessary to compensate for high-temperature conditions when assembling the joint at room temperature. This article describes the factors which must be considered and illustrates how a high-temperature bolted joint is designed.

II

In high-temperature joints, adequate clamping force or preload must be maintained in spite of temperature-induced dimensional changes of the fastener relative to the joint members. the change in preload at any given temperature for a given time can be calculated, and the affect compensated for by proper fastener selection and initial preload.

Three principal factors tend to alter the initial clamping force in a joint at elevated temperatures, provided that the fastener material retains requisite strength at the elevated temperature. These factors are: Modulus of elasticity, coefficient of thermal expansion, and relaxation.

Modulus Of Elasticity: As temperature increases, less stress or load is needed to impart a given amount of elongation or strain to a material than at lower temperatures. This means that a fastener stretched a certain amount at room temperature to develop a given preload will exert a lower clamping force at higher temperature if there is no change in bolt elongation.

Coefficient of Expansion: With most materials, the size of the part increases as the temperature increases. In a joint, both the structure and the fastener grow with an increase in temperature, and this can result,depending on the materials, in an increase or decrease in the clamping force. Thus, matching of materials in joint design can assure sufficient clamping force at both room and elevated temperatures. Table 16 lists mean coefficient of thermal expansion of certain fastener alloys at several temperatures.

Relaxation: At elevated temperatures, a material subjected to constant stress below its yield strength will flow plastically and permanently change size. This phenomenon is called creep. In a joint at elevated temperature, a fastener with a fixed distance between the bearing surface of the head and nut will produce less and less clamping force with time. This characteristic is called relaxation. It differs from creep in that stress changes while elongation or strain remains constant. Such elements as material, temperature, initial stress, manufacturing method, and design affect the rate of relaxation.

Relaxation is the most important of the three factors. It is also the most critical consideration in design of elevated-temperature fasteners. A bolted joint at 1200°F can lose as much as 35 per cent of preload. Failure to compensate for this could lead to fatigue failure through a loose joint even though the bolt was properly tightened initially.

If the coefficient of expansion of the bolt is greater than that of the joined material, a predictable amount of clamping force will be lost as temperature increases. Conversely, if the coefficient of the joined material is greater, the bolt may be stressed beyond its yield or even fracture strength. Or, cyclic thermal stressing may lead to thermal fatigue failure.

Changes in the modulus of elasticity of metals with increasing temperature must be anticipated, calculated, and compensated for in joint design. Unlike the coefficient of expansion, the effect of change in modulus is to reduce clamping force whether or not bolt and structure are the same material, and is strictly a function of the bolt metal.

Since the temperature environment and the materials of the structure are normally "fixed," the design objective is to select a bolt material that will give the desired clamping force at all critical points in the operating range of the joint. To do this, it is necessary to balance out the three factors-relaxation, thermal expansion, and modulus-with a fourth, the amount of initial tightening or clamping force.

In actual joint design the determination of clamping force must be considered with other design factors such as ultimate tensile, shear, and fatigue strength of the fastener at elevated temperature. As temperature increases the inherent strength of the material decreases. Therefore, it is important to select a fastener material which has sufficient strength at maximum service temperature.

Example

The design approach to the problem of maintaining satisfactory elevated-temperature clamping force in a joint can be illustrated by an example. The example chosen is complex but typical. A cut-and-try process is used to select the right bolt material and size for a given design load under a fixed set of operating loads and environmental conditions, Fig.17.

The first step is to determine the change in thickness, Δt, of the structure from room to maximum operating temperature.

For the AISI 4340 material:

$$\Delta t_1 = t_1(T_2 - T_1)\alpha$$

$$\Delta t_1 = (0.05)(800 - 70)\ (7.4 \times 10^{-6})$$

$$\Delta t_1 = 0.002701 \text{ in.}$$

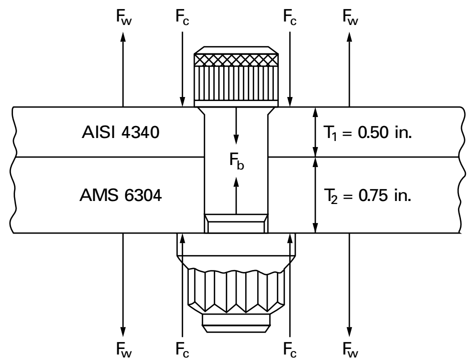
For the AMS 6304 material:

$$\Delta t_2 = (0.75)(800 - 70)(7.6 \times 10^{-6})$$

$$\Delta t_2 = 0.004161 \text{ in.}$$

The total increase in thickness for the joint members is 0.00686 in.

The total effective bolt length equals the total joint thickness plus one-third of the threads engaged by the nut. If it is assumed that the smallest diameter bolt should be used for weight saving, then a 1/4-in. bolt should be tried. Thread engagement is approximately one diameter, and the effective bolt length is:



d = Bolt diam, in.

E = Modulus of elasticity, psi

F_b = Bolt preload, lb

F_c = Clamping force, lb (F_b=F_c)

F_w = Working load=1500 lb static + 100 lb cyclic

L = Effective bolt length, inc.

T_1 = Room temperature= 70°F

T_2 = Maximum operatng temperature for 1000 hr=800°F

t = Panel thickness, in.

a = Coefficient of thermal expansion

Fig. 17 – Parameters for joint operating at 800°F.

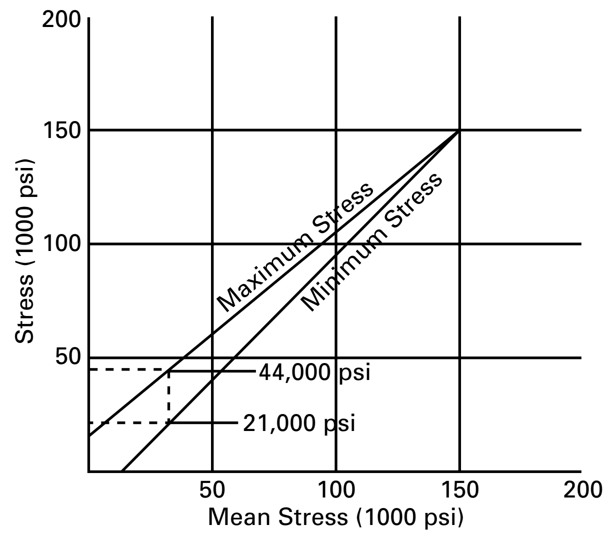


Fig. 18 – Goodman diagram of maximum and minimum operating limits for H-11 fastener at 800°F. Bolts stressed within these limits will give infinite fatigue life.

$L = t_1 + t_2 + (1/3\ d)$

$L = 0.50 + 0.75 + (1/3 \times 0.25)$

$L = 1.333$ in.

The ideal coefficient of thermal expansion of the bolt material is found by dividing the total change in joint thickness by the bolt length times the change in temperature.

$$\alpha b = \frac{\Delta t}{L \times \Delta t}$$

$$\alpha = \frac{.00686}{(1.333)(800 - 70)} = 7.05 \times 10^{-6} \text{ in./in./deg. F}$$

The material, with the nearest coefficient of expansion is with a value of 9,600,000 at 800°F.

To determine if the bolt material has sufficient strength and resistance to fatigue, it is necessary to calculate the stress in the fastener at maximum and minimum load. The bolt load plus the cyclic load divided by the tensile stress of the threads will give the maximum stress. For a 1/4-28 bolt, tensile stress area,from thread handbook H 28, is 0.03637 sq. in. The maximum stress is

$$S_{max} = \frac{\text{Bolt load}}{\text{Stress area}} = \frac{1500 + 100}{0.03637}$$

S_{max} = 44,000 psi

and the minimum bolt stress is 41,200 psi.

H-11 has a yield strength of 175,000 psi at 800°F, Table 3, and therefore should be adequate for the working loads.

A Goodman diagram, Fig. 18, shows the extremes of stress within which the H-11 fastener will not fail by fatigue. At the maximum calculated load of 44,000 psi, the fastener will withstand a minimum cyclic loading at 800°F of about 21,000 psi without fatigue failure.

Because of relaxation, it is necessary to determine the initial preload required to insure 1500-lb. clamping force in the joint after 1000 hr at 800°F.

When relaxation is considered, it is necessary to calculate the maximum stress to which the fastener is subjected. Because this stress is not constant in dynamic joints, the resultant values tend to be conservative. Therefore, a maximum stress of 44,000 psi should be considered although the necessary stress at 800°F need be only 41,200 psi. Relaxation at 44,000 psi can be interpolated from the figure, although an actual curve could be constructed from tests made on the fastener at the specific conditions.

The initial stress required to insure a clamping stress of 44,000 psi after 1000 hr at 800°F can be calculated by interpolation.

$x = 61{,}000 - 44{,}000 = 17{,}000$
$y = 61{,}000 - 34{,}000 = 27{,}000$
$B = 80{,}000 - 50{,}000 = 30{,}000$
$A = 80{,}000 - C$

$$\frac{x}{y} = \frac{A}{B} \quad \frac{17{,}000}{27{,}000} = \frac{80{,}000 - C}{30{,}000}$$

C = 61,100 psi

The bolt elongation required at this temperature is calculated by dividing the stress by the modulus at temperature and multiplying by the effective length of the bolt. That is: $(61{,}000 \times 1.333)/24.6 \times 10^6 = 0.0033$

Since the joint must be constructed at room temperature, it is necessary to determine the stresses at this state. Because the modulus of the fastener material changes with temperature, the clamping force at room temperature will not be the same as at 800°F. To determine the clamping stress at assembly conditions, the elongation should be multiplied by the modulus of elasticity at room temperature.

$.0033 \times 30.6 \times 10^6 = 101{,}145$ psi

The assembly conditions will be affected by the difference between th ideal and actual coefficients of expansion of the joint. The ideal coeffienct for the fastener material was calculated to be 7.05 but the closest material – H-11 – has a coefficient of 7.1. Since this material has a greater expansion than calculated, there will be a reduction in clamping force resulting from the increase in temperature. This amount equals the difference between the ideal and the actual coefficients multiplied by the change in temperature, the length of the fastener, and the modulus of elasticity at 70°F.

II

$[(7.1 - 7.05) \times 10^{-6}][800 - 70][1.333] \times$

$[30.6 \times 10^6] = 1{,}490$ psi

The result must be added to the initial calculated stresses to establish the minimum required clamping stress needed for assembling the joint at room temperature.

$101{,}145 + 1{,}490 = 102{,}635$ psi

Finally, the method of determining the clamping force or preload will affect the final stress in the joint at operating conditions. For example, if a torque wrench is used to apply preload (the most common and simplest method available), a plus or minus 25 per cent variation in induced load can result. Therefore, the maximum load which could be expected in this case would be 1.5 times the minimum, or:

$(1.5)(102{,}635) = 153{,}950$ psi

This value does not exceed the room-temperature yield strength for H-11 given in Table 19.

Since there is a decrease in the clamping force with an increase in temperature and since the stress at operating temperature can be higher than originally calculated because of variations in induced load, it is necessary to ascertain if yield strength at 800°F will be exceeded

$$\frac{(\text{max stress at } 70°\text{F} + \text{change in stress}) \times E \text{ at } 800°\text{F}}{E \text{ at } 70°\text{F}}$$

$$\frac{[153{,}950 + (-1490)] \times 24.6 \times 10^6}{30.6 \times 10^6} = 122{,}565$$

This value is less than the yield strength for H-11 at 800°F, Table 19. Therefore, a 1/4-28 H-11 bolt stressed between 102,635 psi and 153,950 psi at room temperature will maintain a clamping load 1500 lb at 800°F after 1000 hr of operation. A cyclic loading of 100 lb, which results in a bolt loading between 1500 and 1600 lb will not cause fatigue failure at the operating conditions.

Table 16

PHYSICAL PROPERTIES OF MATERIALS USED TO MANUFACTURE ALLOY STEEL SHCS'S

Coefficient of Thermal Expansion, μm/m/°K[1]

20°C to 68°F to	100 212	200 392	300 572	400 752	500 932	600 1112
Material						
5137M, 51B37M[2]	–	12.6	13.4	13.9	14.3	14.6
4137[3]	11.2	11.8	12.4	13.0	13.6	–
4140[3]	12.3	12.7	–	13.7	–	14.5
4340[3]	–	12.4	–	13.6	–	14.5
8735[3]	11.7	12.2	12.8	13.5	–	14.1
8740[3]	11.6	12.2	12.8	13.5	–	14.1

Modulus of Elongation (Young's Modulus)

E = 30,000,000 PSI/in/in

Table 19 - Yield Strength at Various Temperatures

Alloy	Temperature (F) 70	800	1000	1200
Stainless Steels				
Type 302	35,000	35,000	34,000	30,000
Type 403	145,000	110,000	95,000	38,000
PH 15-7 Mo	220,000	149,000	101,000	–
High Strength Iron-Base Stainless Alloys				
A 286	95,000	95,000	90,000	85,000
AMS 5616	113,000	80,000	60,000	40,000
Unitemp 212	150,000	140,000	135,000	130,000
High Strength Iron-Base Alloys				
AISI 4340	200,000	130,000	75,000	–
H-11 (AMS 6485)	215,000	175,000	155,000	–
AMS 6340	160,000	100,000	75,000	–
Nickel-Base Alloys				
Iconel X	115,000	-	-	98,000
Waspaloy	115,000	-	106,000	100,000

NOTES:

1. Developed from ASM, Metals HDBK, 9th Edition, Vol. 1 (°C = °K for values listed)
2. ASME SA574
3. AISI
4. Multiply values in table by .556 for μin/in/°F.

CORROSION IN THREADED FASTENERS

All fastened joints are, to some extent, subjected to corrosion of some form during normal service life. Design of a joint to prevent premature failure due to corrosion must include considerations of the environment, conditions of loading , and the various methods of protecting the fastener and joint from corrosion.

Three ways to protect against corrosion are:

1. Select corrosion-resistant material for the fastener.
2. Specify protective coatings for fastener, joint interfaces, or both.
3. Design the joint to minimize corrosion.

The solution to a specific corrosion problem may require using one or all of these methods. Economics often necessitate a compromise solution.

Fastener Material

The use of a suitably corrosion-resistant material is often the first line of defense against corrosion. In fastener design, however, material choice may be only one of several important considerations. For example, the most corrosion-resistant material for a particular environment may just not make a suitable fastener.

Basic factors affecting the choice of corrosion resistant threaded fasteners are:

- Tensile and fatigue strength.
- Position on the galvanic series scale of the fastener and materials to be joined.
- Special design considerations: Need for minimum weight or the tendency for some materials to gall.
- Susceptibility of the fastener material to other types of less obvious corrosion. For example, a selected material may minimize direct attack of a corrosive environment only to be vulnerable to fretting or stress corrosion.

Some of the more widely used corrosion-resistant materials, along with approximate fastener tensile strength ratings at room temperature and other pertinent properties, are listed in Table 1. Sometimes the nature of corrosion properties provided by these fastener materials is subject to change with application and other conditions. For example, stainless steel and aluminum resist corrosion only so long as their protective oxide film remains unbroken. Alloy steel is almost never used, even under mildly corrosive conditions, without some sort of protective coating. Of course, the presence of a specific corrosive medium requires a specific corrosion-resistant fastener material, provided that design factors such as tensile and fatigue strength can be satisfied.

Protective Coating

A number of factors influence the choice of a corrosion-resistant coating for a threaded fastener. Frequently, the corrosion resistance of the coating is not a principal consideration. At times it is a case of economics. Often, less-costly fastener material will perform satisfactorily in a corrosive environment if given the proper protective coating.

Factors which affect coating choice are:

- Corrosion resistance
- Temperature limitations
- Embrittlement of base metal
- Effect on fatigue life
- Effect on locking torque
- Compatibility with adjacent material
- Dimensional changes
- Thickness and distribution
- Adhesion characteristics

Conversion Coatings: Where cost is a factor and corrosion is not severe, certain conversion-type coatings are effective. These include a black-oxide finish for alloy-steel screws and various phosphate base coatings for carbon and alloy-steel fasteners. Frequently, a rust-preventing oil is applied over a conversion coating.

Paint: Because of its thickness, paint is normally not considered for protective coatings for mating threaded fasteners. However, it is sometimes applied as a supplemental treatment at installation. In special cases, a fastener may be painted and installed wet, or the entire joint may be sealed with a coat of paint after installation.

TABLE 1 – TYPICAL PROPERTIES OF CORROSION RESISTANT FASTENER MATERIALS

Materials Stainless Steels	Tensile Strength (1000 psi)	Yield Strength at 0.2% offset (1000 psi)	Maximum Service Temp (F)	Mean Coefficient of Thermal Expan. (in./in./deg F)	Density (lbs/cu in.)	Base Cost Index	Position on Galvanic Scale
303, passive	80	40	800	10.2	0.286	Medium	8
303, passive, cold worked	125	80	800	10.3	0.286	Medium	9
410, passive	170	110	400	5.6	0.278	Low	15
431, passive	180	140	400	6.7	0.280	Medium	16
17-4 PH	200	180	600	6.3	0.282	Medium	11
17-7 PH	200	185	600	6.7	0.276	Medium	14
AM 350	200	162	800	7.2	0.282	Medium	13
15-7 Mo	200	155	600	–	0.277	Medium	12
A-286	150	85	1200	9.72	0.286	Medium	6
A-286, cold worked	220	170	1200	–	0.286	High	7

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Electroplating: Two broad classes of protective electroplating are: 1. The barrier type-such as chrome plating-which sets up an impervious layer or film that is more noble and therefore more corrosion resistant than the base metal. 2. The sacrificial type, zinc for example, where the metal of the coating is less noble than the base metal of the fastener. This kind of plating corrodes sacrificially and protects the fastener.

Noble-metal coatings are generally not suitable for threaded fasteners-especially where a close-tolerance fit is involved. To be effective, a noble-metal coating must be at least 0.001 in. thick. Because of screw-thread geometry, however, such plating thickness will usually exceed the tolerance allowances on many classes of fit for screws.

Because of dimensional necessity, threaded fastener coatings, since they operate on a different principle, are effective in layers as thin as 0.0001 to 0.0002 in.

The most widely used sacrificial platings for threaded fasteners are cadmium, zinc, and tin. Frequently, the cadmium and zinc are rendered even more corrosion resistant by a posting-plating chromate-type conversion treatment. Cadmium plating can be used at temperatures to 450°F. Above this limit, a nickel cadmium or nickel-zinc alloy plating is recommended. This consists of alternate deposits of the two metals which are heat-diffused into a uniform alloy coating that can be used for applications to 900°F. The alloy may also be deposited directly from the plating bath.

Fastener materials for use in the 900 to 1200°F range (stainless steel, A-286), and in the 1200° to 1800°F range (high-nickel-base super alloys) are highly corrosion resistant and normally do not require protective coatings, except under special environment conditions.

Silver plating is frequently used in the higher temperature ranges for lubrication to prevent galling and seizing, particularly on stainless steel. This plating can cause a galvanic corrosion problem, however, because of the high nobility of the silver.

Hydrogen Embrittlement: A serious problem, known as hydrogen embrittlement, can develop in plated alloy steel fasteners. Hydrogen generated during plating can diffuse into the steel and embrittle the bolt. The result is often a delayed and total mechanical failure, at tensile levels far below the theoretical strength, high-hardness structural parts are particularly susceptible to this condition. The problem can be controlled by careful selection of plating formulation, proper plating procedure, and sufficient baking to drive off any residual hydrogen.

Another form of hydrogen embrittlement, which is more difficult to control, may occur after installation. Since electrolytic cell action liberates hydrogen at the cathode, it is possible for either galvanic or concentration-cell corrosion to lead to embrittling of the bolt material.

Joint Design

Certain precautions and design procedures can be followed to prevent, or at least minimize, each of the various types of corrosion likely to attack a threaded joint. The most important of these are:

For Direct Attack: Choose the right corrosion-resistant material. Usually a material can be found that will provide the needed corrosion resistance without sacrifice of other important design requirements. Be sure that the fastener material is compatible with the materials being joined.

Corrosion resistance can be increased by using a conversion coating such as black oxide or a phosphate-base treatment. Alternatively, a sacrificial coating such as zinc plating is effective.

For an inexpensive protective coating, lacquer or paint can be used where conditions permit.

For Galvanic Corrosion: If the condition is severe, electrically insulate the bolt and joint from each other..

The fastener may be painted with zinc chromate primer prior to installation, or the entire joint can be coated with lacquer or paint.

Another protective measure is to use a bolt that is cathodic to the joint material and close to it in the galvanic series. When the joint material is anodic, corrosion will spread over the greater area of the fastened materials. Conversely, if the bolt is anodic, galvanic action is most severe.

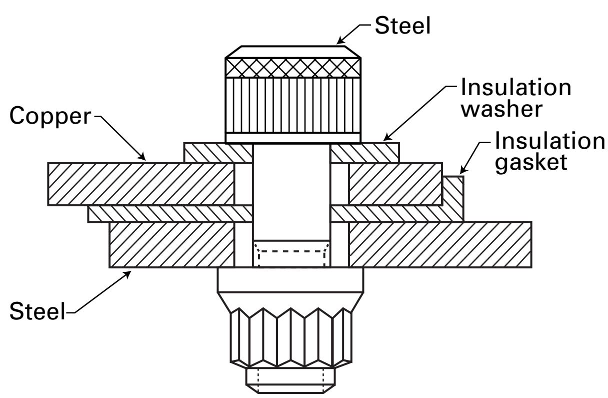


FIG. 1.1 – A method of electrically insulating a bolted joint to prevent galvanic corrosion.

For Concentration-Cell Corrosion: Keep surfaces smooth and minimize or eliminate lap joints, crevices, and seams. Surfaces should be clean and free of organic material and dirt. Air trapped under a speck of dirt on the surface of the metal may form an oxygen concentration cell and start pitting.

For maximum protection, bolts and nuts should have smooth surfaces, especially in the seating areas. Flush-head bolts should be used where possible. Further, joints can be sealed with paint or other sealant material.

For Fretting Corrosion: Apply a lubricant (usually oil) to mating surfaces. Where fretting corrosion is likely to occur: 1. Specify materials of maximum practicable hardness. 2. Use fasteners that have residual compressive stresses on the surfaces that may be under attack. 3. Specify maximum preload in the joint. A higher clamping force results in a more rigid joint with less relative movement possible between mating services.

For Stress Corrosion: Choose a fastener material that resists stress corrosion in the service environment. Reduce fastener hardness (if reduced strength can be tolerated), since this seems to be a factor in stress corrosion.

Minimize crevices and stress risers in the bolted joint and compensate for thermal stresses. Residual stresses resulting from sudden changes in temperature accelerate stress corrosion.

If possible, induce residual compressive stresses into the surface of the fastener by shot-peening or pressure rolling.

For Corrosion Fatigue: In general, design the joint for high fatigue life, since the principal effect of this form of corrosion is reduced fatigue performance. Factors extending fatigue performance are: 1. Application and maintenance of a high preload. 2. Proper alignment to avoid bending stresses.

If the environment is severe, periodic inspection is recommended so that partial failures may be detected before the structure is endangered.

As with stress and fretting corrosion, compressive stresses induced on the fastener surfaces by thread rolling, fillet rolling, or shot peening will reduce corrosion fatigue. Further protection is provided by surface coating.

TYPES OF CORROSION

Direct Attack…most common form of corrosion affecting all metals and structural forms. It is a direct and general chemical reaction of the metal with a corrosive medium-liquid, gas, or even a solid.

Galvanic Corrosion…occurs with dissimilar metals contact. Presence of an electrolyte, which may be nothing more than an individual atmosphere, causes corrosive action in the galvanic couple. The anodic, or less noble material, is the sacrificial element. Hence, in a joint of stainless steel and titanium, the stainless steel corrodes. One of the worst galvanic joints would consist of magnesium and titanium in contact.

Concentration Cell Corrosion…takes place with metals in close proximity and, unlike galvanic corrosion, does not require dissimilar metals. When two or more areas on the surface of a metal are exposed to different concentrations of the same solution, a difference in electrical potential results, and corrosion takes place.

If the solution consists of salts of the metal itself, a metal-ion cell is formed, and corrosion takes place on the surfaces in close contact. The corrosive solution between the two surfaces is relatively more stagnant (and thus has a higher concentration of metal ions in solution) than the corrosive solution immediately outside the crevice.

A variation of the concentration cell is the oxygen cell in which a corrosive medium, such as moist air, contains different amounts of dissolved oxygen at different points. Accelerated corrosion takes place between hidden surfaces (either under the bolt head or nut, or between bolted materials) and is likely to advance without detection.

Fretting…corrosive attack or deterioration occurring between containing, highly-loaded metal surfaces subjected to very slight (vibratory) motion. Although the mechanism is not completely understood, it is probably a highly accelerated form of oxidation under heat and stress. In threaded joints, fretting can occur between mating threads, at the bearing surfaces under the head of the screw, or under the nut. It is most likely to occur in high tensile, high-frequency, dynamic-load applications. There need be no special environment to induce this form of corrosion...merely the presence of air plus vibratory rubbing. It can even occur when only one of the materials in contact is metal.

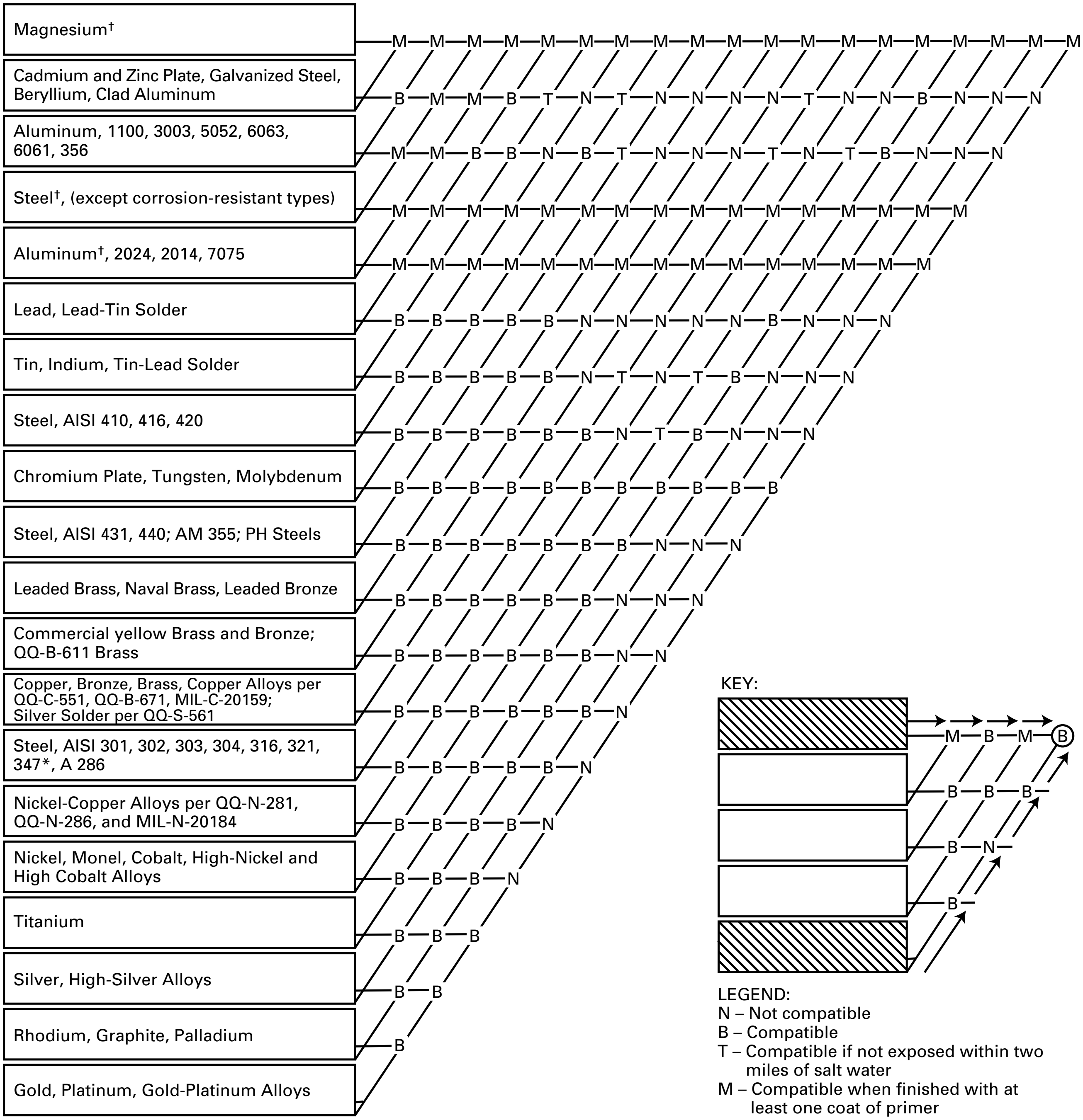
Stress Corrosion Cracking…occurs over a period of time in high-stressed, high-strength joints. Although not fully understood, stress corrosion cracking is believed to be caused by the combined and mutually accelerating effects of static tensile stress and corrosive environment. Initial pitting somehow tales place which, in turn, further increases stress build-up. The effect is cumulative and, in a highly stressed joint, can result in sudden failure.

Corrosion Fatigue…accelerated fatigue failure occurring in the presence of a corrosive medium. It differs from stress corrosion cracking in that dynamic alternating stress, rather than static tensile stress, is the contributing agent.

Corrosion fatigue affects the normal endurance limit of the bolt. The conventional fatigue curve of a normal bolt joint levels off at its endurance limit, or maximum dynamic load that can be sustained indefinitely without fatigue failure. Under conditions of corrosion fatigue, however, the curve does not level off but continues downward to a point of failure at a finite number of stress cycles.

II

GALVANIC CORROSION



LEGEND:
N – Not compatible
B – Compatible
T – Compatible if not exposed within two miles of salt water
M – Compatible when finished with at least one coat of primer

FIG. 19 – Metals compatibility chart

THE IMPACT PERFORMANCE OF THREADED FASTENERS

Much has been written regarding the significance of the notched bar impact testing of steels and other metallic materials. The Charpy and Izod type test relate notch behavior (brittleness versus ductility) by applying a single overload of stress. The results of these tests provide quantitive comparisons but are not convertible to energy values useful for engineering design calculations. The results of an individual test are related to that particular specimen size, notch geometry and testing conditions and cannot be generalized to other sizes of specimens and conditions.

The results of these tests are useful in determining the susceptibility of a material to brittle behavior when the applied stress is perpendicular to the major stress.

In externally threaded fasteners, however, the loading usually is applied in a longitudinal direction. The impact test, therefore, which should be applicable would be one where the applied impact stress supplements the major stress. Only in shear loading on fasteners is the major stress in the transverse direction.

Considerable testing has been conducted in an effort to determine if a relationship exists between the Charpy V notch properties of a material and the tension properties of an externally threaded fastener manufactured from the same material.

Some conclusions which can be drawn from the extensive impact testing are as follows:

1. The tension impact properties of externally threaded fasteners do not follow the Charpy V notch impact pattern.
2. Some of the variables which effect the tension impact properties are:
   A. The number of exposed threads
   B. The length of the fastener
   C. The relationship of the fastener shank diameter to the thread area.
   D. The hardness or fastener ultimate tensile strength

Following are charts showing tension impact versus Charpy impact properties, the effect of strength and diameter on tension impact properties and the effect of test temperature. II

Please note from figure 21 that while the Charpy impact strength of socket head cap screw materials are decreasing at sub-zero temperatures, the tension impact strength of the same screws is increasing. This compares favorable with the effect of cryogenic temperatures on the tensile strength of the screws. Note the similar increase in tensile strength shown in figure 22.

It is recommended, therefore, that less importance be attached to Charpy impact properties of materials which are intended to be given to impact properties for threaded fasteners. If any consideration is to be given to impact properties of bolts or screws, it is advisable to investigate the tension impact properties of full size fasteners since this more closely approximates the actual application.

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TABLE 20
LOW-TEMPERATURE IMPACT PROPERTIES OF SELECTED ALLOY STEELS

AISI no.	composition, %					heat temperature*		Hardness Rc	impact energy, ft.-lb					transition temp. (50% brittle) °F
	C	Mn	Ni	Cr	Mo	quenching temp. F+	tempering temp. F		–300°F	–200°F	–100°F	0°F	100°F	
4340	0.38	0.77	1.65	0.93	0.21	1550	400	52	11	15	20	21	21	–
							600	48	10	14	15	15	16	–
							800	44	9	13	16	21	25	–
							1000	38	15	18	28	36	36	–130
							1200	30	15	28	55	55	55	–185
4360	0.57	0.87	1.62	1.08	0.22	1475	800	48	5	6	10	11	14	–
							1000	40	9	10	13	18	23	–10
							1200	30	12	15	25	42	43	–110
4380	0.76	0.91	1.67	1.11	0.21	1450	800	49	4	5	8	9	10	–
							1000	42	8	8	10	12	15	60
							1200	31	5	11	19	33	38	–50
4620	0.20	0.67	1.85	0.30	0.18	1650	300	42	14	20	28	35	35	–
							800	34	11	16	33	55	55	–
							1000	29	16	34	55	78	78	–
							1200	19	17	48	103	115	117	–
4640	0.43	0.69	1.78	0.29	0.20	1550	800	42	16	17	20	25	27	–
							1000	37	17	22	35	39	69	–190
							1200	29	17	30	55	97	67	–180
4680	0.74	0.77	1.81	0.30	0.21	1450	800	46	5	8	13	15	16	–
							1000	41	11	12	15	19	22	–
							1200	31	11	13	17	39	43	–
8620	0.20	0.89	0.60	0.68	0.20	1650	300	43	11	16	23	35	35	–
							800	36	8	13	20	35	45	–20
							1000	29	25	33	65	76	76	–150
							1200	21	10	85	107	115	117	–195
8630	0.34	0.77	0.66	0.62	0.22	1575	800	41	7	12	17	25	31	0
							1000	34	11	20	43	53	54	–155
							1200	27	18	28	74	80	82	–165
8640	0.45	0.78	0.65	0.61	0.20	1550	800	46	5	10	14	20	23	–
							1000	38	11	15	24	40	40	–110
							1200	30	18	22	49	63	66	–140
8660	0.56	0.81	0.70	0.56	0.25	1475	800	47	4	6	10	13	16	–
							1000	41	10	12	15	20	30	–10
							1200	30	16	18	25	54	60	–90

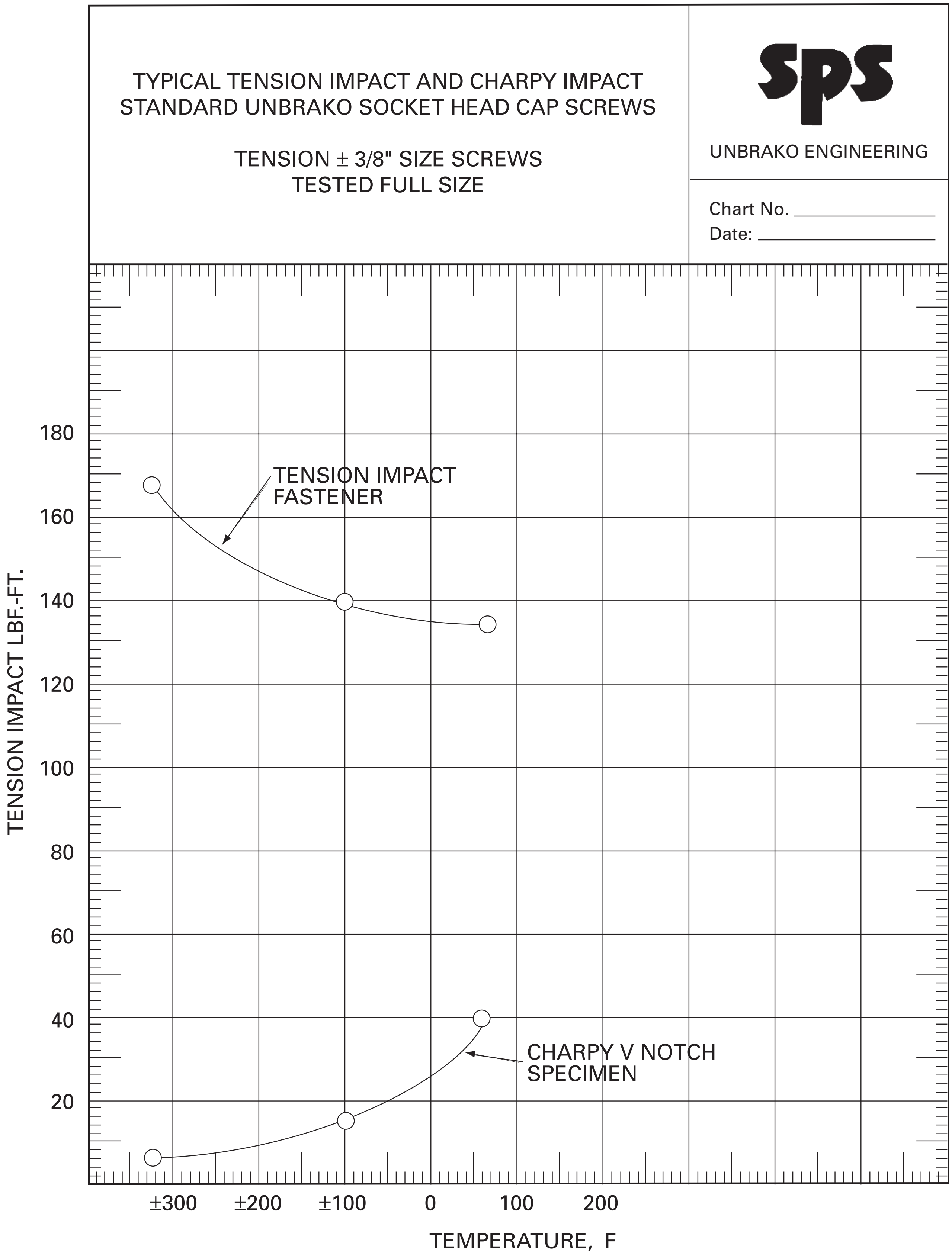
TYPICAL TENSION IMPACT AND CHARPY IMPACT
STANDARD UNBRAKO SOCKET HEAD CAP SCREWS
TENSION ± 3/8" SIZE SCREWS
TESTED FULL SIZE
UNBRAKO ENGINEERING
Chart No.
Date:
TENSION IMPACT
FASTENER
CHARPY V NOTCH
SPECIMEN
TENSION IMPACT LBF.-FT.
180
160
140
120
100
80
60
40
20
±300
±200
±100
0
100
200
TEMPERATURE, F

FIG. 21

II

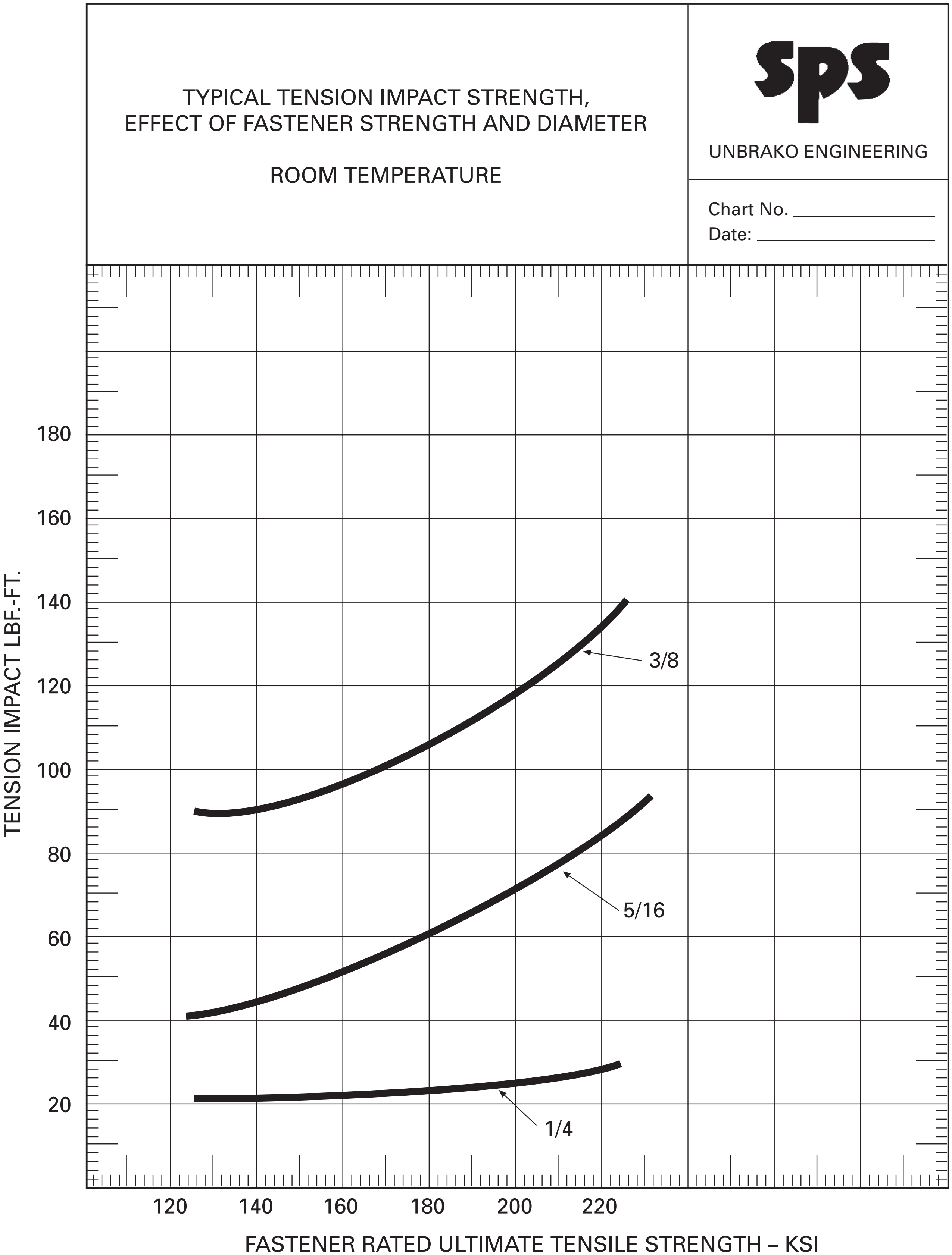
TYPICAL TENSION IMPACT STRENGTH,
EFFECT OF FASTENER STRENGTH AND DIAMETER
ROOM TEMPERATURE
SPS
UNBRAKO ENGINEERING
Chart No.
Date:
TENSION IMPACT LBF.-FT.
180
160
140
120
100
80
60
40
20
3/8
5/16
1/4
120
140
160
180
200
220
FASTENER RATED ULTIMATE TENSILE STRENGTH – KSI

FIG. 22

UNBRAKO PRODUCT ENGINEERING BULLETIN

Standard Inch Socket Head Cap Screws Are Not Grade 8 Fasteners

There is a common, yet reasonable, misconception that standard, inch, alloy steel socket head cap screws are "Grade 8". This is not true. The misconception is reasonable because "Grade 8" is a term generally associated with "high strength" fasteners. A person desiring a "high strength" SHCS may request a "Grade 8 SHCS". This is technically incorrect for standard SHCSs. The term Grade 8 defines specific fastener characteristics which must be met to be called "Grade 8". Three of the most important characteristics are not consistent with requirements for industry standard SHCSs: tensile strength, hardness, and head marking. Some basic differences between several fastener classifications are listed below. The list is not comprehensive but intended to provide a general understanding. SHCSs can be manufactured to meet Grade 8 requirements on a special order basis.

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Fastener Designation	Grade 2	Grade 5	Grade 8	Industry SHCS	Unbrako SHCS
Applicable Standard	SAE J429	SAE J429	SAE J429	ASTM A574	ASTM A574 SPS-B-271
Strength Level, UTS KSI, min.	74 (1/4-3/4) 60 (7/8 - 1 1/2)	120 (1/4 - 1) 105 (1 1/8 - 1 1/2)	150 (1/4 - 11/2)	180 (≤1/2) 170 (> 1/2)	190 (≤ 1/2) 180 (> 1/2)
Hardness, Rockwell	B80-B100 B70-B100	C25-C34 C19-C30	C33-C39	C39-C45 C37-C45	C39-C43 C38-C43
General Material Type	Low or Medium Carbon Steel	Medium Carbon Steel	Medium Carbon Alloy Steel	Medium Carbon Alloy Steel	Medium Carbon Alloy Steel
Identification Requirement	None	Three Radial Lines	Six Radial Lines	SHCS Configuration	Mfr's ID
Typical Fasteners	Bolts Screws Studs Hex Heads	Bolts Screws Studs Hex Heads	Bolts Screws Studs Hex Heads	Socket Head Cap Screws	Socket Head Cap Screws

THREADS IN BOTH SYSTEMS

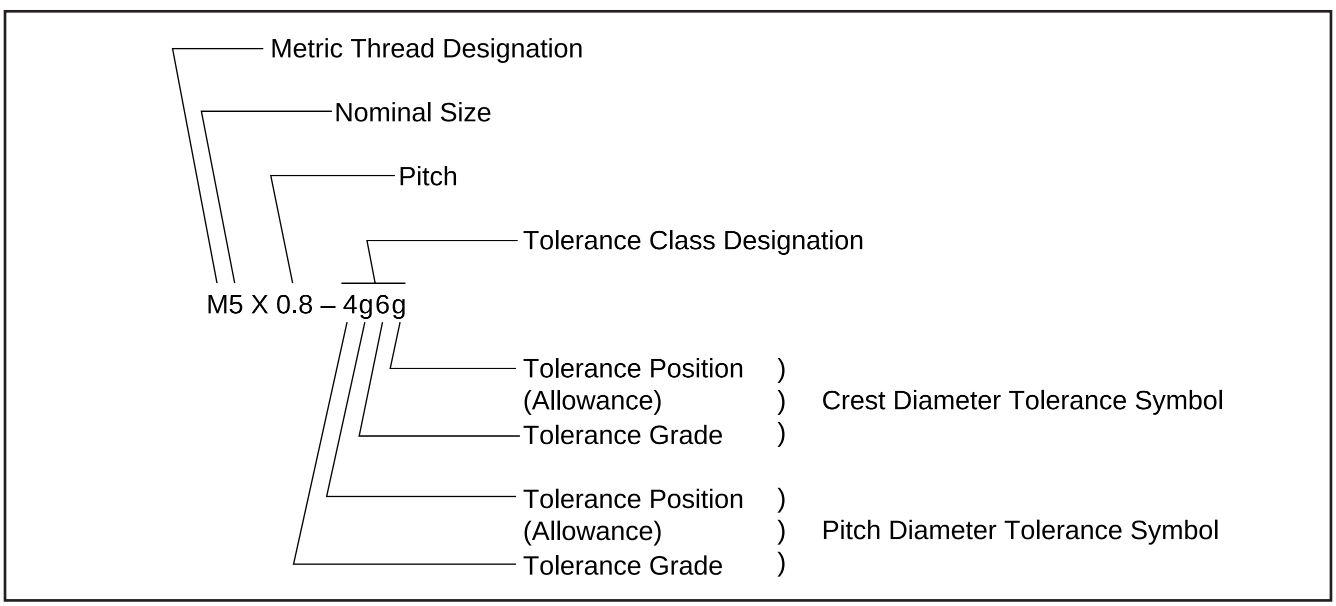
Thread forms and designations have been the subject of many long and arduous battles through the years. Standardization in the inch series has come through many channels, but the present unified thread form could be considered to be the standard for many threaded products, particularly high strength ones such as socket head cap screws, etc. In common usage in U.S.A., Canada and United Kingdom are the Unified National Radius Coarse series, designated UNRC, Unified National Radius Fine series, designated UNRF, and several special series of various types, designated UNS. This thread, UNRC or UNRF, is designated by specifying the diameter and threads per inch along with the suffix indicating the thread series, such as 1/4 - 28 UNRF. For threads in Metric units, a similar approach is used, but with some slight variations. A diameter and pitch are used to designate the series, as in the Inch system, with modifications as follows: For coarse threads, only the prefix M and the diameter are necessary, but for fine threads, the pitch is shown as a suffix. For example, M16 is a coarse thread designation representing a diameter of 16 mm with a pitch of 2 mm understood. A similar fine thread part would be M16 x 1.5 or 16 mm diameter with a pitch of 1.5 mm.

For someone who has been using the Inch system, there are a couple of differences that can be a little confusing. In the Inch series, while we refer to threads per inch as pitch; actually the number of threads is 1/pitch. Fine threads are referenced by a larger number than coarse threads because they “fit” more threads per inch.

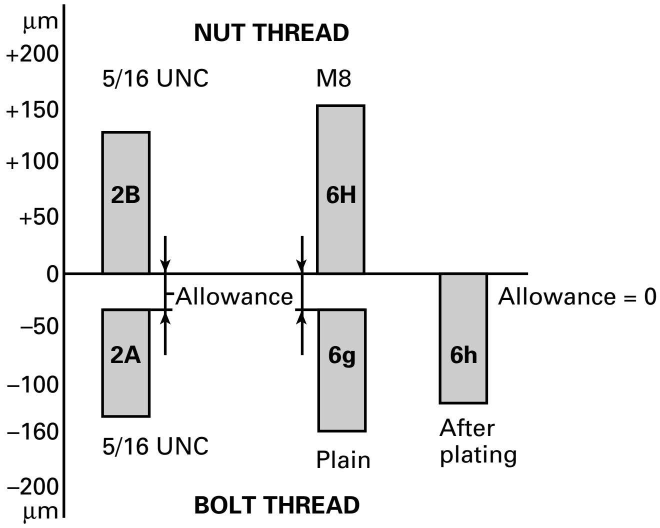
In Metric series, the diameters are in millimeters, but the pitch is really the pitch. Consequently the coarse thread has the large number. The most common metric thread is the coarse thread and falls generally between the inch coarse and fine series for a comparable diameter.

Also to be considered in defining threads is the tolerance and class of fit to which they are made. The International Standards Organization (ISO) metric system provides for this designation by adding letters and numbers in a certain sequence to the callout. For instance, a thread designated as M5 x 0.8 4g6g would define a thread of 5 mm diameter, 0.8 mm pitch, with a pitch diameter tolerance grade 6 and allowance “g”. These tolerances and fields are defined as shown below, similar to the Federal Standard H28 handbook, which defines all of the dimensions and tolerances for a thread in the inch series. The callout above is similar to a designation class 3A fit, and has a like connotation.

COMPLETE DESIGNATIONS



Example of thread tolerance positions and magnitudes.
Comparision 5/16 UNC and M8. Medium tolerance grades – Pitch diameter.



DEVIATIONS

external	internal	basic clearance
h g e	H G	none small large

II

NOTE:
Lower case letters = external threads
Capital letters = internal threads

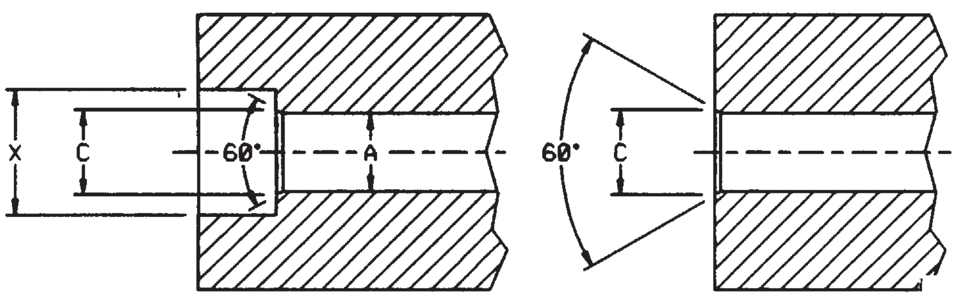
THROUGH-HOLE PREPARATION

II

Close Fit: Normally limited to holes for those lengths of screws threaded to the head in assemblies in which: (1) only one screw is used; or (2) two or more screws are used and the mating holes are produced at assembly or by matched and coordinated tooling.

Normal Fit: Intended for: (1) screws of relatively long length; or (2) assemblies that involve two or more screws and where the mating holes are produced by conventional tolerancing methods. It provides for the maximum allowable eccentricty of the longest standard screws and for certain deviations in the parts being fastened, such as deviations in hole straightness; angularity between the axis of the tapped hole and that of the hole for the shank; differneces in center distances of the mating holes and other deviations.

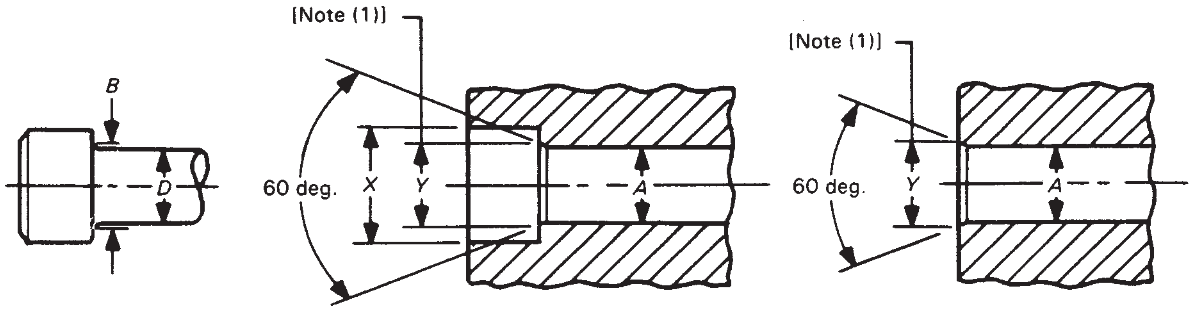
Chamfering: It is considered good practice to chamfer or break the edges of holes that are smaller than "F" maximum in parts in which hardness approaches, equals or exceeds the screw hardness. If holes are not chamfered, the heads may not seat properly or the sharp edges may deform the fillets on the screws, making them susceptible to fatigue in applications that involve dynamic loading. The chamfers, however, should not be larger than needed to ensure that the heads seat properly or that the fillet on the screw is not deformed. Normally, the chamfers do not need to exceed "F" maximum. Chamfers exceeding these values reduce the effective bearing area and introduce the possibility of indentation when the parts fastened are softer than screws, or the possiblity of brinnelling of the heads of the screws when the parts are harder than the screws. (See "F" page 6).



nominal size	basic screw diameter	A: drill size for hole A, close fit, nom.	close fit, dec.	normal fit, nom.	normal fit, dec.	X: counter-bore diameter	C: countersink diameter D Max. + 2F(Max.)	hole dimensions: tap drill size, UNRC	tap drill size, UNRF	**body drill size	counter-bore size
0	0.0600	51*	0.0670	49*	0.0730	1/8	0.074	–	3/64	#51	1/8
1	0.0730	46*	0.0810	43*	0.0890	5/32	0.087	1.5mm	#53	#46	5/32
2	0.0860	3/32	0.0937	36*	0.1065	3/16	0.102	#50	#50	3/32	3/16
3	0.0990	36*	0.1065	31*	0.1200	7/32	0.115	#47	#45	#36	7/32
4	0.1120	1/8	0.1250	29*	0.1360	7/32	0.130	#43	#42	1/8	7/32
5	0.1250	9/64	0.1406	23*	0.1540	1/4	0.145	#38	#38	9/64	1/4
6	0.1380	23*	0.1540	18*	0.1695	9/32	0.158	#36	#33	#23	9/32
8	0.1640	15*	0.1800	10	0.1935	5/16	0.188	#29	#29	#15	5/16
10	0.1900	5*	0.2055	2*	0.2210	3/8	0.218	#25	#21	#5	3/8
1/4	0.2500	17/64	0.2656	9/23	0.2812	7/16	0.278	#7	#3	17/64	7/16
5/16	0.3125	21/64	0.3281	11/32	0.3437	17/32	0.346	F	I	21/64	17/32
3/8	0.0375	25/64	0.3906	13/32	0.4062	5/8	0.415	5/16	Q	25/64	5/8
7/16	0.4375	29/64	0.4531	15/32	0.4687	23/32	0.483	U	25/64	29/64	23/32
1/2	0.5000	33/64	0.5156	17/32	0.5312	13/16	0.552	27/64	29/64	33/64	13/16
5/8	0.6250	41/64	0.6406	21/32	0.6562	1	0.689	35/64	14.5mm	41/64	1
3/4	0.7500	49/64	0.7656	25/32	0.7812	1-3/16	0.828	21/32	11/16	49/64	1-3/16
7/8	0.8750	57/64	0.8906	29/32	0.9062	1-3/8	0.963	49/64	20.5mm	57/64	1-3/8
1	1.0000	1-1/64	1.0156	1-1/32	1.0312	1-5/8	1.100	7/8	59/64	1-1/64	1-5/8
1-1/4	1.2500	1-9/32	1.2812	1-5/32	1.3125	2	1.370	1-7/64	1-11/64	1-9/32	2
1-1/2	1.5000	1-17/32	1.5312	1-9/16	1.5625	2-3/8	1.640	34mm	36mm	1-17/32	2-3/8

** Break edge of body drill hole to clear screw fillet.

DRILL AND COUNTERBORE SIZES FOR METRIC SOCKET HEAD CAP SCREWS



Nominal Size or Basic Screw Diameter	A		X	Y
	Nominal Drill Size		Counterbore Diameter	Countersink Diameter [Note (1)]
	Close Fit [Note (2)]	Normal Fit [Note (3)]		
M1.6	1.80	1.95	3.50	2.0
M2	2.20	2.40	4.40	2.6
M2.5	2.70	3.00	5.40	3.1
M3	3.40	3.70	6.50	3.6
M4	4.40	4.80	8.25	4.7
M5	5.40	5.80	9.75	5.7
M6	6.40	6.80	11.25	6.8
M8	8.40	8.80	14.25	9.2
M10	10.50	10.80	17.25	11.2
M12	12.50	12.80	19.25	14.2
M14	14.50	14.75	22.25	16.2
M16	16.50	16.75	25.50	18.2
M20	20.50	20.75	31.50	22.4
M24	24.50	24.75	37.50	26.4
M30	30.75	31.75	47.50	33.4
M36	37.00	37.50	56.50	39.4
M42	43.00	44.0	66.00	45.6
M48	49.00	50.00	75.00	52.6

HARDNESS – TENSILE CONVERSION

II

INCH
ROCKWELL – BRINELL – TENSILE CONVERSION

Rockwell "C" scale	Brinell hardness number	tensile strength approx. 1000 psi
60	654	336
59	634	328
58	615	319
57	595	310
56	577	301
55	560	292
54	543	283
53	524	274
52	512	265
51	500	257
50	488	249
49	476	241
48	464	233
47	453	225
46	442	219
45	430	212
44	419	206

Rockwell "C" scale	Brinell hardness number	tensile strength approx. 1000 psi
43	408	200
42	398	194
41	387	188
40	377	181
39	367	176
38	357	170
37	347	165
36	337	160
35	327	155
34	318	150
33	309	147
32	301	142
31	294	139
30	285	136
29	279	132
28	272	129
27	265	126

Rockwell		Brinell hardness number	tensile strength approx. 1000 psi
"C" scale	"B" scale		
26		259	123
25		253	120
24		247	118
23		241	115
22	100	235	112
21	99	230	110
20	98	225	107
(19)		220	104
(18)	97	215	103
(17)		210	102
(16)	96	206	100
(15)		201	99
(14)	95	197	97
(13)	94	193	96
(12)	93	190	93
(11)		186	91
(10)	92	183	90

METRIC
ROCKWELL – BRINELL – TENSILE CONVERSION

Rockwell "C" scale	Brinell hardness number	tensile strength approx. MPa
60	654	2,317
59	634	2,261
58	615	2,199
57	595	2,137
56	577	2,075
55	560	2,013
54	543	1,951
53	524	1,889
52	512	1,827
51	500	1,772
50	488	1,717
49	476	1,662
48	464	1,606
47	453	1,551
46	442	1,510
45	430	1,462
44	419	1,420

Rockwell "C" scale	Brinell hardness number	tensile strength approx. MPa
43	408	1,379
42	398	1,338
41	387	1,296
40	377	1,248
39	367	1,213
38	357	1,172
37	347	1,138
36	337	1,103
35	327	1,069
34	318	1,034
33	309	1,014
32	301	979
31	294	958
30	285	938
29	279	910
28	272	889
27	265	869

Rockwell		Brinell hardness number	tensile strength approx. MPa
"C" scale	"B" scale		
26		259	848
25		253	827
24		247	814
23		241	793
22	100	235	772
21	99	230	758
20	98	225	738
(19)		220	717
(18)	97	215	710
(17)		210	703
(16)	96	206	690
(15)		201	683
(14)	95	197	669
(13)	94	193	662
(12)	93	190	641
(11)		186	627
(10)	92	183	621

THREAD STRESS AREAS

Inch and Metric

II

STRESS AREAS FOR THREADED FASTENERS – INCH

Diameter (in.)		Diameter (mm)	Threads Per in.		Square Inches		
					Tensile Stress Area Per H-28		
			UNRC	UNRF	UNRC	UNRF	Nominal Shank
#0	0.06	1.52	–	80	–	0.00180	0.002827
#1	0.07	1.85	64	72	0.00263	0.00278	0.004185
#2	0.09	2.18	56	64	0.00370	0.00394	0.005809
#3	0.10	2.51	48	56	0.00487	0.00523	0.007698
#4	0.11	2.84	40	48	0.00604	0.00661	0.009852
#5	0.13	3.18	40	44	0.00796	0.00830	0.012272
#6	0.14	3.51	32	40	0.00909	0.01015	0.014957
#8	0.16	4.17	32	36	0.0140	0.01474	0.021124
#10	0.19	4.83	24	32	0.0175	0.0200	0.028353
1/4	0.25	6.35	20	28	0.0318	0.0364	0.049087
5/16	0.31	7.94	18	24	0.0524	0.0580	0.076699
3/8	0.38	9.53	16	24	0.0775	0.0878	0.11045
7/16	0.44	11.11	14	20	0.1063	0.1187	0.15033
1/2	0.50	12.70	13	20	0.1419	0.1599	0.19635
9/16	0.56	14.29	12	18	0.182	0.203	0.25
5/8	0.63	15.88	11	18	0.226	0.256	0.31
3/4	0.75	19.05	10	16	0.334	0.373	0.44179
7/8	0.88	22.23	9	14	0.462	0.509	0.60132
1	1.00	25.40	8	12	0.606	0.663	0.79
1-1/8	1.13	28.58	7	12	0.763	0.856	0.99402
1-1/4	1.25	31.75	7	12	0.969	1.073	1.2272
1-3/8	1.38	34.93	6	12	1.155	1.315	1.4849
1-1/2	1.50	38.10	6	12	1.405	1.581	1.7671
1-3/4	1.75	44.45	5	12	1.90	2.19	2.4053
2	2.00	50.80	4-1/2	12	2.50	2.89	3.1416
2-1/4	2.25	57.15	4-1/2	12	3.25	3.69	3.9761
2-1/2	2.50	63.50	4	12	4.00	4.60	4.9088
2-3/4	2.75	69.85	4	12	4.93	5.59	5.9396
3	3.00	76.20	4	12	5.97	6.69	7.0686

STRESS AREAS FOR THREADED FASTENERS – METRIC

Nominal Dia. Thread and Pitch (mm)	Thread Tensile Stress Area (mm^2)	Nominal Shank Area (mm^2)
1.6 x 0.35	1.27	2.01
2.0 x 0.4	2.07	3.14
2.5 x 0.45	3.39	4.91
3.0 x 0.5	5.03	7.07
4.0 x 0.7	8.78	12.6
5.0 x 0.8	14.2	19.6
6.0 x 1	20.1	28.3
8.0 x 1.25	36.6	50.3
10 x 1.5	58.00	78.5
12 x 1.75	84.3	113
14 x 2	115	154
16 x 2	157	201
18 x 2.5	192	254
20 x 2.5	245	314
22 x 2.5	303	380
24 x 3	353	452
27 x 3	459	573
30 x 3.5	561	707
33 x 3.5	694	855
36 x 4	817	1018
42 x 4.5	1120	1385
48 x 5	1470	1810

II

A General information

A.1 Description

O-Rings offer the designer an efficient and economical sealing element for a wide range of static or dynamic applications.

Inexpensive production methods and its ease of use have made the O-Ring the most widely used seal.

A wide choice of elastomer materials for both standard and special applications allow the O-Ring to be used to seal practically all liquid and gaseous media.

O-Rings are vulcanised in moulds and are characterised by their circular form with annular cross section. The dimensions of the O-Ring are defined by the inside diameter d_1 and the cross section d_2 (Figure 1).

Cross sections of approx. 0.35 to 40 mm and inside diameters up to 5,000 mm and more are available.

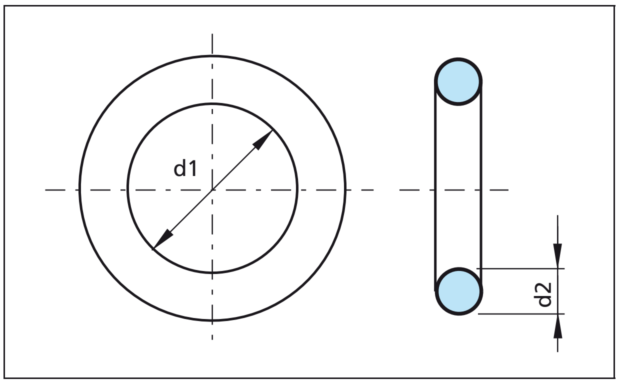


Figure 1 O-Ring dimensioning

Advantages

Compared with other sealing elements, the O-Ring has a wide range of advantages:

- Simple, one piece groove design reduces hardware and design costs
- Compact design allows smaller hardware
- Easy, foolproof installation reduces risk
- Applicable to a wide range of sealing problems, static, dynamic, single or double acting
- Wide compound choice for compatibility with most fluids
- Ex stock availability of many sizes worldwide for easy maintenance and repair.

A.2 Applications

O-Rings are used as sealing elements or as energising elements for hydraulic slipper seals and wipers and thus cover a large number of fields of application. There are no fields of industry where the O-Ring is not used. From an individual seal for repairs or maintenance to a quality assured application in aerospace, automotive or general engineering. The O-Ring is used predominantly for static sealing applications:

- As a radial static seal, e.g. for bushings, covers, pipes, cylinders
- As an axial static seal, e.g. for flanges, plates, caps.

O-Rings in dynamic applications are recommended **only for moderate service conditions**. They are limited by the speed and the pressure against which they are to seal:

- For low duty sealing of reciprocating pistons, rods, plungers, etc.
- For sealing of slowly pivoting, rotating or spiral movements on shafts, spindles, rotary transmission leadthroughs, etc.

A.3 Method of operation

O-Rings are double-acting sealing elements. The inital squeeze, which acts in a radial or axial direction depending on the installation, gives the O-Ring its initial sealing capability. These forces are superimposed by the system pressure to create the total sealing force which increases as the systeme pressure increases (Figure 2).

Under pressure, the O-Ring behaves in a similar way to a fluid with high surface tension. The pressure is transmitted uniformly to all directions.

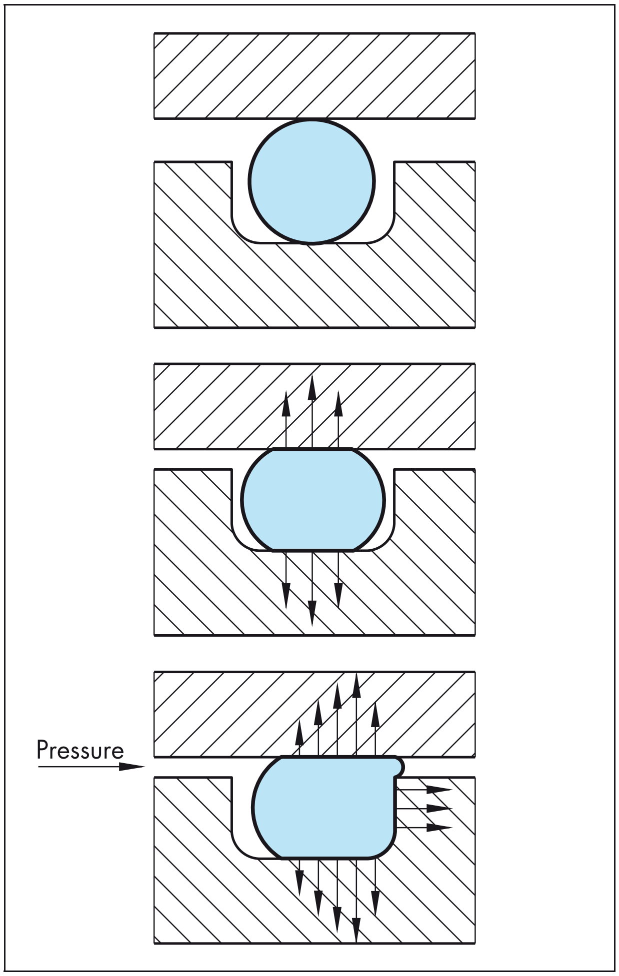


Figure 2 O-Ring sealing forces with and without system pressure

B Technical information

B.1 Materials

B.1.1 Elastomers

Equipment manufacturers and end users expect sealing systems to operate leak free and to maintain long service life. Reliability is crucial to effective low maintenance cost operations. To find the perfect sealing solution in each individual case both material performance and seal design are critically important. One of the main used material groups for sealings are the elastomers. They show good properties like elasticity or good chemical compatibility.

II

The following tables provide a summary of the various elastomer material groups. Trelleborg Sealing Solutions can offer a large number of materials within each group.

If no particular specifications are given for the material, standard NBR (Nitrile Rubber) in 70 Shore A will be supplied (see chapter "B.1.5 Standard materials").

Table I Elastomers

Designation	Trade Name*	Abbreviation		
		ISO 1629	ASTM 1418	TSS
Acrylonitrile-Butadiene Rubber (Nitrile Rubber)	Europrene® Krynac® Nipol N® Perbunan NT Breon®	NBR	NBR	N
Hydrogenated Acrylonitrile-Butadiene Rubber	Therban® Zetpol®	HNBR	HNBR	H
Polyacrylate Rubber	Noxtite® Hytemp® Nipol AR®	ACM	ACM	A
Chloroprene Rubber	Baypren® Neoprene®	CR	CR	WC
Ethylene Propylene Diene Rubber	Dutral® Keltan® Vistalon® Buna EP®	EPDM	EPDM	E
Silicone Rubber	Elastoseal® Rhodorsil® Silastic® Silopren®	VMQ	VMQ	S
Fluorosilicone Rubber	Silastic®	FVMQ	FVMQ	F
Tetrafluorethylene-Propylene Copolymer Elastomer	Aflas®	FEPM	TFE / P**	WT
Butyl Rubber	Esso Butyl®	IIR	IIR	WI
Styrene-Butadiene Rubber	Buna S® Europrene® Polysar S®	SBR	SBR	WB
Natural Rubber		NR	WR	WR
Fluorocarbon Rubber	Dai-El® Fluorel® Tecnoflon® Viton®	FKM	FKM	V
Perfluoro Rubber	Isolast® Kalrez®	FFKM	FFKM	J
Polyester Urethane Polyether Urethane	Zurcon® Adiprene® Pellethan® Vulcollan® Desmopan®	AU EU	AU EU	WU WU

* Selection of registered trade names
** Abbreviation not yet standardised.

ASTM = American Society for Testing and Materials
ISO = International Organisation for Standardisation

Designation	Trade Name*	Abbreviation		
		ISO 1629	ASTM 1418	TSS
Chlorosulphonated Polyethylene Rubber	Hypalon®	CSM	CSM	WM
Polysulphide Elastomer	Thiokol®	-	TWT	WY
Epichlorohydrin Elastomer	Hydrin®	-	-	WO

* Selection of registered trade names
** Abbreviation not yet standardised.

ASTM = American Society for Testing and Materials
ISO = International Organisation for Standardisation

Table II The most important types of synthetic rubber, their grouping and abbreviations

Chemical name	Abbreviation	
	DIN / ISO 1629	ASTM D - 1418
M - Group (saturated carbon molecules in main macro-molecule-chain) - Polyacrylate Rubber - Ethylene Acrylate Rubber - Chlorosulphonated Polyethylene Rubber - Ethylene Propylene Diene Rubber - Ethylene Propylene Rubber - Fluorocarbon Rubber - Perfluoro Rubber	 AC**M** AE**M** CS**M** EPD**M** EP**M** FK**M** FFK**M**	 AC**M** CS**M** EPD**M** EP**M** FK**M** FFK**M**
O - Group (with oxygen molecules in the main macro-molecule chain) - Epichlorohydrin Rubber - Epichlorohydrin Copolymer Rubber	 C**O** EC**O**	 C**O** EC**O**
R - Group (unsaturated hydrogene carbon chain) - Chloroprene Rubber - Butyl Rubber - Nitrile Butadiene Rubber - Natural Rubber - Styrene Butadiene Rubber - Hydrogenated Nitrile Butadiene Rubber	 C**R** II**R** NB**R** N**R** SB**R** HNB**R**	 C**R** II**R** NB**R** N**R** SB**R** HNB**R**
Q - Group (with Silicone in the main chain) - Fluorosilicone Rubber - Methyl Vinyl Silicone Rubber	 FVM**Q** VM**Q**	 FVM**Q** VM**Q**
U - Group (with carbon, oxygen and nitrogen in the main chain) - Polyester Urethane - Polyether Urethane	 A**U** E**U**	 A**U** E**U**

II

B.1.2 Application parameters of elastomers

Elastomers as all other organic chemicals have limited use. External influences such as various media, oxygen or ozone as well as pressure and temperature will affect the material properties and therefore their sealing capability.

Elastomers will amongst others swell, shrink or harden and develop cracks or even tears.
The following information illustrates the different application parameters.

Elastomer heat resistance / swelling in oil

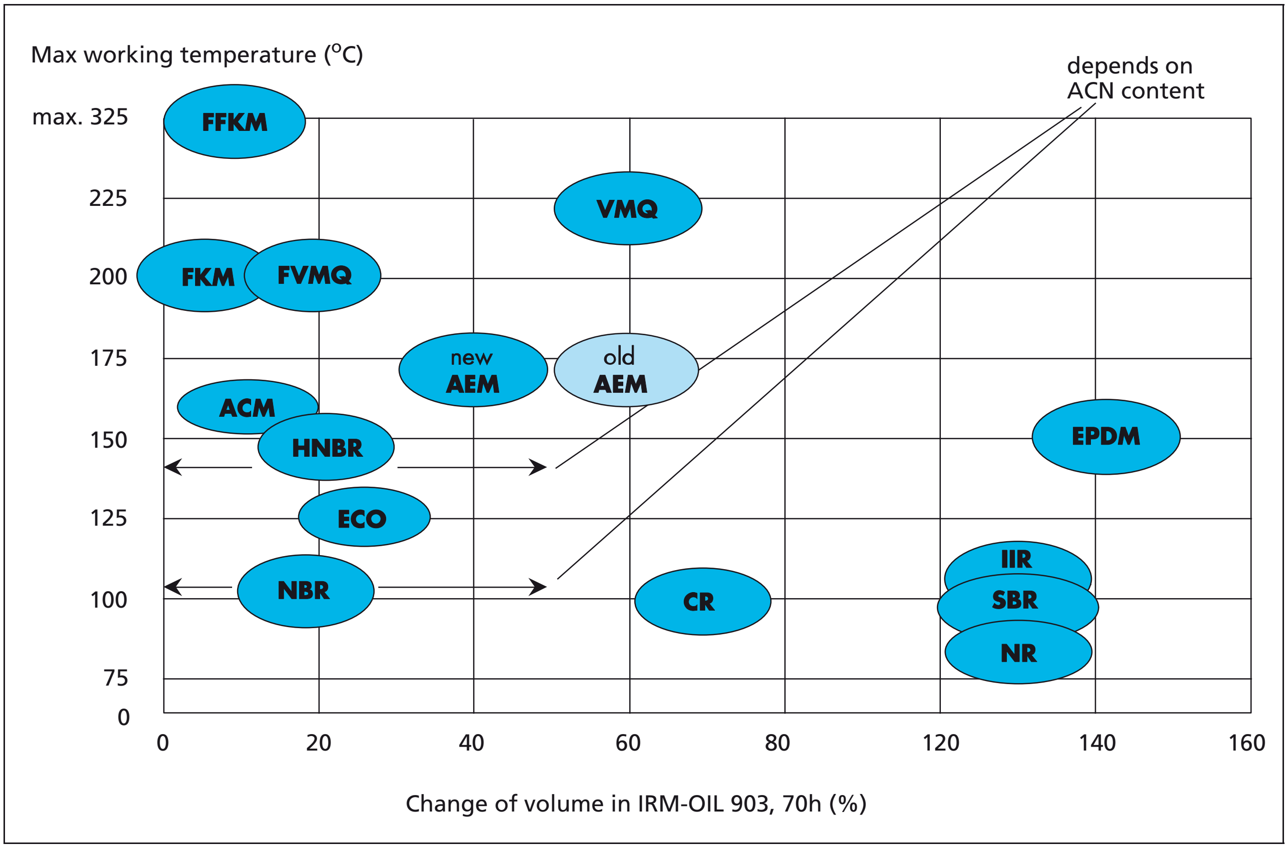


Figure 3 Change of volume in IRM-Oil 903 (old ASTM-Oil No 3)

Temperature range

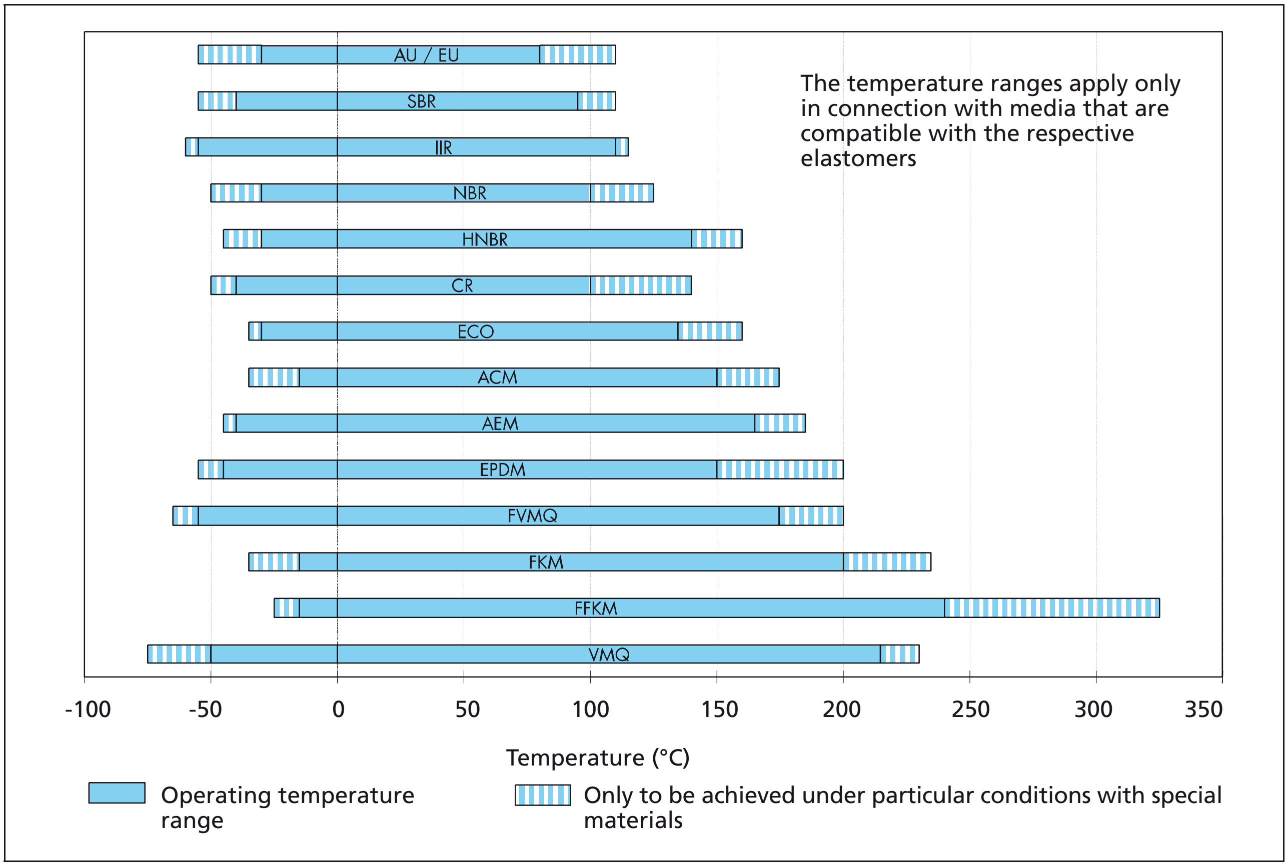


Figure 4 Temperature range of various elastomers

General field of application

Elastomer materials are used to cover a large number of fields of application. Details regarding resistance in special media are shown in chapter "Chemical compatibility", page 9.

The various elastomers can be characterised as follows:

NBR (Nitrile Butadiene Rubber):

The properties of the Nitrile Rubber depend mainly on the ACN content which ranges between 18% and 50%. In general they show good mechanical properties. The operating temperatures range between -30°C and +100°C (for a short period of time up to +120°C). Suitable formulated NBR can be used down to -60°C.
NBR is mostly used with mineral based oils and greases.

FKM (Fluorocarbon Rubber)

Depending on structure and fluorine content FKM materials can differ with regards to their chemical resistance and cold-flexibility.

FKM is known especially for its non-flammability, low gas permeability and excellent resistance to ozone, weathering and aging.
The operating temperatures of the Fluorocarbon Rubber range between -20°C and +200°C (for a short period of time up to +230°C). Suitable formulated FKM can be used down to -35°C. FKM is also often used with mineral based oils and greases at high temperatures.

EPDM (Ethylene Propylene Diene Rubber)

EPDM shows good heat, ozone and aging resistance. In addition they also exhibit high levels of elasticity, good low temperature behaviour as well as good insulating properties.
The operating temperatures of applications for EPDM range between -45°C and +150°C (for a short period of time up to +175°C). With sulphur cured types the range is reduced to -45°C and +120°C (for short period of time up to +150°C).
EPDM can often be found in applications with brake fluids (based on glycol) and hot water.

HNBR (Hydrogenated Nitrile Butadiene Rubber)

HNBR is made via selective hydrogenation of the NBR butadiene groups. The properties of the HNBR rubber depend on the ACN content which ranges between 18% and 50% as well as on the degree of saturation. HNBR shows good mechanical properties.
The operating temperature of HNBR ranges between -30°C and +140°C (for a short period of time up to +160°C) in contact with mineral oils and greases. Special types can be used down to -40°C.

VMQ (Silicone Rubber)

VMQ shows excellent heat resistance, cold flexibility, dielectric properties and especially good resistance to weather, ozone an UV rays.
Specific VMQ formulations are resistant to aliphatic engine and gear oils, water up to +100°C and high-molecular chlorinated hydrocarbons. The temperature range is between -60°C and +200°C (temporary up to +230°C).

FVMQ (Fluorosilicone Rubber)

FVMQ has a good heat resistance, very good low temperature flexibility, good electrical properties and excellent resistance to weather, ozone and UV rays. FVMQ shows a significant better chemical resistance than standard Silicone especially in hydrocarbons, aromatic mineral oils, fuel and low molecular aromatic hydrocarbons e.g. Benzene and Toluene. The temperature range is between -55°C and +175°C (temporary up to +200°C).

CR (Chloroprene Rubber)

In general the CR materials show relatively good resistances to ozone, weathering, chemicals and aging. Also they show good non-flammability, good mechanical properties and cold flexibility.
The operating temperatures range between -40°C and +100°C (for a short period of time up to +120°C). Special types can be used down to -55°C.
CR materials are found in sealing applications such as refrigerants, for outdoor applications and in the glue industry.

ACM (Polyacrylate Rubber)

ACM shows excellent resistance to ozone, weathering and hot air, although it shows only a medium physical strength, low elasticity and a relatively limited low temperature capability.
The operating temperatures range from -20°C and +150°C (for a short period of time up to +175°C). Special types can be used down to -35°C.
ACM-materials are mainly used in automotive applications which require special resistance to lubricants containing many additives (incl. sulphur) at high temperatures.

FFKM (Perfluoro Rubber)

Perfluoroelastomers show broad chemical resistance similar to PTFE as well as good heat resistance. They show low swelling with almost all media.
Depending on the material the operating temperatures range between -25°C and +240°C. Special types can be used up to +325°C.
Applications for FFKM can be mostly found in the chemical and process industries and in all applications with either aggressive environments or high temperatures.

Polyurethane (Zurcon® Polyurethane)

Polyurethanes are an exceptionally complex material group. They are individually designed and fit various applications' needs. Therefore it is not possible to unify the materials' properties.
Zurcon® polyurethane materials from Trelleborg Sealing Solutions are customized to appropriate applications and stand out due to their excellent elastic properties and optimum abrasion resistance. Outstanding tensile strength, low compression set and good resistance to O_2 and O_3 are further significant characteristics. Depending on the individual Zurcon® polyurethane type the application temperature range from below -50°C up to +110°C, temporary even higher, is feasible.

Chemical compatibility

For the pre-selection of a suitable material group a comprehensive chemical compatibility guide is available. This can be downloaded from our website www.tss.trelleborg.com or you can contact your local Trelleborg Sealing Solutions company for further details.

It is important to recognise that when using this guide, the ratings shown are based on published data and immersion tests. These tests are conducted under laboratory conditions predominantly at room temperature and may not represent adequately the conditions in the field. Relative short term laboratory tests may not pick up all the additives and impurities which may exist in long term service applications.

Care must be taken to ensure that all aspects of the application are considered carefully before a material is selected. For example at elevated temperatures some aggressive fluids can cause a much more marked effect on an elastomer than at room temperature.

Physical properties as well as fluid compatibility need to be considered. Compression set, hardness, abrasion resistance and thermal expansion can influence the suitability of a material for a particular application.

It is recommended that users conduct their own tests to confirm the suitability of the selected material for each application.

Our experienced technical staff can be consulted for further information on specific applications.

B.1.3 Characteristics and inspection of elastomers

Hardness

One of the most often named properties regarding Polymer materials is hardness. Even so the values can be quite misleading.

Hardness is the resistance of a body against penetration of an even harder body - of a standard shape defined pressure.

There are two procedures for hardness tests regarding test samples and finished parts made out of elastomer material:

1. Shore A/D
   according to ISO 868 / ISO 7619 / DIN 53 505 / ASTM D 2240
   Measurement for test samples
2. Durometer IRHD (International Rubber Hardness Degree) according to ISO 48 / ASTM 1414 and 1415
   Measurement of test samples and finished parts

The hardness scale has a range of 0 (softest) to 100 (hardest). The measured values depend on the elastic qualities of the elastomers, especially on the tensile strength.

The test should be carried out at temperatures of 23 ±2°C - not earlier than 16 hours after the last vulcanisation process (manufacturing stage). If other temperatures are being used this should be mentioned in the test report.

Tests should only be carried out with samples which have not been previously stressed mechanically.

Hardness tests according to Shore A / D

The hardness test device Shore A (indentor with pyramid base) is a sensible application in the hardness range 10 to 90. Samples with a larger hardness should be tested with the device Shore D (indentor with spike).
Test specimen:
Diameter min. 30 mm
Thickness min. 6 mm
Upper and lower sides smooth and flat
When thin material is being tested it can be layered providing minimal sample thickness is achieved by a maximum of 3 layers. All layers must be at minimum 2 mm thick.

The measurement is done at three different places at a defined distance and time.

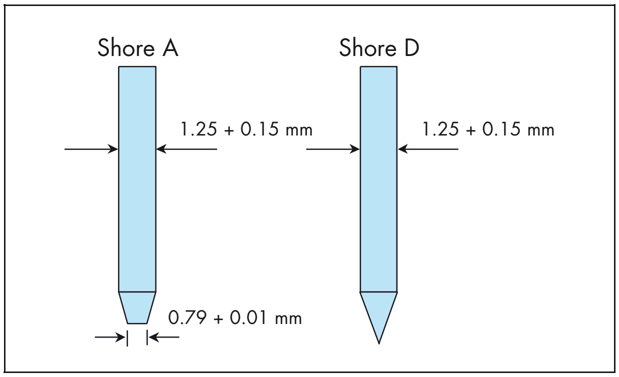


Figure 5 Indentor according to Shore A / D

Hardness test according to IRHD

The test of the Durometer according to IRHD is used with test samples as well as with finished goods.

The thickness of the test material has to be adjusted according to the range of hardness. According to ISO 48 there are two hardness ranges.

Soft: 10 to 35 IRHD ⇒ Sample thickness 10 to 15 mm / procedure "L"

Normal: over 35 IRHD ⇒ Sample thickness 8 to 10 mm / procedure "N"
Sample thickness 1.5 to 2.5 mm / procedure "M"

The hardness determined with finished parts or samples usually vary in hardness determined from specimen samples, especially those with a curved surface.

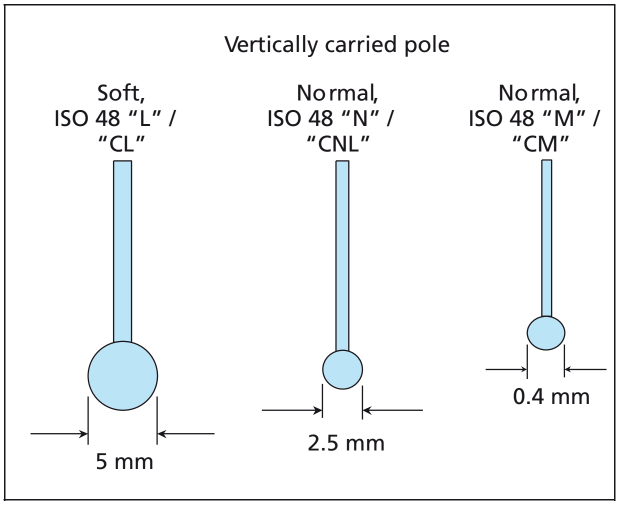


Figure 6 Indentor according to IRHD

Influencing parameters on the hardness test for polymer materials

Various sample thicknesses and geometries as well as various tests can show different hardness values even though the same materials have been used.

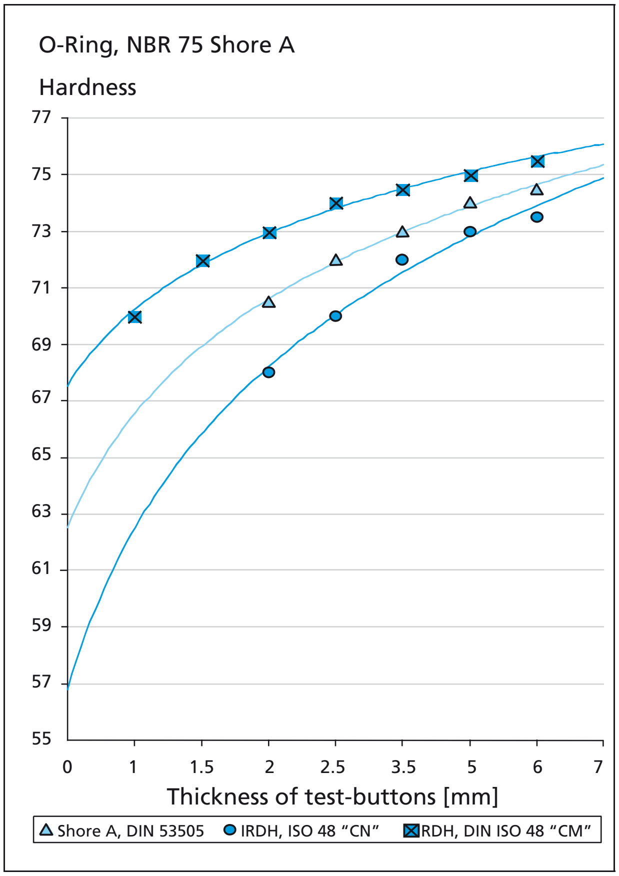


Figure 7 Ranges of hardness depending on sample thickness and test method

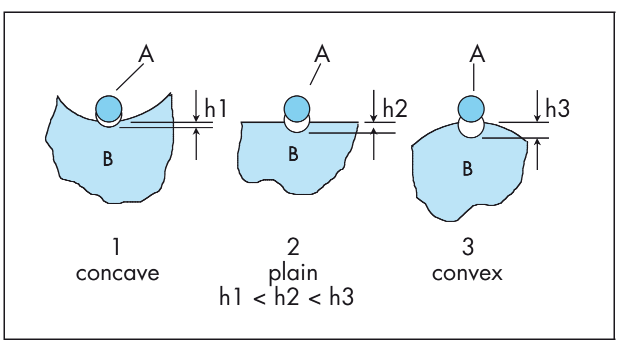


Figure 8 Range of hardness depending on surface geometry for the equivalent material characteristics.

With equivalent material characteristics of the elastomer sample B, the indentor penetrates the deepest at the surface 3 (convex) and therefore establishes the softest area.

As the concave geometry (3) has a stronger effect on smaller width O-Rings, the tolerances on hardness for widths under 2.0 mm should be increased up to +5 / -8 IRHD.

Compression set

An important parameter regarding the sealing capability is the compression set (CS) of the O-Ring material. Elastomers when under compression show aside from an elastic element also a permanent plastic deformation (Figure 9).

The compression set is determined in accordance with ISO 815 as follows:

Standard test piece:	Cylindrical disc, diameter 13 mm and height 6 mm
Deformation:	25%
Tension release time:	30 minutes

$$CS = \frac{h_0 - h_2}{h_0 - h_1} \cdot 100 (\%)$$

Where h_0 = Original height (cross section d_2)
h_1 = Height in the compressed state
h_2 = Height after tension release

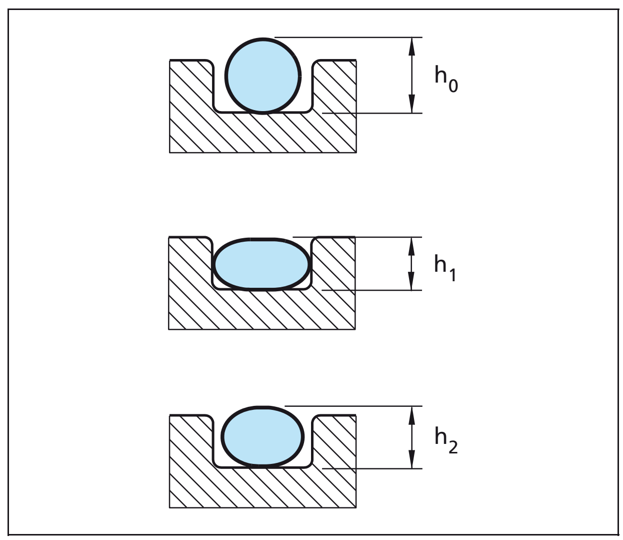


Figure 9 Illustration of the compression set

The accuracy of the measured value depends on:

- Test sample thickness
- Deformation
- Measurement deviations

Therefore the values which have been identified with the test sample cannot be transferred onto the finished part. The result of the measured finished parts are strongly influenced by geometrics and measurements as well as the measuring accuracy of the test equipment.

The following illustration shows the influence of various measuring deviations (in mm) in respect to the established compression set CS depending on the cross section of the measured O-Rings.

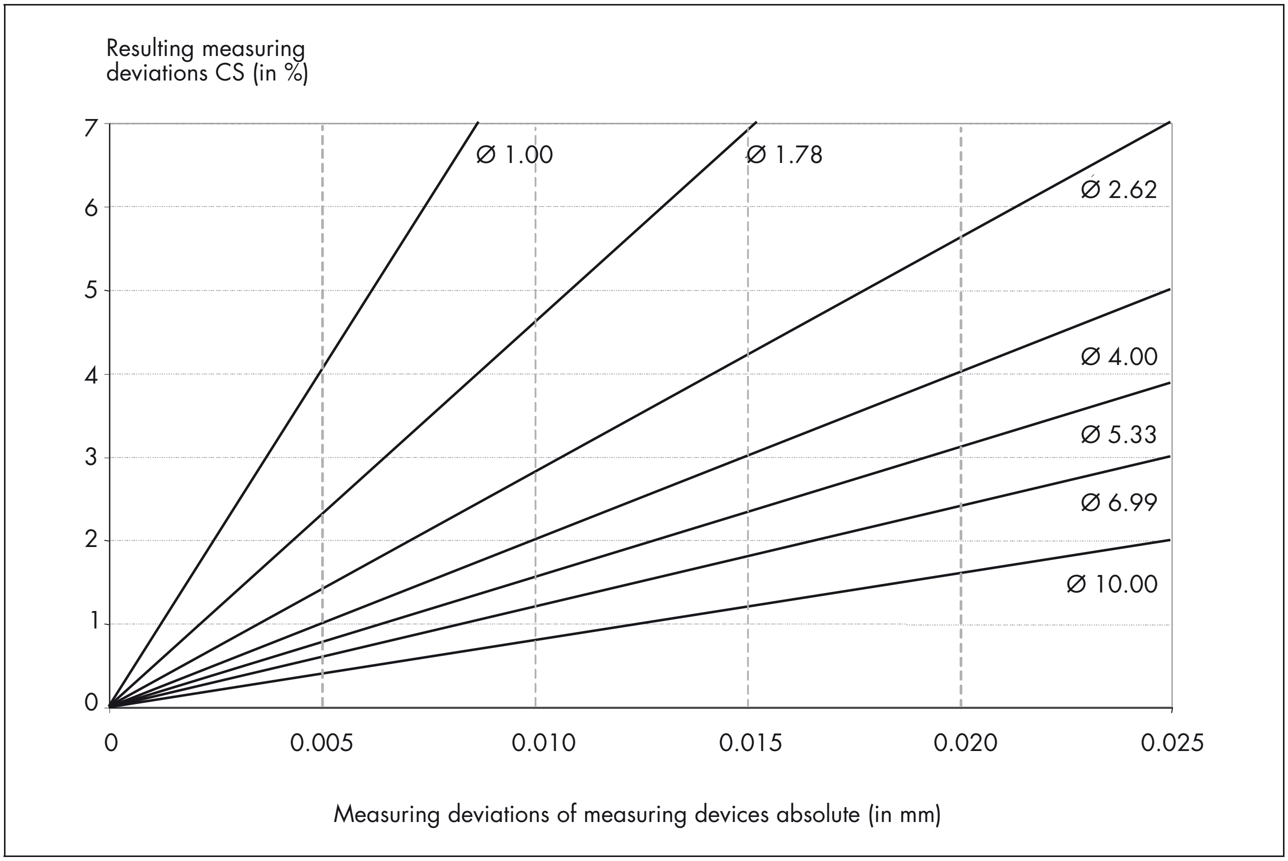


Figure 10 Measuring deviations CS depending on O-Ring cross section and measuring accuracy of the test equipment (schematic illustration)

B.1.4 Special requirements - authorities and approvals

Seals often have to meet the highest performance standards and the most stringent of environment and safety demands.
Also official authorities and associations make great demands on seals or materials which are to be used in their industries. This is often the case if seals are used for water or gas applications.
The following table shows common authorities and their requirements.

Table IV Authorities and approvals

Approval / Examination Certificate / Guideline	Application	Criteria / Standards	Tests / Examinations / Contents	Authorities / Associations	Institutes / Laboratories
ACS Licensing	Polymers exposed to drinking water	French Standard AFNOR XP P41-250, part 1-3 Synoptic Paper 1226	- Analysis of dispensing according "Synoptic Documents" - Storage test (microbe analysis)	ACS (Accréditation de conformité sanitaire)	3 certified test laboratories in France: Paris / Vandoevre / Lille
BAM recommendation	Seals for the use in gas or oxygen fittings	- reactive behaviour with lubricants - limits for pressure and temperature (DIN 4060) - seals and components		BAM (Bundesanstalt für Materialforschung und -prüfung)	BAM, Berlin
BfR Recommendation (former: BgVV)	Polymers exposed to food	BfR Guidelines ("Polymers exposed to food") various paragraphs, depending on the application of the seal	- Chemical and physical tests - Biological tests - Sterilisation tests - Taste tests	BfR (Bundesanstalt für Risikobewertung)	BAM, Berlin HY (Hygiene-Institut, Gelsenkirchen)
DVGW Release for Gas	Seals for gas services and gas applications	EN 549 EN 682		DVGW, Bonn (Deutscher Verein des Gas- und Wasserfaches e.V.)	Test Laboratory for Gas, Karlsruhe, MPA NRW, Dortmund
DVGW Release for drinking water	Seals for pro-cessing storage and distribution of drinking water	BfR Guidelines ("Polymers exposed to food")	Various classifications and tests - depending on the application	DVGW, Bonn (Deutscher Verein des Gas- und Wasserfaches e.V.)	Environmental Hygiene Institute, Gelsenkirchen TZW, Karlsruhe
DVGW W270 recommendation	Materials exposed to drinking water	DVGW, worksheet W 270	Microbiological testing: reproduction of microorganisms on materials	DVGW, Bonn (Deutscher Verein des Gas- und Wasserfaches e.V.)	TZW, Karlsruhe HY (Hygiene Institution), Gelsenkirchen
FDA guideline	Materials for food and pharmaceutical	"White List" (Register of permitted dispensing components), e.g. according to 21. CFR Part 177.2600	- Component test according "White List" - Extended for foods containing water or oil - Extraction test for polar / non polar solvents	FDA (Food and Drug Administration)	In house or external laboratories
International Military Releases	Applications for military devices	Various military specifications and standards depending on the application	- Depending on application and specification		Various test laboratories
KTW certificate	Polymers exposed to drinking water, Cold- warm- and hot water	BfR Guidelines ("Polymers exposed to food") part 1.3.13	- Extraction test - Odour- and taste test - Register of permitted components	DVGW, Bonn (Deutscher Verein des Gas- und Wasserfaches e.V.)	Environmental Hygiene Institute, Gelsenkirchen TZW, Karlsruhe BAM, Berlin

Approval / Examination Certificate / Guideline	Application	Criteria / Standards	Tests / Examinations / Contents	Authorities / Associations	Institutes / Laboratories
NSF Release	Food and Sanitary	NSF Standard criteria	Depending on application: - Test of components - Test of component group - Physical and chemical Material tests - Toxicological and micro biological tests	NSF (National Sanitation Foundation)	NSF, USA UL, USA
UL Listing	Application of seals for electrical equipment + appliances	UL-guidelines	- Chemical comparability test - Additional tests depending on application	UL (Under-writers Laboratory)	Underwriters laboratory in USA/England
USP examination	For medical and pharmaceutical use	Different specifications: USP 26 et seqq., chapter 87, 88, Class I to VI,...	Depending on specification: - intracutaneous reactive tests - systemic Injections - muscle implantations	USP (United States Pharmacopeia, USA)	Different test laboratories
WRAS Release (former: WRC)	Polymers exposed to drinking water	British Standard BS 6920 BS 2494	- Dispensing test - Microbe test - Extraction test - Hot water test	WRAS (Water Regulations Advisory Scheme)	Various accredited test laboratories in England
18-03 3-A Sanitary	Food Products	18-03 3-A Sanitary Standards for multiple-use rubber and rubber-like materials used as product contact surfaces in dairy equipment	Chemical and physical properties acc. to Class I to III	Organisations: LAFIS, IAFP, USPHS, EHEDG, DIC	Various laboratories

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B.1.5 Standard materials

The following tables show the physical properties of Trelleborg Sealing Solutions standard materials. They concern minimum values. That means that a standard material meets at least the given values. Many of the Trelleborg Sealing Solutions materials (even when defined as standard) have better physical properties.

Table V Material specification for standard NBR

				NBR 70 Shore A	NBR 80 Shore A	NBR 90 Shore A
Hardness		DIN 53 505 ASTM D 2240	Shore A	70 ± 5	80 ± 5	90 ± 5
Tensile strength		DIN 53 504 ASTM D 412	MPa N/mm²	> 14	> 12	> 10
Elongation at break		DIN 53 504 ASTM D 412	%	> 200	> 150	> 100
Compression set	24h / 100 °C	DIN ISO 815B ASTM D 395B	%	< 25	< 30	< 30
Heat aging	72h / 100 °C	DIN 53 508 ASTM D 573				
Change of hardness			Shore A	max +8	max +8	max +8
Change of tensile strength			%	max -25	max -25	max -30
Change of elongation at break			%	max -25	max -25	max -30
Resistance in ASTM-OIL # 1	72h / 100 °C	DIN 53 521 ASTM D 471				
Change of hardness			Shore A	max +6	max +6	max +6
Change of volume			%	max -8	max -8	max -8
Resistance in ASTM-OIL # 3	72h / 100 °C	DIN 53 521 ASTM D 471				
Change of hardness			Shore A	max -10	max -10	max -10
Change of volume			%	max +15	max +15	max +15
Temperature range Maximum and minimum operating temperatures depend on the specific application criteria.				-30 °C to +100 °C	-25 °C to +100 °C	-25 °C to +100 °C

Table VI Material specification for standard EPDM

				EPDM 70 Shore A sulphur cured	EPDM 70 Shore A peroxide cured	EPDM 75 Shore A peroxide cured
Hardness		DIN 53 505 ASTM D 2240	Shore A	70 ± 5	70 ± 5	75 ± 5
Tensile strength		DIN 53 504 ASTM D 412	MPa N/mm²	> 10	> 10	> 10
Elongation at break		DIN 53 504 ASTM D 412	%	> 150	> 125	> 125
Compression set	24h / 100 °C	DIN ISO 815B ASTM D 395B	%	< 20		
	24h / 150 °C		%		< 30	< 30
Heat aging	72h / 100 °C	DIN 53 508 ASTM D 573		x		
	72h / 150 °C				x	x
Change of hardness			Shore A	max +10	max +10	max +10
Change of tensile strength			%	max -10	max -20	max -20
Change of elongation at break			%	max -20	max -20	max -20
Resistance in water	72h / 100 °C	DIN 53 521 ASTM D 471				
Change of hardness			Shore A	max -10	max -3	max -3
Change of volume			%	max +10	max +3	max +3
Temperature range Maximum and minimum operating temperatures depend on the specific application criteria.				-45 °C to +120 °C	-45 °C to +140 °C	-45 °C to +140 °C

Table VII Material specification for standard Silicone

				Silicone 60 Shore A	Silicone 70 Shore A
Hardness		DIN 53 505 ASTM D 2240	Shore A	60 ± 5	70 ± 5
Tensile strength		DIN 53 504 ASTM D 412	MPa N/mm²	> 5	> 5
Elongation at break		DIN 53 504 ASTM D 412	%	> 100	> 100
Compression set	24h / 175 °C	DIN ISO 815B ASTM D 395B	%	< 35	< 35
Heat aging	72h / 225 °C	DIN 53 508 ASTM D 573			
Change of hardness			Shore A	max +15	max +15
Change of tensile strength			%	max -40	max -40
Change of elongation at break			%	max -40	max -40
Resistance in ASTM-Oil # 1	72h / 100 °C	DIN 53 521 ASTM D 471			
Change of hardness			Shore A	max -10	max -10
Change of volume			%	max +20	max +20
Temperature range Maximum and minimum operating temperatures depend on the specific application criteria.				-55 °C to +200 °C	-55 °C to +200 °C

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Table VIII   Material specification for standard FKM

				FKM 70 Shore A	FKM 75 Shore A	FKM 80 Shore A	FKM 90 Shore A
Hardness		DIN 53 505 ASTM D 2240	Shore A	70 ± 5	75 ± 5	80 ± 5	90 ± 5
Tensile strength		DIN 53 504 ASTM D 412	MPa N/mm²	> 10	> 10	> 10	> 10
Elongation at break		DIN 53 504 ASTM D 412	%	> 125	> 125	> 120	> 100
Compression set	24h / 175 °C	DIN ISO 815B ASTM D 395B	%	< 20	< 20	< 20	< 20
Heat aging	72h / 250 °C	DIN 53 508 ASTM D 573					
Change of hardness			Shore A	max +10	max +10	max +10	max +10
Change of tensile strength			%	max -25	max -25	max -25	max -25
Change of elongation at break			%	max -25	max -25	max -25	max -25
Resistance in ASTM-Oil # 3	72h / 150 °C	DIN 53 521 ASTM D 471					
Change of hardness			Shore A	max -5	max -5	max -5	max -5
Change of volume			%	max +5	max +5	max +5	max +5
Resistance in ASTM-FUEL C	72h / RT	DIN 53 521 ASTM D 471					
Change of hardness			Shore A	max -5	max -5	max -5	max -5
Change of volume			%	max +10	max +10	max +10	max +10
Temperature range Maximum and minimum operating temperatures depend on the specific application criteria.				-18 °C to +200 °C	-18 °C to +200 °C	-18 °C to +200 °C	-15 °C to +200 °C

Table IX Material specification for standard HNBR

				HNBR 70 Shore A partially saturated	HNBR 75 Shore A partially saturated
Hardness		DIN 53 505 ASTM D 2240	Shore A	70 ± 5	75 ± 5
Tensile strength		DIN 53 504 ASTM D 412	MPa N/mm²	> 15	> 15
Elongation at break		DIN 53 504 ASTM D 412	%	> 250	> 250
Compression set	24h / 125 °C	DIN ISO 815B ASTM D 395B	%	< 35	< 35
Heat aging	72h / 150 °C	DIN 53 508 ASTM D 573			
Change of hardness			Shore A	max +10	max +10
Change of tensile strength			%	max -30	max -30
Change of elongation at break			%	max -30	max -30
Resistance in ASTM-Oil # 1	72h / 150 °C	DIN 53 521 ASTM D 471			
Change of hardness			Shore A	max +10	max +10
Change of volume			%	max -10	max -10
Resistance in ASTM-Oil # 3	72h / 150 °C	DIN 53 521 ASTM D 471			
Change of hardness			Shore A	max -15	max -15
Change of volume			%	max +20	max +20
Temperature range Maximum and minimum operating temperatures depend on the specific application criteria.				-30 °C to +130 °C	-30 °C to +130 °C

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Trelleborg Sealing Solutions offers various materials, which provide additional advantages, in addition to the standard materials previously described. The advantages include a wide range of available molds, special operating temperature range, special media resistance and institutional approvals for the portable water, pharmaceutical and beverage industries.

The following table shows preferred materials, which are characterized by their wide spectrum of use. They can be used for standard applications as well as for special industrial applications.

Table X Preferred materials

Material Type	Hardness Shore A (± 5)	Color	Operating temperature range	Material code	Description
NBR Nitrile Butadiene Rubber	70	black	-30 °C to +100 °C	N7083	Preferable for **sizes acc. to AS 568 B**, preferably used for energizing elements, good overall performance
			-50 °C to +100 °C	N7T40	"Polar", **extremely good low temperature properties**, preferably used for static applications in mineral oil and for energizing elements, preferable for sizes acc. **to AS 568 B**
			-30 °C to +100 °C	N7003	Preferable for metric sizes, good overall performance, wide range of molds available
			-30 °C to +100 °C	N7024	Good overall performance, **preferable for large quantities**
			-30 °C to +100 °C	N7027	Preferable for potable water applications: **KTW, ACS, NSF61, NSF51, DIN EN 549: 0 °C / 80 °C, W270, FDA**, also suitable for use in gas applications
	90	black	-25 °C to +100 °C	N9002	Good overall performance, **wide range of molds available**

Material Type	Hardness Shore A (± 5)	Color	Operating temperature range	Material code	Description
HNBR Hydrogenated Nitrile Butadiene Rubber	70	black	-30 °C to +140 °C	H7671	Good overall performance, **wide range of molds available**
			-35 °C to +140 °C	H7503	**Wide range of operating temperature**, good resistance to mineral oil, good overall performance
FKM Fluorocarbon Rubber	70	green	-20 °C to +200 °C	V70GA	Preferable for **sizes acc. to AS 568 B**, preferably used for energizing elements, good overall performance, **DVGW, BAM**
			-18 °C to +200 °C	V70G2	Preferable for **sizes acc. to AS 568 B**, good overall performance
	75	black	-20 °C to +200 °C	VC009	Preferable for **sizes acc. to BS 4518** (metric), standard FKM
	80	green	-18 °C to +200 °C	V80G2	Good overall performance, **wide range of molds available**
		black	-18 °C to +200 °C	V8003	Good overall performance, **wide range of molds available**
			-20 °C to +200 °C	V8605	For pharmaceutical and food and beverage industries, **FDA**
	90	green	-15 °C to +200 °C	V90G1	Good overall performance, **wide range of molds available**
		black	-15 °C to +200 °C	V9670	Good overall performance, **wide range of molds available**
EPDM Ethylene Propylene Diene Rubber	70	black	-45 °C to +150 °C	E7502	Peroxide cured, for pharmaceutical and food and beverage industries, **KTW, WRAS, FDA, USP Class VI, USP 26**, plasticizer content < 3 %
			-45 °C to +125 °C	E7002	**Sulfur cured**, standard EPDM, wide range of molds available
			-45 °C to +140 °C	E7515	**Peroxide cured**, standard EPDM, wide range of molds available
			-45 °C to +150 °C	E7T41	Peroxide cured, **extremely low compression set** in hot water and steam. Excellent resistance to ozone, can be used in contact with copper and brass
			-45 °C to +140 °C	E7518	Peroxide cured, preferable for the use in potable water: **KTW, WRAS, FDA, NSF61, NSF51, W270, W534, EN 681, ACS, USP Class VI, USP 26**, plasticizer content < 1 %
VMQ Methyl Vinyl Silicon Rubber	60	red	-50 °C to +200 °C	S60R1	Good overall performance, **wide range of molds available**
	70	red	-50 °C to +200 °C	S70R2	**Sulfur cured**, good overall performance, **wide range of molds available**

The stated operating temperatures exclude any kind of load. Actual operating temperatures may differ depending on media and load type.

At time of publication the information contained in this literature, including availability or institutional approvals, is believed to be correct and accurate.

Further materials are available on request.

B.2 Design recommendations

The following design recommendations cannot be used for the special Isolast® materials. Please use the Isolast® brochure or contact our specialists for further details.

B.2.1 Installation recommendations

General recommendations

Before starting installation, check the following points:

- Lead-in chamfers made according to drawing?
- Bores deburred and edges rounded?
- Machining residues, e.g. chips, dirt and foreign particles, removed?
- Screw thread tips covered?
- Seals and components greased or oiled?
  Ensure media compatibility with the elastomer material. Trelleborg Sealing Solutions recommends to use the fluid to be sealed.
- Do not use lubricants with solid additives, e.g. molybdenum disulphide or zinc sulphide.

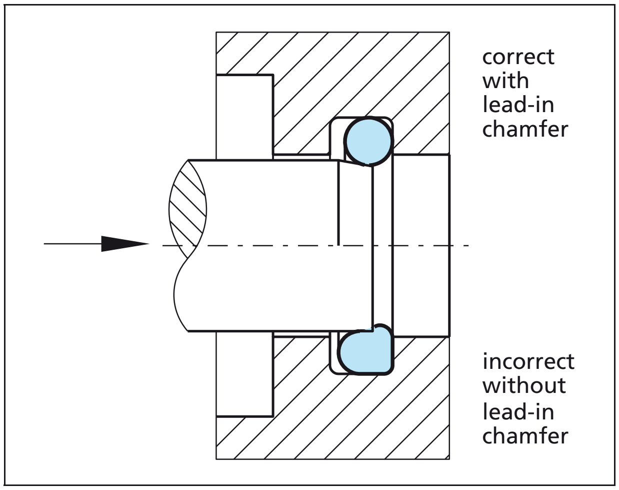

Figure 11 Rod installation with O-Ring

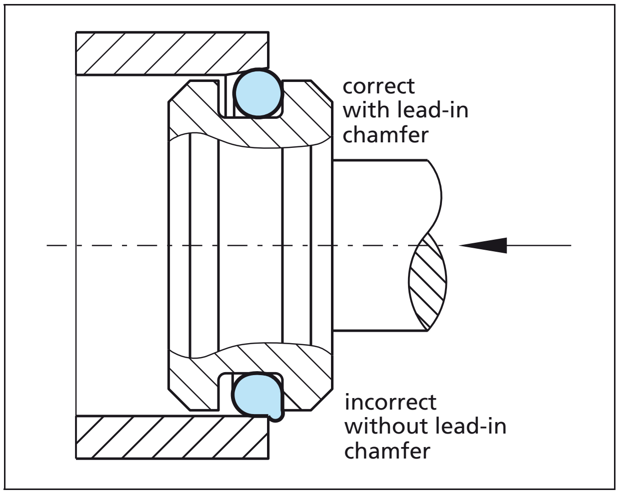

Figure 12 Piston installation with O-Ring

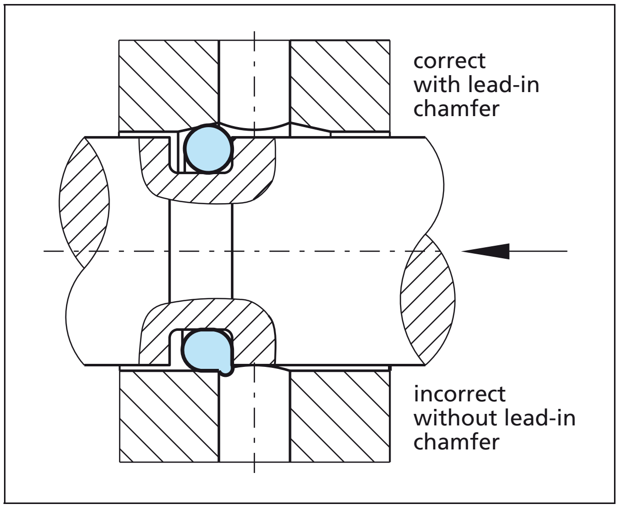

Figure 13 O-Ring installation over transverse bores

Manual installation

- Use tools without sharp edges!
- Ensure that the O-Ring is not twisted, use installation aids to assist correct positioning
- Use installation aids wherever possible
- Do not over stretch O-Rings
- Do not stretch O-Rings made out of cord at the joint.

Installation over threads, splines etc.

Should the O-Ring have to be stretched over threads, splines, keyways etc., then an assembly mandrel is essential. This mandrel can either be manufactured in a soft metal or a plastic material obviously without burrs or sharp edges.

Automatic installation

Automatic O-Ring installation requires good preparation. The surfaces of the O-Rings are frequently treated by several methods (see chapter "O-Ring friction reduced"). This offers a number of benefits during installation by

- Reducing the installation forces
- Non-stick effects, easy removal

The handling and installation of dimensionally unstable components requires a great deal of experience. Reliable automated installation thus demands special handling and packing of the O-Rings.

Please ask our specialists for further details.

B.2.2 Initial compression

An initial compression (squeeze) of the O-Ring in the groove is essential to ensure its function as a primary or secondary sealing element (Figure 14). It serves to:

- Achieve the initial sealing capability
- Bridge production tolerances
- Assure defined frictional forces
- Compensate for the compression set
- Compensate for wear

Depending on the application, the following values apply for the initial squeeze as a proportion of the cross section (d_2):

Dynamic applications:	6 to 20%
Static applications:	15 to 30%

The design of the grooves can be based on the guide values for the initial squeeze shown in the diagrams in Figure 15 and 16. These take into account the relationship between loads and cross sections according to ISO 3601-2 (version 1987).

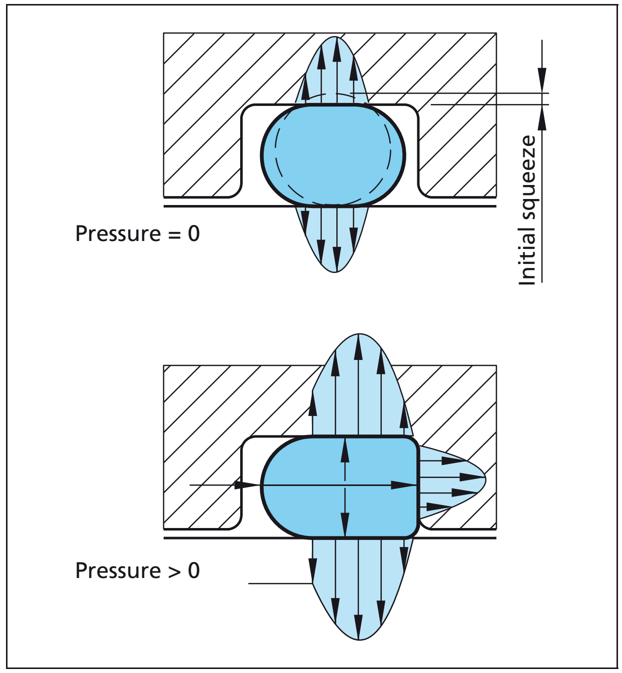

Figure 14 O-Ring contact pressure installed and under service pressure

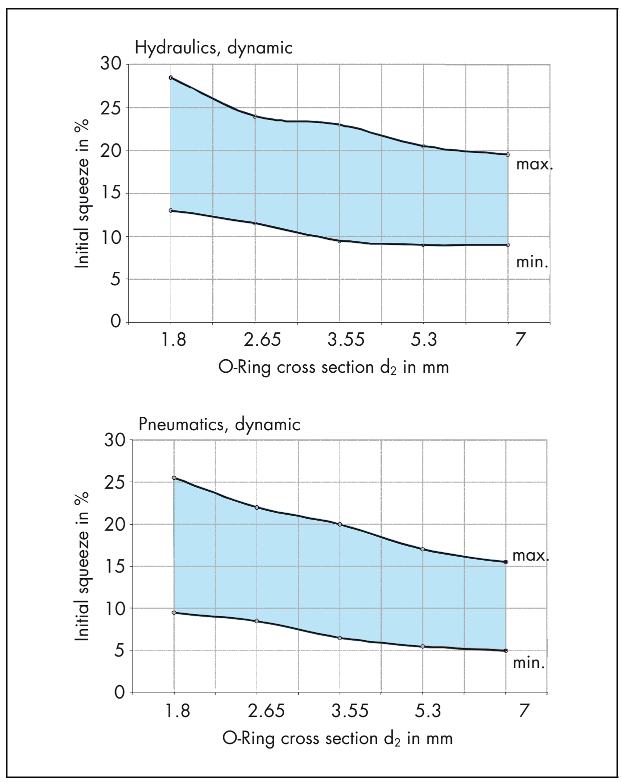

Figure 15 Permissible range of initial squeeze as a function of cross section, radial dynamic

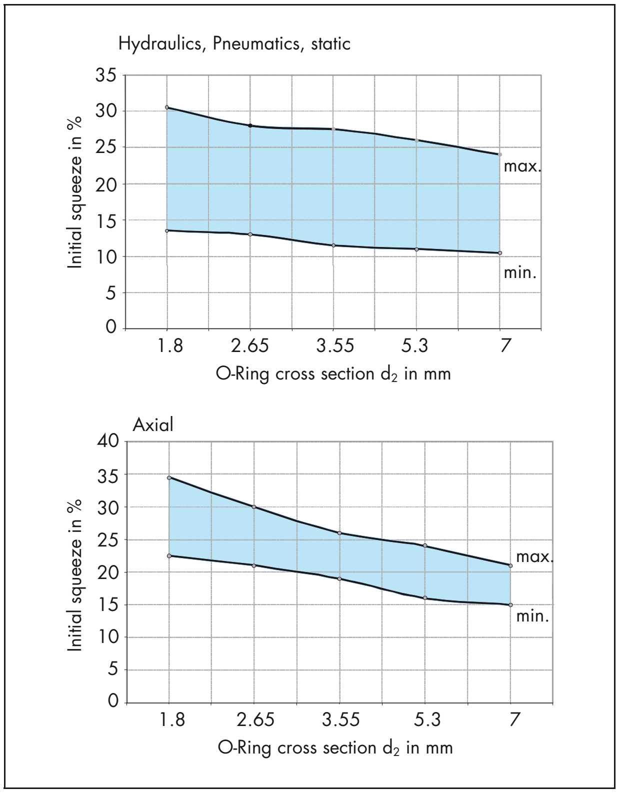

Figure 16 Permissible range of Initial squeeze as a function of cross section, radial static and axial

Compression forces

The deformation forces vary depending on the extent of the initial squeeze and the Shore hardness. Figure 17 shows the specific compression force per cm of the seal circumference as a function of the cross section.

The compression forces shown can be used to estimate the total force to be applied for static installation of O-Rings.

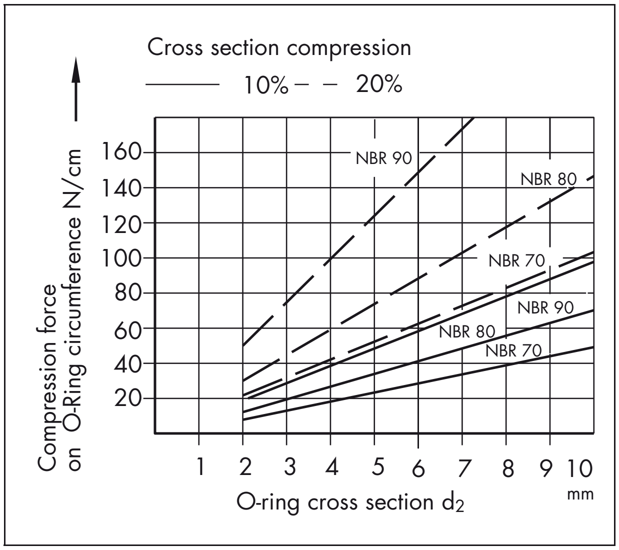

Figure 17 Compression forces on the O-Ring circumference depending on the material

B.2.3 Elongation - compression

With a radial sealing configuration, the O-Ring in an internal groove - “outside sealing“ - should be stretched over the diameter of the groove. Maximum elongation in the installed state is 6% for O-Rings with an inner diameter > 50 mm and 8% for O-Rings with an inner diameter < 50 mm.

With external grooves - “inside sealing“ - the O-Ring is preferably compressed along its circumference. The maximum circumferential compression in the installed state is 3%.

Exceeding these values will result in too large increase or decrease in the O-Ring cross section. Consequently this can effect the service life of the seal.

The reduction in cross section diameter (d_2) can be calculated as

$$Reduction_{max} = \frac{d_{2min}}{10} \cdot \sqrt{6 \cdot \left(\frac{d_{3max} - d_{1min}}{d_{1min}} \right)}$$

with d_{1min} = minimum inside diameter of the O-Ring
d_{2min} = minimum cross section of the O-Ring
d_{3max} = maximum housing diameter

but for approximation it can be assumed, in percentage, to be half the amount of stretch. An elongation of 1% corresponds to a reduction of the cross section (d_2) of approx. 0.5%.

II

B.2.4 Methods of installation and design of seal housing

Methods of installation

O-Rings can be used in components in a wide variety of ways.

During the design stage installation must be taken into consideration. In order to avoid damage during installation it should not be necessary to pass the O-Ring over edges or bores. When long sliding movements are involved, the seal seat should be recessed, if possible, or the O-Rings arranged so that they only have to travel short distances during installation to reduce risk of twisting.

Radial installation (static and dynamic)

Inner sealing
The O-Ring size should be selected so that the inside diameter d_1 has the smallest possible deviation from the diameter to be sealed d_5 (Figure 18).

Outer sealing.
The O-Ring size should be selected so that the inside diameter d_1 is equal to or smaller than groove diameter d_3 (Figure 18).

Axial installation, (static)
During axial-static installation, the direction of the pressure should be taken into consideration when choosing the O-Ring size (Figure 19). With internal pressure the O-Ring should be chosen so that the outside diameter of the O-Ring is approx. 1 to 2% larger than the outer groove diameter d_7. With external pressure the O-Ring is chosen approx. 1 to 3% smaller than the inner groove diameter d_8.

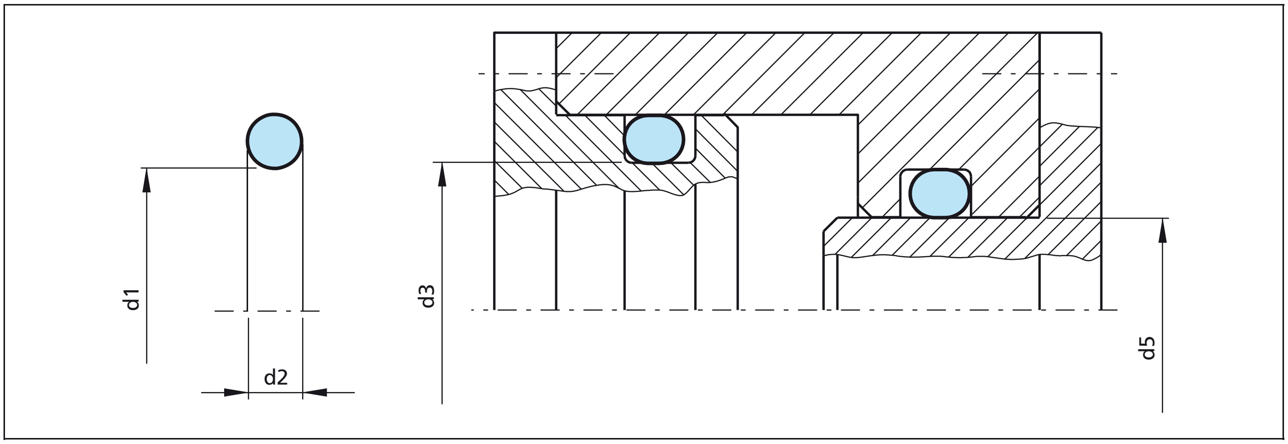

Figure 18 Radial installation, static and dynamic

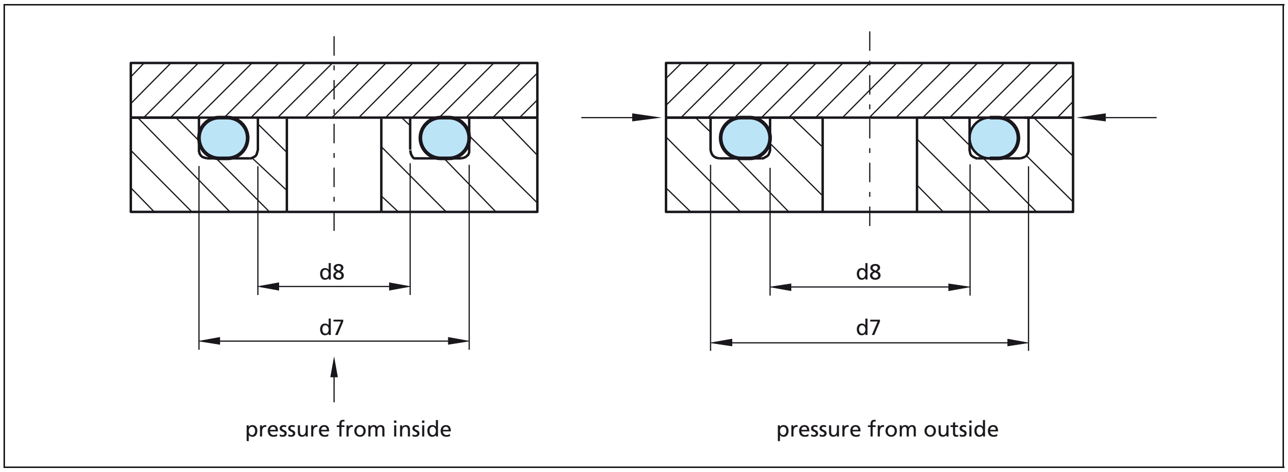

Figure 19 Axial installation, static

O-Ring as a rotary seal

In some applications, e.g. with short running periods, the O-Ring can also be used as a rotary seal for sealing shafts. In this case, the following points should be observed:

In order to be able to function as a rotary seal, O-Rings must be installed in accordance with specific guidelines, the rotary seal principle.

The rotary seal principle is based on the fact that an elongated elastomer ring contracts when heated (Joule effect). With the normal design criteria the O-Ring inside diameter d_1 will be slightly smaller than the shaft diameter, and the heat generated by friction would cause the ring to contract even more. This results in a higher pressure on the rotating shaft so that a lubricating film is prevented from forming under the seal and even higher friction occurs. The result would be increased wear and a premature failure of the seal.

Using the rotary seal principle, this is prevented by the seal ring being selected so that its inside diameter is approximately 2 to 5% larger than the shaft diameter to be sealed. The installation in the groove means that the seal ring is compressed radially and is pressed against the shaft by the groove diameter. The seal ring is thus slightly corrugated in the groove, a fact which helps to improve the lubrication.

Special materials are available for rotary seal applications. Trelleborg Sealing Solutions does not recommend the use of O-Rings as rotary seals. Please contact your local Trelleborg Sealing Solutions company for further details.

Technical data

O-Rings can be used in a wide range of applications. Temperature, pressure and media determine the choice of appropriate materials. In order to be able to assess the suitability of the O-Ring as a sealing element for a given application, the interaction of all the operating parameters have to be taken into consideration.

Working Pressure

Static application
- up to 5 MPa for O-Rings with inside diameter > 50 mm without Back-up Ring
  up to 10 MPa for O-Rings with inside diameter
- < 50 mm without Back-up Ring
  (depends on the material, the cross section and the clearance)
- up to 40 MPa with Back-up Ring
- up to 250 MPa with special Back-up Ring

Please note the permissible extrusion gaps.

Dynamic application
- Reciprocating up to 5 MPa without Back-up Ring
- Higher pressures with Back-up Ring

Speed
Reciprocating up to 0.5 m/s
Rotating up to 0.5 m/s
Depending on material and application.

Temperature
From -60 °C to +325 °C
Depending on material and media resistance.

When assessing the application criteria, the peak and continuous operating temperature and the running period must be taken into consideration. For rotating applications the temperature increase due to frictional heat must be taken into account.

Media
With the wide range of the available materials, each with different properties, it is possible to seal against practically all liquids, gases and chemicals. Please note when selecting the most suitable material the information in chapter “B.1 Materials“, and in our O-Ring Material Guide.

II

Groove design / Groove dimensions
Lead-in chamfers

Correct design can help to eliminate possible sources of damage and seal failure from the outset.

Since O-Ring are squeezed during installation, lead-in chamfers and rounded edges must be provided (Figure 20 and 21).

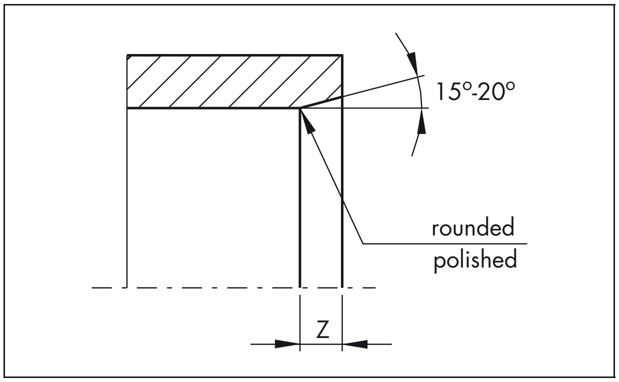

Figure 20 Lead-in chamfers for bores, tubes

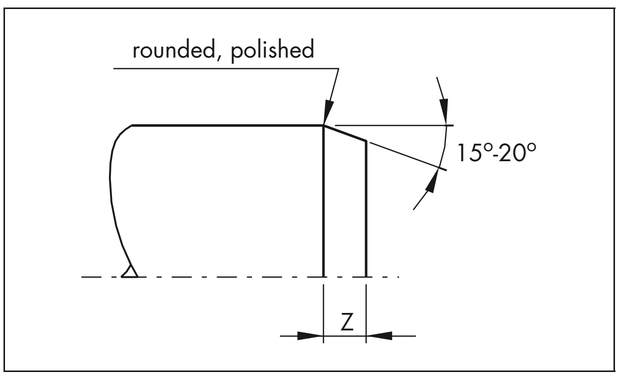

Figure 21 Lead-in chamfers for shafts, rods

The minimum length of the lead-in chamfer is listed in table XI as a function of the cross section d_2.

Table XI Lead-in chamfers

Lead-inchamfers length Z min.		O-Ring cross section d_2
15°	**20°**	
2.5	1.5	up to 1.78 1.80
3.0	2.0	up to 2.62 2.65
3.5	2.5	up to 3.53 3.55
4.5	3.5	up to 5.33 5.30
5.0	4.0	up to 7.00
6.0	4.5	above 7.00

The surface roughness of a lead-in chamfer is:
$R_Z \leq 6.3$ µm $R_a \leq 0.8$ µm

Radial clearance

The tolerances given in table XV and the maximum permissible radial clearance S (extrusion gap) given in the table XII must be maintained.

If the clearance is too large, there is a risk of seal extrusion which can result in the destruction of the O-Ring (Figure 22).

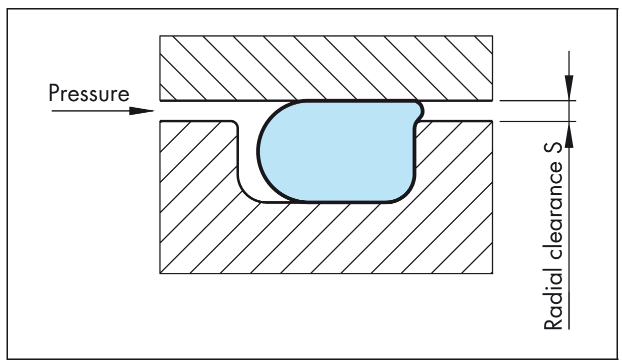

Figure 22 Radial clearance "S"

The permissible radial clearance S between the sealed parts depends on the system pressure, the cross section and the hardness of the O-Ring.

Table XII contains recommendations for the permissible clearance S as a function of O-Ring cross section and shore hardness. The table is valid for elastomeric materials with the exception of polyurethane and FEP encapsulated O-Rings.

For pressure above 5 MPa for O-Rings with Inside diameter > 50 mm and above 10 MPa for O-Rings with Inside diameter < 50 mm we recommend the use of Back-up Rings.

II

Table XII Radial clearance S

O-Ring cross section d_2	up to 2	2 - 3	3 - 5	5 - 7	above 7
O-Rings with hardness of 70 Shore A					
Pressure MPa	Radial clearance S				
≤ 3.50	0.08	0.09	0.10	0.13	0.15
≤ 7.00	0.05	0.07	0.08	0.09	0.10
≤ 10.50	0.03	0.04	0.05	0.07	0.08
O-Rings with hardness of 90 Shore A					
Pressure MPa	Radial clearance S				
≤ 3.50	0.13	0.15	0.20	0.23	0.25
≤ 7.00	0.10	0.13	0.15	0.18	0.20
≤ 10.50	0.07	0.09	0.10	0.13	0.15
≤ 14.00	0.05	0.07	0.08	0.09	0.10
≤ 17.50	0.04	0.05	0.07	0.08	0.09
≤ 21.00	0.03	0.04	0.05	0.07	0.08
≤ 35.00	0.02	0.03	0.03	0.04	0.04

These values assume that the parts are fitted concentrically to one another and do not expand under pressure. If this is not the case, the clearance should be kept correspondingly smaller.

For static applications we recommend a fit of H8/f7.

O-Rings made from polyurethane can bridge larger clearances thanks to their high extrusion resistance and greater dimensional stability. See also chapter “Polyurethane O-Rings“.

Surfaces

Under pressure, elastomers adapt to irregular surfaces. For gas or liquid tight joints, however, certain minimum demands must be made on the surface quality of the surfaces to be sealed.

Fundamentally grooves, scratches, pit marks, concentric or spiral machining scores, etc. are not permissible. Higher demands must be placed on dynamic mating surfaces than on static surfaces.

At present no uniform definitions exist for describing the mating surfaces. In practice, the specification of the R_a value is not sufficient to permit an assessment of the surface quality. Our recommendations therefore contain amongst others various terms and definitions in accordance with DIN 4768 and DIN EN ISO 4287.

Table XIII Surface finish

Type of Load	Surface	R_t µm	R_z µm	R_a µm
Radial-dynamic	Mating surface * (bore, rod, shaft)	1.0 - 2.5	0.63 - 1.6	0.1 - 0.4
	groove flanks, groove diameter	≤ 10.0	≤ 6.3	≤ 1.6
Radial-static Axial-static	Maiting surface groove flanks, groove diameter	≤ 10.0 ≤ 16.0	≤ 6.3	≤ 1.6
	For pulsating pressures Mating surface groove flanks, groove diameter	≤ 6.3 ≤ 10.0	≤ 6.3	≤ 0.8 ≤ 1.6

* spiralfree grinding

The above is for guidance only and covers the majority of sealing applications. However Trelleborg Sealing Solutions should be consulted in areas of particular concern.

II

Trapezoidal groove

The trapezoidal (dovetail) groove should only be used in special cases, e.g. overhead installation, in order to retain the O-Ring (Figure 23). The installation dimensions are summarised in table XIV. The trapezoidal groove is only recommended for O-Ring cross section from 3.53 mm. The inside diameter of the O-Ring results from the mean groove diameter minus the cross section.

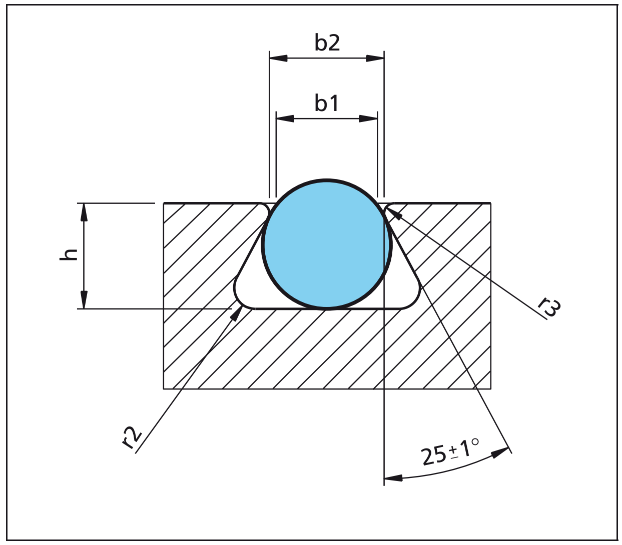

Figure 23 Installation in trapezoidal groove

Table XIV Installation dimensions for trapezoidal groove

O-Ring cross section d2	Groove dimensions				
	Groove width **b1** ±0.05	Groove width **b2** ±0.05	Groove depth **h** ±0.05	Radius (max.)	
				r3	**r2**
3.53 3.55	2.90	3.20	2.90	0.25	0.80
4.00	3.40	3.70	3.20	0.25	0.80
5.00	4.30	4.60	4.20	0.25	0.80
5.33 5.30	4.60	4.90	4.60	0.25	0.80
5.70	4.75	5.25	4.80	0.40	0.80
6.00	5.05	5.55	5.10	0.40	0.80
7.00	6.00	6.50	6.00	0.40	1.60
8.00	6.85	7.45	6.90	0.50	1.60
8.40	7.25	7.85	7.30	0.50	1.60

Rectangular groove

A rectangular groove is preferred for all new designs. Designs with bevelled groove flanks up to 5° are permissible. If Back-up Rings are used, straight groove flanks are necessary.

To reduce risk of extrusion the radius r ideally should not exceed the maximum permissible radial clearance S (see table XII).

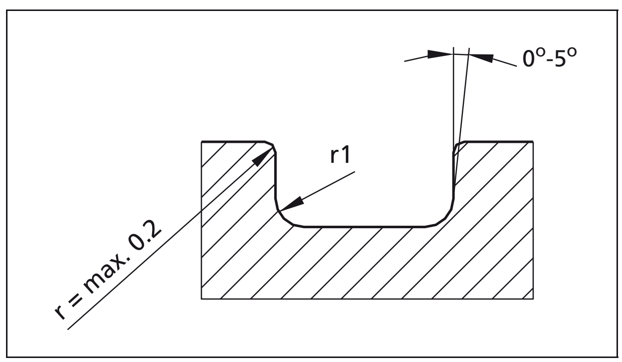

Figure 24 Groove specifications

Installation recommendations

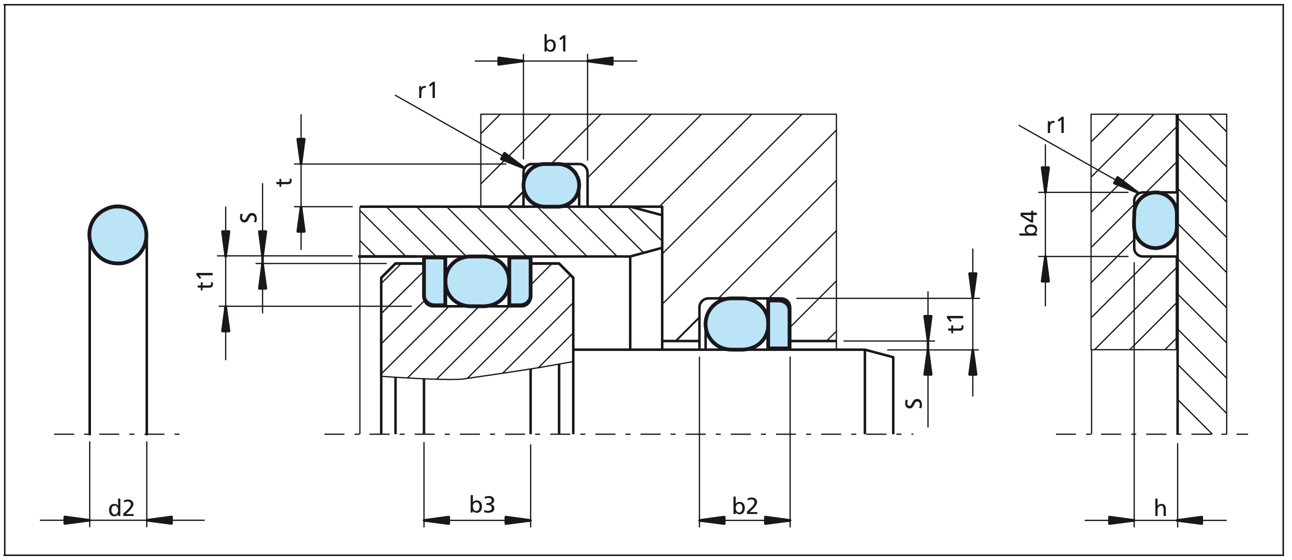

Figure 25 Installation drawing

Radial clearance, see chapter “Design Recommendations“, page .

Surface specifications, see chapter “Design Recommendations“, page .

Groove width b2 and b3: When using Back-up Rings the groove is to be widened by the corresponding Back-up Ring thickness (b2: one Back-up Ring, b3:two Back-up Rings).

Table XV Installation dimensions

Cross section	Radial installation			Axial installation		Radius[1)]
	Groove depth		Groove width	Groove depth	Groove width	
d2	Dynamic **t1** +0.05	Static **t** +0.05	**b1** +0.2	**h** +0.05	**b4** +0.2	**r1**± 0.2
0.50	-	**0.35**	0.80	0.35	0.80	0.20
0.74	-	**0.50**	1.00	0.50	1.00	0.20
1.00	-	**0.70**	1.40	0.70	1.40	0.20
1.02	-	**0.70**	1.40	0.70	1.40	0.20
1.20	-	**0.85**	1.70	0.85	1.70	0.20
1.25	-	**0.90**	1.70	0.90	1.80	0.20
1.27	-	**0.90**	1.70	0.90	1.80	0.20
1.30	-	**0.95**	1.80	0.95	1.80	0.20
1.42	-	**1.05**	1.90	1.05	2.00	0.30
1.50	1.25	**1.10**	2.00	1.10	2.10	0.30
1.52	1.25	**1.10**	2.00	1.10	2.10	0.30
1.60	1.30	**1.20**	2.10	1.20	2.20	0.30
1.63	1.30	**1.20**	2.10	1.20	2.20	0.30
1.78*	1.45	**1.30**	2.40	1.30	2.60	0.30
1.80	1.45	**1.30**	2.40	1.30	2.60	0.30
1.83	1.50	**1.35**	2.50	1.35	2.60	0.30

Cross section	Radial installation			Axial installation		Radius[1)]
	Groove depth		Groove width	Groove depth	Groove width	
d2	Dynamic **t1** +0.05	Static **t** +0.05	**b1** +0.2	**h** +0.05	**b4** +0.2	**r1**± 0.2
1.90	1.55	**1.40**	2.60	1.40	2.70	0.30
1.98	1.65	**1.50**	2.70	1.50	2.80	0.30
2.00	1.65	**1.50**	2.70	1.50	2.80	0.30
2.08	1.75	**1.55**	2.80	1.55	2.90	0.30
2.10	1.75	**1.55**	2.80	1.55	2.90	0.30
2.20	1.85	**1.60**	3.00	1.60	3.00	0.30
2.26	1.90	**1.70**	3.00	1.70	3.10	0.30
2.30	1.95	**1.75**	3.10	1.75	3.10	0.30
2.34	1.95	**1.75**	3.10	1.75	3.10	0.30
2.40	2.05	**1.80**	3.20	1.80	3.30	0.30
2.46	2.10	**1.85**	3.30	1.85	3.40	0.30
2.50	2.15	**1.90**	3.30	1.85	3.40	0.30
2.62*	2.25	**2.00**	3.60	2.00	3.80	0.30
2.65	2.25	**2.00**	3.60	2.00	3.80	0.30
2.70	2.30	**2.05**	3.60	2.05	3.80	0.30
2.80	2.40	**2.10**	3.70	2.10	3.90	0.60
2.92	2.50	**2.20**	3.90	2.20	4.00	0.60
2.95	2.50	**2.20**	3.90	2.20	4.00	0.60
3.00	2.60	**2.30**	4.00	2.30	4.00	0.60
3.10	2.70	**2.40**	4.10	2.40	4.10	0.60
3.50	3.05	**2.65**	4.60	2.65	4.70	0.60
3.53*	3.10	**2.70**	4.80	2.70	5.00	0.60
3.55	3.10	**2.70**	4.80	2.70	5.00	0.60
3.60	3.15	**2.80**	4.80	2.80	5.10	0.60
4.00	3.50	**3.10**	5.20	3.10	5.30	0.60
4.50	4.00	**3.50**	5.80	3.50	5.90	0.60
5.00	4.40	**4.00**	6.60	4.00	6.70	0.60
5.30	4.70	**4.30**	7.10	4.30	7.30	0.60
5.33*	4.70	**4.30**	7.10	4.30	7.30	0.60
5.50	4.80	**4.50**	7.10	4.50	7.30	0.60
5.70	5.00	**4.60**	7.20	4.60	7.40	0.60
6.00	5.30	**4.90**	7.40	4.90	7.60	0.60
6.50	5.70	**5.40**	8.00	5.40	8.20	1.00
6.99*	6.10	**5.80**	9.50	5.80	9.70	1.00
7.00	6.10	**5.80**	9.50	5.80	9.70	1.00
7.50	6.60	**6.30**	9.70	6.30	9.90	1.00
8.00	7.10	**6.70**	9.80	6.70	10.00	1.00
8.40	7.50	**7.10**	10.00	7.10	10.30	1.00
9.00	8.10	**7.70**	10.60	7.70	10.90	1.50
9.50	8.60	**8.20**	11.00	8.20	11.40	1.50

Cross section	Radial installation			Axial installation		Radius[1)]
	Groove depth		Groove width	Groove depth	Groove width	
	Dynamic	Static				
d2	**t1** +0.05	**t** +0.05	**b1** +0.2	**h** +0.05	**b4** +0.2	**r1**± 0.2
10.00	9.10	**8.60**	11.60	8.60	12.00	2.00
12.00	11.00	**10.60**	13.50	10.60	14.00	2.00

* Preferred sizes

1) If a Back-up Ring is used the recommended radius r1 should always be r1=0.25 ±0.2mm.

The given installation dimensions cannot be used for FFKM materials (Isolast®). Please use the Isolast® brochure or contact our specialists for further details.

II

D.3 PTFE O-Rings

O-Rings in Polytetrafluoroethylene (PTFE) are closed, circular rings with annular cross section. The dimensions are - as with the elastomer O-Ring - characterised by the inside diameter d_1 and the cord diameter d_2 (Figure 33). PTFE O-Rings are not moulded but produced by machining. The rings can therefore be manufactured in all sizes.

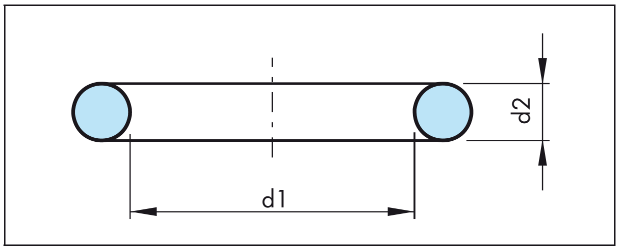

Figure 33 O-Ring dimensions

Advantages

- Very good chemical resistance, compatible with most liquids and chemicals, with the exception of liquid alkaline metals and some fluorine compounds.
- Wide temperature range from approx. -200 °C to +260 °C
- Suitable for contact with foodstuffs, pharmaceutical and medicinal products
- Physiologically safe, can be sterilised
- Low friction, no adhesion
- Available for all diameters up to approx. 1,000 mm.

Applications

Fields of application

PTFE O-Rings are used wherever the chemical and thermal resistance of the normal elastomer O-Rings is no longer sufficient. These are primarily applications in the chemical industry, foodstuffs industry, pharmaceuticals and medical technology. PTFE O-Rings are used only as static seals, e.g. on flange connections, on covers, etc.

Technical data

Working pressure:	Up to 40 MPa
Temperature:	-200 °C to +260 °C
Media:	Practically all liquids, gases and chemicals

Materials

Standard material: Virgin, unfilled PTFE (polytetrafluoroethylene), Material Code PT00

PTFE is a partially crystalline thermoplastic characterised by a very high chemical and thermal resistance. PTFE has the highest resistance to chemicals of all plastics and can be used for almost any application. It has a slightly limited resistance to molten alkaline metals, to elementary fluorine and to certain halogen materials.

The material undergoes no changes on exposure to ageing, light and ozone. The water absorption rate is less than 0.01%.

Design recommendations

PTFE O-Rings have low elasticity. The O-Ring size should therefore be chosen to suit the nominal diameter (rod or bore) to be scaled. Installation in axial easily accessible and radial split grooves is to be preferred.

The general information on the construction, design and surfaces given for the elastomer O-Rings applies also to PTFE O-Rings.

Methods of installation

PTFE O-Ring can only be stretched or compressed to a very limited extent during installation.

During installation, e.g. on flanges, the cold flow tendency of the thermoplastic PTFE should be taken into consideration. Under pressure, PTFE deforms plastically also in the cold state, i.e. a permanent deformation takes place. If flange seals are not tightened sufficiently to give metal/metal contact, the elastic deformation and thus the elastic tension can deteriorate.

Installation rocommendations

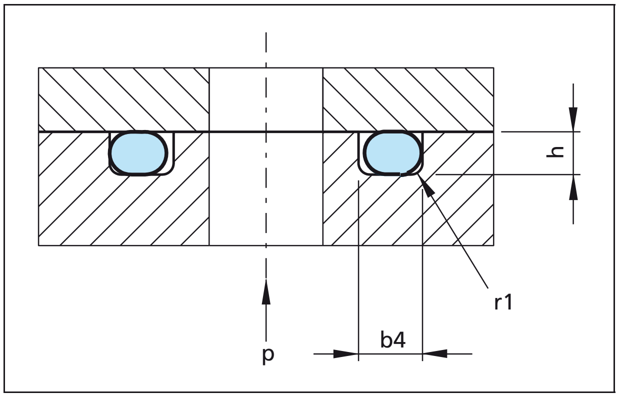

Figure 34 Axial installation, static, inside pressure

Table XXVII Installation dimensions

Cross section	Groove dimensions		Radius
d_2	Groove depth **h** +0.05	Groove width b_4 +0.1	r_1
1.50	1.30	1.7	0.2
1.60	1.40	1.8	0.3
1.78 1.80	1.60	2.0	0.4
2.00	1.80	2.2	0.5
2.40	2.15	2.6	0.5
2.50	2.25	2.8	0.5
2.62 2.65	2.35	2.9	0.6
3.00	2.70	3.3	0.8
3.53 3.55	3.15	3.9	1.0
4.00	3.60	4.4	1.0
5.00	4.50	5.5	1.0
5.33 5.30	4.80	5.9	1.2
5.70	5.10	6.3	1.2
6.00	5.60	6.6	1.2
7.00	6.30	7.7	1.5
8.00	7.20	8.8	1.5
8.40	7.55	9.2	2.0

II

Available dimensions

PTFE O-Rings are available in the same dimensions as the elastomer O-Rings. See O-Ring dimensions, page 32-97.

Ordering example

O-Ring, 40 x 3
Dimensions: Inside diameter d_1 = 40.0 mm
Cross section d_2 = 3.0 mm

O-Ring dimensions and TSS Part No. see Table XVI, page 32-97.

Ordering can also be made according to O-Ring dimensions and material.

TSS Article No. OR3004000 - PT00

TSS Part No.

Quality Index (Standard)

Material Code (Standard)

E.1 Quality criteria

The cost-effective use of seals and bearings is highly influenced by the quality criteria applied in production. Seals and bearings from Trelleborg Sealing Solutions are continuously monitored according to strict quality standards from material acquisition through to delivery.

Certification of our production plants in accordance with international standards QS 9000 / ISO 9000 meets the specific requirements for quality control and management of purchasing, production and marketing functions.

Our quality policy is consistently controlled by strict procedures and guidelines which are implemented within all strategic areas of the company.

All testing of materials and products is performed in accordance with accepted test standards and specifications, e.g. random sample testing in accordance with ISO 2859-1:2004-01 AQL 1,0 general inspection level II.

Inspection specifications correspond to standards applicable to individual product groups (e.g. for O-Rings: ISO 3601).

Our sealing materials are produced free of chlorofluorinated hydrocarbons and carcinogenic elements.

The tenth digit of our part number defines the quality characteristics of the part. A hyphen indicates compliance with standard quality criteria outlined in this catalogue. Customer-specific requirements are indicated by a different symbol in this position. Customers who require special quality criteria should contact their local Trelleborg Sealing Solutions sales office for assistance. We have experience in meeting all Customer quality requirements.

E.2 Storage and shelf live of polymer sealing material

Seals and bearings are often stored for longer time periods. Due to wrong storage conditions the physical prop-erties of elastomers may change during storage. Because of hardening, softening, crack initiation, breakage or other degradation they can become unusable. These types of material deterioration are the result of particular factors or a combination of factors such as deformation, high temperatures, contact with oxygen, ozone, light, humidity or other media.

A few simple precautions can help to extend shelf life of seals considerably. Basic instructions for the storage, cleaning and maintenance of elastomer sealing elements are described in international standards, such as:
DIN 7716 / BS 3F68, ISO 2230 or DIN 9088

These standards provide several recommendations for the storage and the shelf life of elastomers, depending on the type of material.

The following requirements for storage of elastomers and other polymers, based on the recommendations of these standards, need to be followed to preserve the physical and chemical properties of such seals.

Heat

The storage temperature should preferably be between +5 °C and +25 °C. Direct contact with heat sources such as boilers, radiators or direct sunlight are to be avoided. During storage at low temperatures, elastomers can stiffen. Therefore the handling of seals at low temperatures must be done very carefully in order to avoid deformation or damage.

Humidity

The relative humidity in the storage area should be below 70 %. Extreme humid or extreme dry conditions are to be avoided. Condensation must not develop.

Light

Elastomer seals must be protected from light sources during storage. In particular direct sunlight and strong artificial light with an ultraviolet content shall be avoided. The original storage bags, especially plastic bags, are to be favored if they provide UV protection.

In case of strong external light exposure it is recommended to mask the windows of the storage rooms with red or orange covers or screens.

Radiation

Elastomer seals are to be stored protected from all sources of ionizing radiation likely to cause damage to the stored parts.

Oxygen and ozone

If possible elastomers should be stored in the original packaging or in airtight containers in order to protect them from circulating air.

Ozone is harmful to many sealing materials. Therefore no equipment producing ozone (i.e. mercury vapor lamps, high voltage electrical equipment, electric motors or other producers of electric sparks or electric dis-charges) shall be kept in the storage areas. Also combustion emissions and organic vapors should be avoided as they may produce ozone via photochemical processes.

Deformation

If possible elastomer materials should be stored free from tension, compression or other deformation. Parts delivered in a tension-free condition should remain in their original packaging.

Contact with liquids and lubricants

Elastomer seals shall not come in contact with solvents, oils, greases or any other media at any time during storage, unless so packed by the manufacturer.

Contact with metal and non-metals

Direct contact with certain metals such as manganese, iron and particularly copper and its alloys, e.g. brass, are known to have damaging effects on some rubbers. Elastomer seals shall not be stored in contact with such metals.

Because of possible transfer of plasticizers or other ingredients, rubbers shall not be stored in contact with PVC. To avoid a mix-up different rubbers should preferably be stored separately from each other.

Cleaning

If necessary, cleaning should be carried out using soap and water (demineralized water to avoid lime stains) or denatured alcohol. However water shall not come into contact with fabric reinforced components, polyurethane rubbers or metal components without anti-corrosive protection. Disinfectants or other organic solvents as well as sharp-edged objects shall not be used. The cleaned parts should be dried at room temperature and shall not be placed near heat sources.

Shelf life and shelf life control

The shelf life of seals depends to a large extend on the polymer type. When stored under the above recommended conditions the below listed shelf life for the different materials can be considered.

NR, SBR	2 years
AU, TFE/P, Thermoplastics	4 years
CR, CSM, ECO, HNBR, IIR, NBR	6 years
ACM, AEM, EPDM	8 years
FKM, FMQ, FVMQ, VMQ	10 years
FFKM, Isolast®	18 years
PTFE	unlimited

Elastomer seals need to be checked after the above periods. If the seals are OK an extension of the shelf life is possible.

Elastomer parts and components with less than 1.5 mm thickness are stronger affected by oxidation degrada-tion even if stored under ideal conditions according to the above described. Therefore they need to be checked and tested more frequently than mentioned above.

Pre-assembled elastomer parts and seals

Generally it is not recommendable to store elastomer seals in assembled condition. If it is necessary to do so it is recommended that the units should be checked at least every six months. The maximum shelf life period a rubber component is allowed to remain assembled within a stored unit is a total of the initial period stated above and the extension period. The inspection interval will depend on the design and geometry of the unit.

ParkerSteel COLOUR CODES

II

Bright & Black Carbon Alloy Steel Bars

	BS EN 10277:1999 NUMBER	NAME	BS 970:1991	BS970:1955	Colour
1	**1.0401**	**C15**	080A15	EN 3B / EN 32B	Blue
2	**1.0715**	**11SMn30**	230M07	EN 1A Freecutting	Green
3	**1.0718**	**11SMn30Pb**	230M07Pb	EN 1A Leaded	Magenta
4	**1.0727**	**46S20**	212A42	EN 8DM	Orange
5	**1.0402/1.1151**	**C22/C22E**	070M20	EN 3B	Blue/Red
6	-	-	214M10	EN 202	Red/Green
7	**1.0511/1.1186**	**C40/C40E**	080M40	EN 8	Yellow
8	-	-	080A42	EN 8D	Yellow/Green
9	**1.0535/1.1203**	**C55/C55E**	070M55	EN 9	Yellow/Blue
10	-	-	Key Steel	EN 6A Key Steel	Red/Yellow
11	**1.0721**	**10S20**	210M15	EN 32M	Red/Brown
12	**1.0407/1.1148**	**C16/C16E**	080M15	EN 32	Red
13	-	-	606M36T	EN 16MT	White/Brown
14	-	-	605M36T	EN 16T	White
15	**1.7225**	**42CrMo4**	708M40T	EN 19T	Yellow/White
16	**1.7225**	**42CrMo4**	709M40 Annealed	EN 19 Annealed	Yellow/White/Pink
17	**1.6582**	**34CrNiMo6**	817M40T	EN 24T	Brown
18	**1.6582**	**34CrNiMo6**	817M40 Annealed	EN 24 Annealed	Brown/Green
19	**1.5752**	**15NiCr13**	655M13	EN 36	Red/White
20	-	-	665M17	EN 34	Black/Yellow
21	**1.7361**	**32CrMo12**	722M24	EN 40B	Brown/Silver
22	-	-	AISI/SAE8620*	EN 362	Purple/Yellow
23	**1.3505**	**100Cr6**	535A99*	EN 31 (Nearest Standard)	Purple/White
24	-	-	Supacut 45R*	-	Blue/Gold
25	-	-	Supacut 55T*	-	RedGold

* Not to BS 970

Stainless Steel Bars

	BS EN 10088-3:1995 NUMBER	NAME	BS 970-1:1991	Colour
1	**1.4305**	**X8CrNiS19-9**	303S31	White
2	**1.4301**	**X5CrNi18-10**	304S11	Green
3	**1.4401**	**X5CrNiMo17-12-2**	316S11	Orange
4	**1.4541**	**X6CrNiTi18-10**	321S31	Magenta
5	**1.4005**	**X12CrS13**	416S21	Red
6	**1.4057**	**X17CrNi16-2**	431S29	Brown
7	**1.4125**	**X105CrMo17**	440C	**Blue**

Aluminium Bars

	BS EN 755 pt 1-8:1996 & pt 9:2001 NUMBER	NAME	BS 1474:1987	Colour
1	**AW-6082**	**EN AW-AlSi1MgMn**	HE30	Red
2	**AW-6063**	**EN AW-AlMg0,7Si**	HE9	Blue
3	**AW-2011**	**EN AW-AlCu6BiPb**	FC1	Orange
4	**AW-2014**	**EN AW-AlCu4SiMg**	HE15	Green
5	**AW-2030**	**EN AW-AlCu4PbMg**	-	Magenta
6	**AW-6005**	**EN AW-AlSiMg**	-	Yellow
7	**AW-7075**	**EN AW-AlZn5,5MgCu**	-	White
8	**AW-6262**	**AlMg1SiPb**	-	Pink

Brass Bars

	BS EN 12164:1998 NUMBER	NAME	BS 2874:1986	Colour
1	**CW614N**	**CuZn39Pb3**	CZ121	Green
2	**CW617N**	**CuZn40Pb2**	CZ122	Red
3	**CW606N**	**CuZn37Pb2**	CZ131	White
4	**CW721R**	**CuZn40Mn1Pb1AlFeSn**	CZ114	Yellow
5	**CW712R**	**CuZn36Sn1Pb**	CZ112	Blue

For a full list of our stock range contact our sales team on 08705 783333, or FreeFax 0800 521932, and request a current Technical Stocklist or a convenient Pocket Stocklist. Alternatively, you can visit www.parkersteel.co.uk to view and order stock lengths, as well us get your material cut to length.

08705 783333 **0800 521932**
www.parkersteel.co.uk STEEL STOCKHOLDERS

Bright & Black Carbon Alloy Steel Bars

	BS EN 10277:1999 NUMBER	BS EN 10277:1999 NAME	BS 970:1991	BS970:1955	Guide to specifications
1	1.0401	C15	080A15	EN 3B / EN 32B	For general engineering, machinable & weldable. Low tensile for lightly stressed components
2	1.0715	11SMn30	230M07	EN 1A Freecutting	Freecutting steel for fast machining, long tool life and good surface finish
3	1.0718	11SMn30Pb	230M07Pb	EN 1A Leaded	Leaded freecutting steel for even faster machining & longer tool life. Excellent surface finish of machined components
4	1.0727	46S20	212A42	EN 8DM	A freecutting version of 080M40 medium carbon steel giving improved machining properties
5	1.0402/1.1151	C22/C22E	070M20	EN 3B	A general purpose, low tensile, mild steel for manufacturing lightly stressed components including studs, bolts, etc.
6	-	-	214M10	EN 202	A semi freecutting higher tensile case-hardening steel for production runs on automatics and capstans. Used for gears, cams, rollers, etc.
7	1.0511/1.1186	C40/C40E	080M40	EN 8	A medium tensile weldable steel. Fair resistance to wear. Suitable for general engineering components
8	-	-	080A42	EN 8D	A minimum 40 tn steel. As 080M40 but has a closer chemical composition and is supplied to analysis only
9	1.0535/1.1203	C55/C55E	070M55	EN 9	Medium tensile mild steel with good wear resistance. Suitable for turned parts where toughness is not of prime importance
10	-	-	Key Steel	EN 6A Key Steel	For making key ways, keys and general engineering use, normally supplied to a + tolerance to BS4235 & BS 46
11	1.0721	10S20	210M15	EN 32M	A semi freecutting carbon case-hardening steel with improved machining properties. Suitable for use on CNC lathes & automatics
12	1.0407/1.1148	C16/C16E	080M15	EN 32	A carbon case hardening steel for use where the use or cost of an alloy steel could not be justified
13	-	-	606M36T	EN 16MT	A resulphurised Manganese-Molybdenum freecutting alloy steel. Longer tool life & a very good surface finish. Supplied in 'T' condition.
14	-	-	605M36T	EN 16T	A Manganese-Molybdenum high tensile alloy steel, hardened and tempered to the 'T' condition. Excellent ductility without brittleness
15	1.7225	42CrMo4	708M40T	EN 19T	A medium carbon low alloy steel supplied in the hardened and tempered condition, can be flame hardened for increased wear resistance
16	1.7225	42CrMo4	709M40 Annealed	EN 19 Annealed	A medium carbon low alloy steel used where hardness, ductility and shock resistance are required
17	1.6582	34CrNiMo6	817M40T	EN 24T	A 1.12% nickel chrome direct hardening steel supplied hardened and tempered to the 'T' condition leaving the bars easy to machine
18	1.6582	34CrNiMo6	817M40 Annealed	EN 24 Annealed	As above but supplied in the annealed condition. For subsequent hardening and tempering after machining
19	1.5752	15NiCr13	655M13	EN 36	A 3.14% nickel chrome case hardening steel. Can be hardened and tempered to a very deep case with a very tough core
20	-	-	665M17	EN 34	A medium alloy case hardening steel which after carburising and hardening give a hard wearing case and a core strength of 700N/mm²
21	1.7361	32CrMo12	722M24	EN 40B	An alloy steel with good resistance to shock, the low temperature nitriding process gives a hard clean surface free from distortion
22	-	-	AISI/SAE8620	EN 362	A low alloy case hardening steel conforming to American specifications. High wear resistance and a strong core after carburising
23	1.3505	100Cr6	535A99	EN 31	A high carbon-chromium bearing steel for bearing rings and rolling elements
24	-	-	Supacut 45R	-	Consistant and optimum machinability with maximum response to induction hardening. Can also be manufactured to the heat treated condition 'R'
25	-	-	Supacut 55T	-	A superior alloy freecutting steel. Designed to give high tensile, high impact resistance and longer tool life

II

Stainless Steel Bars

	BS EN 10088-3:1995 NUMBER	BS EN 10088-3:1995 NAME	BS 970-1:1991	Guide to specifications
1	1.4305	X8CrNiS19-9	303S31	Good machining qualities. Can be welded but subsequent heat treatment is recommended to retain corrosion resistance. Fair forming qualities
2	1.4301	X5CrNi18-10	304S11	Reasonable weldability, general resistance to corrosion, excellent for polishing
3	1.4401	X5CrNiMo17-12-2	316S11	Stainless steel with added molybdenum for greatly improved corrosion resistance. Used in chemical, marine & the food preparation industry
4	1.4541	X6CrNiTi18-10	321S31	High resistance to corrosion. Work hardens fairly rapidly. Used in general engineering, petrol chemical & catering industries
5	1.4005	X12CrS13	416S21	13% Chrome type. Added selenium improves machining speeds. Will respond to thermal hardening and tempering. Not recommended for welding
6	1.4057	X17CrNi16-2	431S29	An 18% Cr, 2% Ni stainless steel which can be hardened and tempered. High tensile strength properties. Not recommended for welding
7	1.4125	X105CrMo17	440C	Martensitic, magnetic, Can be hardened. High carbon content for maximum hardness. Good corrosion resistance. Fair machinability

Aluminium Bars

	BS EN 755 pt 1-8:1996 & pt 9:2001 NUMBER	BS EN 755 pt 1-8:1996 & pt 9:2001 NAME	BS 1474:1987	Guide to specifications
1	AW-6082	EN AW-AlSiMgMn	HE30	A structural alloy with good stength and good resistance to corrosion
2	AW-6063	EN AW-AlMg0,7Si	HE9	Medium strength alloy for architectural extrusions. Very good for complicated shapes
3	AW-2011	EN AW-AlCu6BiPb	FC1	A freecutting alloy for use in automatic lathes
4	AW-2014	EN AW-AlCu4SiMg	HE15	Combines high strength with good ductility in the solution treated condition. Stressed components of all types in aircraft
5	AW-2030	EN AW-AlCu4PbMg	-	Excellent machining alloy. Limited corrosion resistance
6	AW-6005	EN AW-AlSiMg	-	A medium strength alloy. Heat treatable. Good weldability and corrosion resistance. Used for intricate profiles, e.g. beer barrels
7	AW-7075	EN AW-AlZn5,5MgCu	-	Heat treatable. Age hardens naturally, therefore will recover properties in heat effected zones after welding. Susceptible to stress corrosion
8	AW-6262	AlMg1SiPb	-	High strength. Very good machining. Excellent corrosion resistance

Brass Bars

	BS EN 12164:1998 NUMBER	BS EN 12164:1998 NAME	BS 2874:1986	Guide to specifications
1	CW614N	CuZn39Pb3	CZ121	High speed machining brass. Excellent machinability, very limited cold working. Also used for hot stamping
2	CW617N	CuZn40Pb2	CZ122	A freecutting brass. Most popular alloy for hot stamping. Excellent machinability but very limited cold ductility
3	CW606N	CuZn37Pb2	CZ131	A freecutting brass with improved ductility. Good machinability and some cold workability. Used for cold heading and riveting
4	CW721R	CuZn39AlFeMn	CZ114	A manganese bronze high tensile general purpose Brass with superior corrosion resistance. Used for heavy engineering. Poor machinability
5	CW712R	CuZn38Si1	CZ112	Leaded Naval Brass. The addition of tin improves corrosion resistance, especially in sea water. Lead improves machinability

A DESIGNERS'
HANDBOOK SERIES
Nº 9014

II

DESIGN GUIDELINES FOR THE SELECTION AND USE OF STAINLESS STEEL

NiDI

Distributed by
NICKEL DEVELOPMENT INSTITUTE
courtesy of AMERICAN IRON AND STEEL INSTITUTE
AND SPECIALTY STEEL INSTITUTE OF NORTH AMERICA

CONTENTS

INTRODUCTION

Stainless steels are iron-base alloys containing 10.5% or more chromium. They have been used for many industrial, architectural, chemical, and consumer applications for over a half century. Currently there are being marketed a number of stainless steels originally recognized by the American Iron and Steel Institute (AISI) as standard alloys. Also commercially available are proprietary stainless steels with special characteristics. (See Appendix A.)

With so many stainless steels from which to choose, designers should have a ready source of information on the characteristics and capabilities of these useful alloys. To fill this need, the Committee of Stainless Steel Producers initially prepared this booklet. The data was reviewed and updated by the Specialty Steel Industry of North America (SSINA). Written especially for design engineers, it presents an overview of a broad range of stainless steels – both standard and proprietary – their compositions, their properties, their fabrication, and their use. More detailed information on the 60 standard grades, with special emphasis on the manufacture, finish designations and dimensional and weight tolerances of the product forms in which they are marketed, is contained in the Iron and Steel Society of the AIME (the American Institute of Mining, Metallurgical and Petroleum Engineers) "Steel Products Manual–Stainless and Heat Resisting Steels." The AIME undertook the publication, updating and sale of this manual after the AISI discontinued publication in 1986.

II

IDENTIFICATION

Reference is often made to stainless steel in the singular sense as if it were one material. Actually there are over 50 stainless steel alloys. Three general classifications are used to identify stainless steels. They are: 1. Metallurgical Structure. 2. The AISI numbering system: namely 200, 300, and 400 Series numbers. 3. The Unified Numbering System, which was developed by American Society for Testing and Materials (ASTM) and Society of Automotive Engineers (SAE) to apply to all commercial metals and alloys.

There are also a number of grades known by common names that resemble AISI designations but that are not formally recognized by AISI. These common names, which are neither trademarks nor closely associated with a single producer, are shown and identified in the tables. These common (non-AISI) names do not appear in the ASTM specifications, so it is important to use the UNS designations with these grades.

On the following pages there is a description of these classifications. Tables 1-5 list stainless steels according to metallurgical structure: austenitic, ferritic, martensitic, precipitation hardening, and duplex.

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Austenitic stainless steels (Table 1) containing chromium and nickel are identified as 300 Series types. Alloys containing chromium, nickel and manganese are identified as 200 Series types. The stainless steels in the austenitic group have different compositions and properties, but many common characteristics. They can be hardened by cold working, but not by heat treatment. In the annealed condition, all are essentially nonmagnetic, although some may become slightly magnetic by cold working. They have excellent corrosion resistance, unusually good formability, and increase in strength as a result of cold work.

Type 304 (frequently referred to as 18-8 stainless) is the most widely used alloy of the austenitic group. It has a nominal composition of 18% chromium and 8% nickel.

Table 1
AUSTENITIC STAINLESS STEELS

TYPE	Equivalent UNS	TYPE	Equivalent UNS
201	S20100	310	S31000
202	S20200	310S	S31008
205	S20500	314	S31400
301	S30100	316	S31600
302	S30200	316L	S31603
302B	S30215	316F	S31620
303	S30300	316N	S31651
303Se	S30323	317	S31700
304	S30400	317L	S31703
304L	S30403	317LMN	S31726
302HQ	S30430	321	S32100
304N	S30451	330	N08330
305	S30500	347	S34700
308	S30800	348	S34800
309	S30900	384	S38400
309S	S30908		

Ferritic stainless steels (Table 2) are straight-chromium 400 Series types that cannot be hardened by heat treatment, and only moderately hardened by cold working. They are magnetic, have good ductility and resistance to corrosion and oxidation. Type 430 is the general-purpose stainless of the ferritic group.

Table 2
FERRITIC STAINLESS STEELS

TYPE	Equivalent UNS	TYPE	Equivalent UNS
405	S40500	430FSe	S43023
409	S40900	434	S43400
429	S42900	436	S43600
430	S43000	442	S44200
430F	S43020	446	S44600

Martensitic stainless steels (Table 3) are straight-chromium 400 Series types that are hardenable by heat treatment. They are magnetic. They resist corrosion in mild environments. They have fairly good ductility, and some can be heat treated to tensile strengths *exceeding 200,000 psi (1379 MPa).*

Type 410 is the general-purpose alloy of the martensitic group.

Table 3
MARTENSITIC STAINLESS STEELS

TYPE	Equivalent UNS	TYPE	Equivalent UNS
403	S40300	420F	S42020
410	S41000	422	S42200
414	S41400	431	S43100
416	S41600	440A	S44002
416Se	S41623	440B	S44003
420	S42000	440C	S44004

Precipitation-hardening stainless steels (Table 4) are chromium-nickel types, some containing other alloying elements, such as copper or aluminum. They can be hardened by solution treating and aging to high strength.

Table 4
PRECIPITATION HARDENING STAINLESS STEELS

UNS	UNS
S13800	S17400
S15500	S17700

II

Duplex stainless steels (Table 5) have an annealed structure which is typically about equal parts of austenite and ferrite. Although not formally defined, it is generally accepted that the lesser phase will be at least 30% by volume.

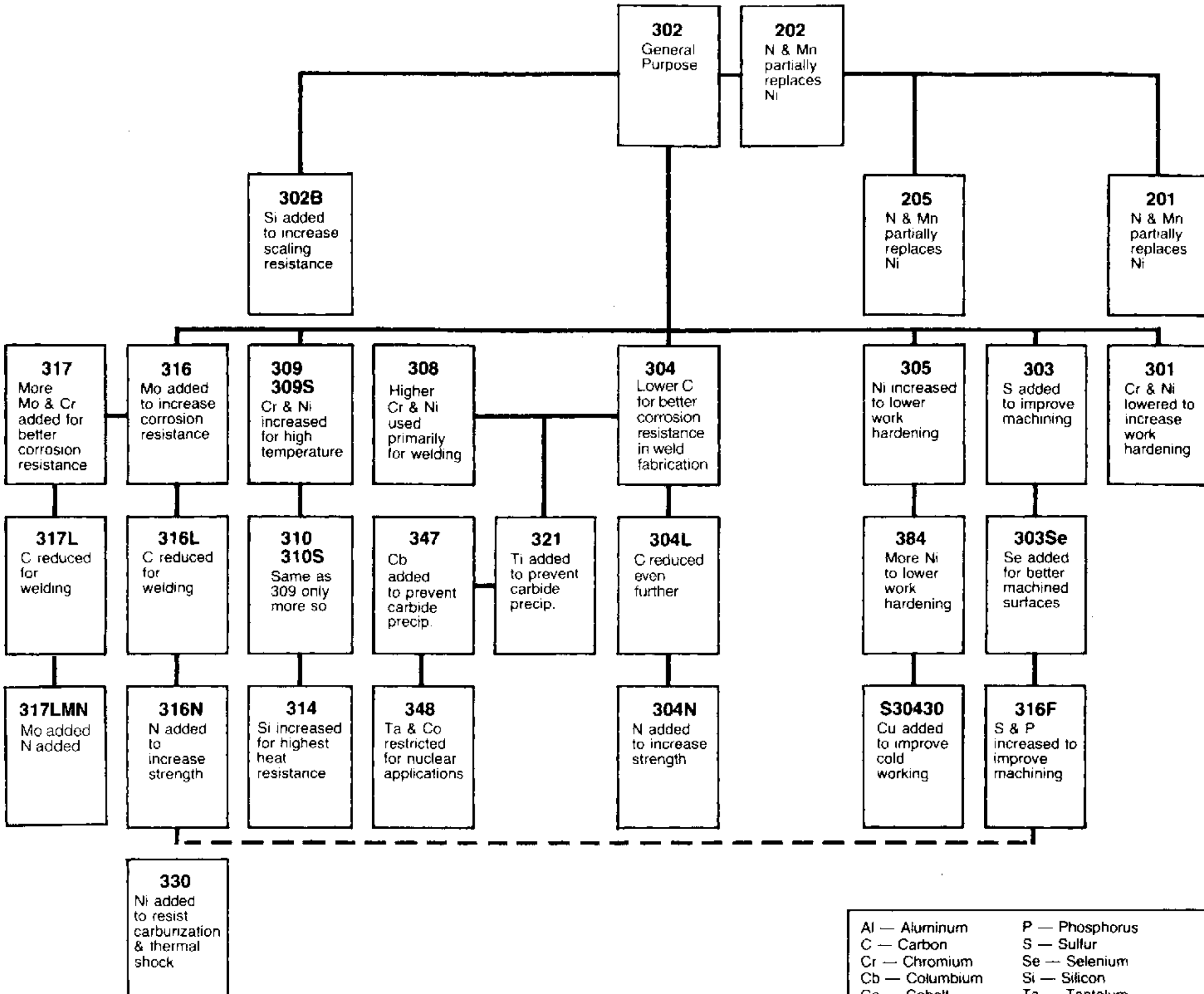
Duplex stainless steels offer several advantages over the common austenitic stainless steels. The duplex grades are highly resistant to chloride stress corrosion cracking, have excellent pitting and crevice corrosion resistance and exhibit about twice the yield stength as conventional grades. Type 329 and 2205 are typical alloys.

With respect to the Unified Numbering System, the UNS designations are shown alongside each AISI type number, in Tables 1-5, except for four stainless steels (see Tables 4 and 5) for which UNS designations only are listed.

Table 5
DUPLEX

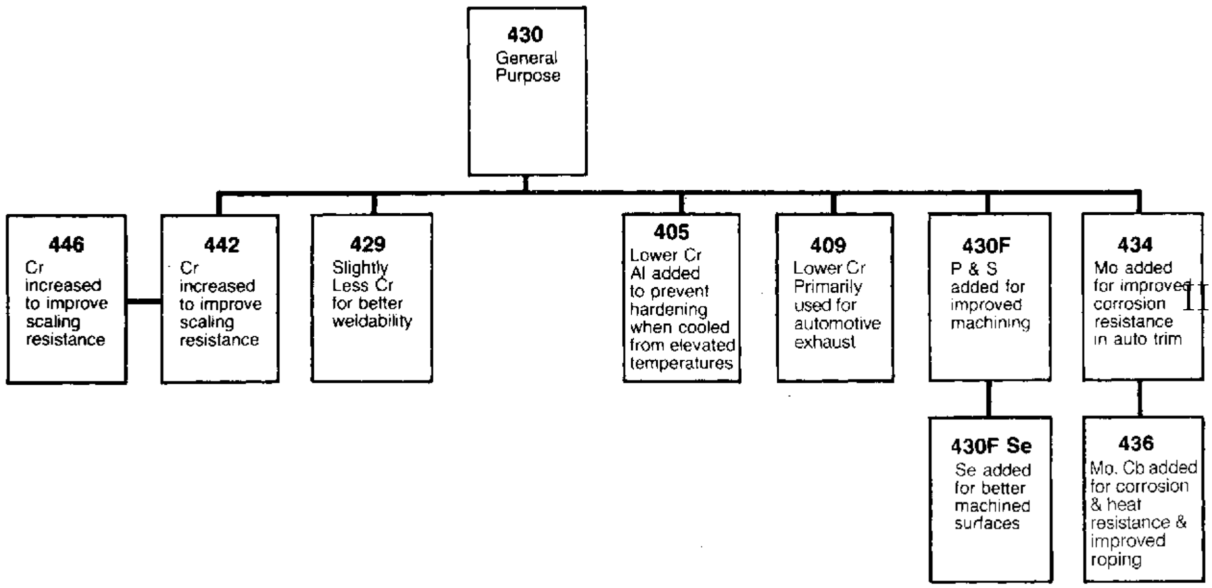
Type	UNS
329	S32900
2205	S31803,S32205

AUSTENITIC

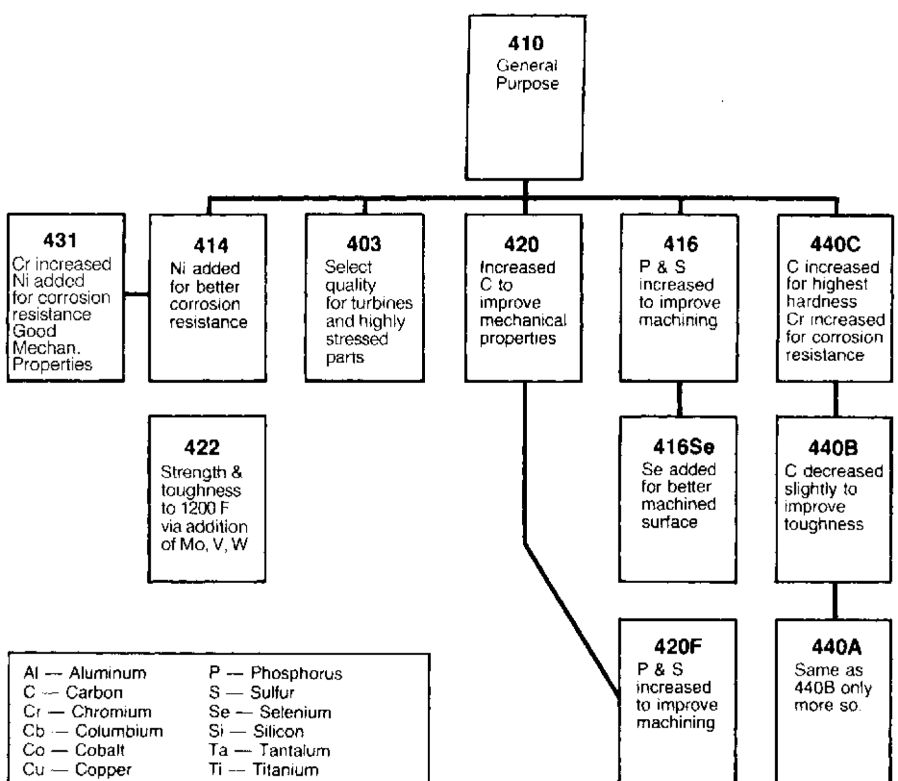


Al — Aluminum	P — Phosphorus
C — Carbon	S — Sulfur
Cr — Chromium	Se — Selenium
Cb — Columbium	Si — Silicon
Co — Cobalt	Ta — Tantalum
Cu — Copper	Ti — Titanium
Mn — Manganese	V — Vanadium
Mo — Molybdenum	W — Tungsten
N — Nitrogen	
Ni — Nickel	

FERRITIC



MARTENSITIC



Al — Aluminum	P — Phosphorus
C — Carbon	S — Sulfur
Cr — Chromium	Se — Selenium
Cb — Columbium	Si — Silicon
Co — Cobalt	Ta — Tantalum
Cu — Copper	Ti — Titanium
Mn — Manganese	V — Vanadium
Mo — Molybdenum	W — Tungsten
N — Nitrogen	
Ni — Nickel	

GUIDELINES FOR SELECTION

Stainless steels are engineering materials with good corrosion resistance, strength, and fabrication characteristics. They can readily meet a wide range of design criteria – load, service life, low maintenance, etc. Selecting the proper stainless steel essentially means weighing four elements. In order of importance, they are:

1. **Corrosion or Heat Resistance –** the primary reason for specifying stainless. The specifier needs to know the nature of the environment and the degree of corrosion or heat resistance required.

2. **Mechanical Properties –** with particular emphasis on strength at room, elevated, or low temperature. Generally speaking, the combination of corrosion resistance and strength is the basis for selection.

3. **Fabrication Operations –** and how the product is to be made is a third-level consideration. This includes forging, machining, forming, welding, etc.

4. **Total Cost –** in considering total cost, it is appropriate to consider not only material and production costs, but the life cycle cost including the cost-saving benefits of a maintenance-free product having a long life expectancy.

II

CORROSION RESISTANCE

Chromium is the alloying element that imparts to stainless steels their corrosion-resistance qualities by combining with oxygen to form a thin, invisible chromium-oxide protective film on the surface. (Figure 1. Figures are shown in Appendix B.) Because the passive film is such an important factor, there are precautions which must be observed in designing stainless steel equipment, in manufacturing the equipment, and in operation and use of the equipment, to avoid destroying or disturbing the film.

In the event that the protective (passive) film is disturbed or even destroyed, it will, in the presence of oxygen in the environment, reform and continue to give maximum protection.

The protective film is stable and protective in normal atmospheric or mild aqueous environments, but can be improved by higher chromium, and by molybdenum, nickel, and other alloying elements. Chromium improves film stability; molybdenum and chromium increase resistance to chloride penetration; and nickel improves film resistance in some acid environments.

Material Selection

Many variables characterize a corrosive environment – i.e., chemicals and their concentration, atmospheric conditions, temperature, time – so it is difficult to select which alloy to use without knowing the exact nature of the environment. However, there are guidelines:

Type 304 serves a wide range of applications. It withstands ordinary rusting in architecture, it is resistant to food-processing environments (except possibly for high-temperature conditions involving high acid and chloride contents), it resists organic chemicals, dyestuffs, and a wide variety of inorganic chemicals. Type 304 L (low carbon) resists nitric acid well and sulfuric acids at moderate temperature and concentrations. It is used extensively for storage of liquified gases, equipment for use at cryogenic temperatures (304N), appliances and other consumer products, kitchen equipment, hospital equipment, transportation, and waste-water treatment.

Type 316 contains slightly more nickel than Type 304, and 2-3% molybdenum giving it better resistance to corrosion than Type 304, especially in chloride environments that tend to cause pitting. Type 316 was developed for use in sulfite pulp mills because it resists sulfuric acid compounds. Its use has been broadened, however, to handling many chemicals in the process industries.

Type 317 contains 3-4% molybdenum (higher levels are also available in this series) and more chromium than Type 316 for even better resistance to pitting and crevice corrosion.

Type 430 has lower alloy content than Type 304 and is used for highly polished trim applications in mild atmospheres. It is also used in nitric acid and food processing.

Type 410 has the lowest alloy content of the three general-purpose stainless steels and is selected for highly stressed parts needing the combination of strength and corrosion resistance, such as fasteners. Type 410 resists corrosion in mild atmospheres, steam, and many mild chemical environments.

Type 2205 may have advantages over Type 304 and 316 since it is highly resistant to chloride stress corrosion cracking and is about twice as strong.

Table 6 lists the relative corrosion resistance of the AISI standard numbered stainless steels in seven broad categories of corrosive environments. Table 7 details more specific environments in which various grades are used, such as acids, bases, organics, and pharmaceuticals.

The above comments on the suitability of stainless steels in various environments are based on a long history of successful application, but they are intended only as guidelines. Small differences in chemical content and temperature, such as might occur during processing, can affect corrosion rates. The magnitude can be considerable, as suggested by Figures 2 and 3. Figure 2 shows small quantities of hydrofluoric and sulfuric acids having a serious effect on Type 316 stainless steel in an environment of 25% phosphoric acid, and Figure 3 shows effects of temperature on Types 304 and 316 in very concentrated sulfuric acid.

Service tests are most reliable in determining optimum material, and ASTM G4 is a recommended practice for carrying out such tests. Tests should cover conditions both during operation and shutdown. For instance, sulfuric, sulfurous and polythionic acid condensates formed in some processes during shutdowns may be more corrosive than the process stream itself. Tests should be conducted under the worst operating conditions anticipated.

Several standard reference volumes discuss corrosion and corrosion control, including Uhlig's *Corrosion Handbook;* LaQue and Copsons' *Corrosion Resistance Of Metals and Alloys;* Fontana and Greens' *Corrosion Engineering; A Guide to Corrosion Resistance* by Climax Molybdenum Company; the *Corrosion Data Survey* by the National Association of Corrosion Engineers; and the *ASM Metals Handbook.* Corrosion data, specifications, and recommended practices relating to stainless steels are also issued by ASTM. Stainless steels resist corrosion in a broad range of conditions, but they are not immune to every environment. For example, stainless steels perform poorly in reducing environments, such as 50% sulfuric and hydrochloric acids at elevated temperatures. The corrosive attack experienced is a breakdown of the protective film over the entire metal surface.

Such misapplications of stainless steels are rare and are usually avoided. The types of attack which are more likely to be of concern are pitting, crevice attack, stress corrosion cracking, and intergranular corrosion, which are discussed in Appendix A.

Table 6
Relative Corrosion Resistance of AISI Stainless Steels (1)

TYPE Number	UNS Number	Mild Atmospheric and Fresh Water	Atmospheric		Chemical		
			Industrial	Marine	Mild	Oxidizing	Reducing
201	(S20100)	x	x	x	x	x	
202	(S20200)	x	x	x	x	x	
205	(S20500)	x	x	x	x	x	
301	(S30100)	x	x	x	x	x	
302	(S30200)	x	x	x	x	x	
302B	(S30215)	x	x	x	x	x	
303	(S30300)	x	x		x		
303 Se	(S30323)	x	x		x		
304	(S30400)	x	x	x	x	x	
304L	(S30403)	x	x	x	x	x	
	(S30430)	x	x	x	x	x	
304N	(S30451)	x	x	x	x	x	
305	(S30500)	x	x	x	x	x	
308	(S30800)	x	x	x	x	x	
309	(S30900)	x	x	x	x	x	
309S	(S30908)	x	x	x	x	x	
310	(S31000)	x	x	x	x	x	
310S	(S31008)	x	x	x	x	x	
314	(S31400)	x	x	x	x	x	
316	(S31600)	x	x	x	x	x	x
316F	(S31620)	x	x	x	x	x	x
316L	(S31603)	x	x	x	x	x	x
316N	(S31651)	x	x	x	x	x	x
317	(S31700)	x	x	x	x	x	x
317L	(S31703)	x	x	x	x	x	
321	(S32100)	x	x	x	x	x	
329	(S32900)	x	x	x	x	x	x
330	(N08330)	x	x	x	x	x	x
347	(S34700)	x	x	x	x	x	
348	(S34800)	x	x	x	x	x	
384	(S38400)	x	x	x	x	x	
403	(S40300)	x			x		
405	(S40500)	x			x		
409	(S40900)	x			x		
410	(S41000)	x			x		
414	(S41400)	x			x		
416	(S41600)	x					
416 Se	(S41623)	x					
420	(S42000)	x					
420F	(S42020)	x					
422	(S42200)	x					
429	(S42900)	x	x		x	x	
430	(S43000)	x	x		x	x	
430F	(S43020)	x	x		x		
430F Se	(S43023)	x	x		x		
431	(S43100)	x	x	x	x		
434	(S43400)	x	x	x	x	x	
436	(S43600)	x	x	x	x	x	
440A	(S44002)	x			x		
440B	(S44003)	x					
440C	(S44004)	x					
442	(S44200)	x	x		x	x	
446	(S44600)	x	x	x	x	x	
	(S13800)	x	x		x	x	
	(S15500)	x	x	x	x	x	
	(S17400)	x	x	x	x	x	
	(S17700)	x	x	x	x	x	

- The "X" notations indicate that a specific stainless steel type may be considered as resistant to the corrosive environment categories.

  This list is suggested as a guideline only and does not suggest or imply a warranty on the part of the Specialty Steel Industry of the United States or any of the member companies. When selecting a stainless steel for any corrosive environment, it is always best to consult with a corrosion engineer and, if possible, conduct tests in the environment involved under actual operating conditions.

Table 7
Where Different Grades Are Used (15)

Environment	Grades
Acids	
Hydrochloric acid	Stainless generally is not recommended except when solutions are very dilute and at room temperature.
"Mixed acids"	There is usually no appreciable attack on Type 304 or 316 as long as sufficient nitric acid is present.
Nitric acid	Type 304L or 430 is used.
Phosphoric acid	Type 304 is satisfactory for storing cold phosphoric acid up to 85% and for handling concentrations up to 5% in some unit processes of manufacture. Type 316 is more resistant and is generally used for storing and manufacture if the fluorine content is not too high. Type 317 is somewhat more resistant than Type 316. At concentrations up to 85%, the metal temperature should not exceed 212 ºF (100 ºC) with Type 316 and slightly higher with Type 317. Oxidizing ions inhibit attack and other inhibitors such as arsenic may be added.
Sulfuric acid	Type 304 can be used at room temperature for concentrations over 80%. Type 316 can be used in contact with sulfuric acid up to 10% at temperatures up to 120 ºF (50 ºC) if the solutions are aerated; the attack is greater in airfree solutions. Type 317 may be used at temperatures as high as 150 ºF (65 ºC) with up to 5% concentration. The presence of other materials may markedly change the corrosion rate. As little as 500 to 2000 ppm of cupric ions make it possible to use Type 304 in hot solutions of moderate concentration. Other additives may have the opposite effect.
Sulfurous acid	Type 304 may be subject to pitting, particularly if some sulfuric acid is present. Type 316 is usable at moderate concentrations and temperatures.
Bases	
Ammonium hydroxide, sodium hydroxide, caustic solutions	Steels in the 300 series generally have good corrosion resistance at virtually all concentrations and temperatures in weak bases, such as ammonium hydroxide. In stronger bases, such as sodium hydroxide, there may be some attack, cracking or etching in more concentrated solutions and at higher temperatures. Commercial purity caustic solutions may contain chlorides, which will accentuate any attack and may cause pitting of Type 316 as well as Type 304.
Organics	
Acetic acid	Acetic acid is seldom pure in chemical plants but generally includes numerous and varied minor constituents. Type 304 is used for a wide variety of equipment including stills, base heaters, holding tanks, heat exchangers, pipelines, valves and pumps for concentrations up to 99% at temperatures up to about 120 ºF (50 ºC). Type 304 is also satisfactory for contact with 100% acetic acid vapors, and–if small amounts of turbidity or color pickup can be tolerated–for room temperature storage of glacial acetic acid. Types 316 and 317 have the broadest range of usefulness, especially if formic acid is also present or if solutions are unaerated. Type 316 is used for fractionating equipment, for 30 to 99% concentrations where Type 304 cannot be used, for storage vessels, pumps and process equipment handling glacial acetic acid, which would be discolored by Type 304. Type 316 is likewise applicable for parts having temperatures above 120 ºF (50 ºC), for dilute vapors and high pressures. Type 317 has somewhat greater corrosion resistance than Type 316 under severely corrosive conditions. None of the stainless steels has adequate corrosion resistance to glacial acetic acid at the boiling temperature or at superheated vapor temperatures.
Aldehydes	Type 304 is generally satisfactory.
Amines	Type 316 is usually preferred to Type 304.
Cellulose acetate	Type 304 is satisfactory for low temperatures, but Type 316 or Type 317 is needed for high temperatures.
Citric, formic and tartaric acids	Type 304 is generally acceptable at moderate temperatures, but Type 316 is resistant to all concentrations at temperatures up to boiling.
Esters	From the corrosion standpoint, esters are comparable with organic acids.
Fatty acids	Up to about 300 ºF (150 ºC), Type 304 is resistant to fats and fatty acids, but Type 316 is needed at 300 to 500 ºF (150 to 260 ºC) and Type 317 at higher temperatures.
Paint vehicles	Type 316 may be needed if exact color and lack of contamination are important.
Phthalic anhydride	Type 316 is usually used for reactors, fractionating columns, traps, baffles, caps and piping.
Soaps	Type 304 is used for parts such as spray towers, but Type 316 may be preferred for spray nozzles and flake-drying belts to minimize offcolor product.
Synthetic detergents	Type 316 is used for preheat, piping, pumps and reactors in catalytic hydrogenation of fatty acids to give salts of sulfonated high molecular alcohols.
Tall oil (pulp and paper industry)	Type 304 has only limited usage in tall-oil distillation service. High-rosin-acid streams can be handled by Type 316L with a minimum molybdenum content of 2.75%. Type 316 can also be used in the more corrosive high-fatty-acid streams at temperatures up to 475 ºF (245 ºC), but Type 317 will probably be required at higher temperatures.
Tar	Tar distillation equipment is almost all Type 316 because coal tar has a high chloride content; Type 304 does not have adequate resistance to pitting.
Urea	Type 316L is generally required.
Pharmaceuticals	Type 316 is usually selected for all parts in contact with the product because of its inherent corrosion resistance and greater assurance of product purity.

Table 8
AUSTENITIC STAINLESS STEELS

Type	C	Mn	P	S	Si	Cr	Ni	Mo	Other	Tensile Strength ksi	Tensile Strength MPa	Yield Strength (0.2% offset) ksi	Yield Strength (0.2% offset) MPa	Elongation in 2" (50.80mm) %	Hardness (Rockwell)	Product Form
	Chemical Analysis % (Max. unless noted otherwise)									Nominal Mechanical Properties (Annealed Sheet unless noted otherwise)						
201	0.15	5.50/7.50	0.060	0.030	1.00	16.00/18.00	3.50/5.50		0.25N	95	655	45	310	40	B90	
202	0.15	7.50/10.00	0.060	0.030	1.00	17.00/19.00	4.00/6.00		0.25N	90	612	45	310	40	690	
205	0.12/0.25	14.00/15.50	0.030	0.030	0.50	16.50/18.00	1.00/1.75		0.32/0.40N	120.5	831	69	476	58	B98	(Plate)
301	0.15	2.00	0.045	0.030	1.00	16.00/18.00	6.00/8.00			110	758	40	276	60	B85	
302	0.15	2.00	0.045	0 030	1 00	17.00/19.00	8.00/10.00			90	612	40	276	50	B85	
302B	0.15	2.00	0.045	0.030	2.00/3.00	17.00/19.00	8.00/10.00			95	655	40	276	55	B85	
303	0.15	2.00	0.20	0.15 (min)	1.00	17.00/19.00	8.00/10.00	0.60*		90	621	35	241	50		(Bar)
303Se	0.15	2.00	0.20	0.060	1.00	17.00/19.00	8.00/10.00		0.15Se (min)	90	621	35	241	50		(Bar)
304	0.08	2.00	0.045	0.030	1.00	18.00/20.00	8.00/10.50			84	579	42	290	55	B80	
304L	0.030	2.00	0.045	0.030	1.00	18.00/20.00	8.00/12.00			81	558	39	269	55	B79	
S30430	0.08	2.00	0.045	0.030	1.00	17.00/19.00	8.00/10.00		3.00/4.00Cu	73	503	31	214	70	B70	(Wire)
304N	0.08	2.00	0.045	0.030	1.00	18.00/20.00	8.00/10.50		0.10/0.16N	90	621	48	331	50	B85	
305	0.12	2.00	0.045	0.030	1.00	17.00/19.00	10.50/13.00			85	586	38	262	50	B80	
308	0.08	2.00	0.045	0.030	1.00	19.00/21.00	10.00/12.00			115	793	80	552	40		(Wire)
309	0.20	2.00	0.045	0.030	1.00	22.00/24.00	12.00/15.00			90	621	45	310	45	B85	
309S	0.08	2.00	0.045	0.030	1.00	22.00/24.00	12.00/15.00			90	621	45	310	45	B85	
310	0.25	2.00	0.045	0.030	1.50	24.00/26.00	19.00/22.00			95	655	45	310	45	B85	
310S	0.08	2.00	0.045	0.030	1.50	24.00/26.00	19.00/22.00			95	655	45	310	45	B85	
314	0.25	2.00	0.045	0.030	1.50/3.00	23.00/26.00	19.00/22.00			100	689	50	345	40	B85	
316	0.08	2.00	0.045	0.030	1.00	16.00/18.00	10.00/14.00	2.00/3.00		84	579	42	290	50	B79	
316F	0.08	2.00	0.20	0.10min	1.00	16.00/18.00	10.00/14.00	1.75/2.50		85	586	38	262	60	B85	
316L	0.030	2.00	0.045	0.030	1.00	16.00/18.00	10.00/14.00	2.00/3.00		81	558	42	290	50	B79	
316N	0.08	2.00	0.045	0.030	1.00	16.00/18.00	10.00/14.00	2.00/3.00	0.10/0.16N	90	621	48	331	48	B85	
317	0.08	2.00	0.045	0.030	1.00	18.00/20.00	11.00/15.00	3.00/4.00		90	621	40	276	45	B85	
317L	0.030	2.00	0.045	0.030	1.00	18.00/20.00	11.00/15.00	3.00/4.00		86	593	38	262	55	B85	
317LMN	0.030	2.00	0.045	0.030	0.75	17.00/20.00	13.50/17.50	4.00/5.00	0.10/0.20N	96	662	54	373	49	B88	
321	0.08	2.00	0.045	0.030	1.00	17.00/19.00	9.00/12.00		5xC Ti (min)	90	621	35	241	45	B80	
330	0.08	2.00	0.040	0.030	0.75/1.50	17.00/20.00	34.00/37.00		0.10Ta 0.20Cb	80	552	38	262	40	B80	
347	0.08	2.00	0.045	0.030	1.00	17.00/19.00	9.00/13.00		10xC Cb (min)	95	655	40	276	45	B85	
348	0.08	2.00	0.045	0.030	1.00	17.00/19.00	9.00/13.00		Cb + Ta 10xC (min) Ta 0.10 max Cc 0.20 max	95	655	40	276	45	B85	
384	0.08	2.00	0.045	0.030	1.00	15.00/17.00	17.00/19.00			75	517	35	241	55	B70	(Wire)

* May be added at manufacturer's option.

II

MECHANICAL AND PHYSICAL PROPERTIES (Room Temperature)

Austenitic Stainless Steels

The austenitic stainless steels cannot be hardened by heat treatment but can be strengthened by cold work, and thus they exhibit a wide range of mechanical properties. At room temperature, austenitic stainless steels exhibit yield strengths between 30 and 200 ksi (207-1379 MPa), depending on composition and amount of cold work. They also exhibit good ductility and toughness even at high strengths, and this good ductility and toughness is retained at cryogenic temperatures. The chemical compositions and nominal mechanical properties of annealed austenitic stainless steels are given in Table 8.

II

The difference in effect of cold work of Types 301 and 304 is indicated by the stress strain diagrams in Figure 11.

Carbon and nitrogen contents affect yield strength, as shown by the differences among Types 304, 304L, and 304N. The effect of manganese and nitrogen on strength can be seen by comparing Types 301 and 302 against Types 201 and 202.

Figures 12, 13, 14, and 15 illustrate other effects of small composition changes. For example, at a given amount of cold work, Types 202 and 301 exhibit higher yield and tensile strengths than Types 305 and 310.

Austenitic stainless steels which can be cold worked to high tensile and yield strengths, while retaining good ductility and toughness, meet a wide range of design criteria. For example, sheet and strip of austenitic steels – usually Types 301 and 201 – are produced in the following tempers:

Temper	Tensile Strength Minimum ksi	Tensile Strength Minimum MPa	Yield Strength Minimum ksi	Yield Strength Minimum MPa
¼-Hard	125	862	75	517
½-Hard	150	1034	110	758
¾-Hard	175	1207	135	931
Full-Hard	185	1276	140	965

In structural applications, the toughness and fatigue strength of these steels are important. At room temperature in the annealed condition, the austenitic steels exhibit Charpy V-notch energy absorption values in excess of 100 ft.-lb. The effect of cold rolling Type 301 on toughness is illustrated in Figure 16. This shows Type 301 to have good toughness even after cold rolling to high tensile strengths.

Fatigue or endurance limits (in bending) of austenitic stainless steels in the annealed condition shown in Table 9 are about one-half the tensile strength.

New Design Specification

Until recently, design engineers wanting to use austenitic stainless steels structurally had to improvise due to the lack of an appropriate design specification. The familiar American Institute for Steel Construction and AISI design specifications for carbon steel design do not apply to the design of stainless steel members because of differences in strength properties, modulus of elasticity, and the shape of the stress strain curve. Figure 17 shows that there is no well-defined yield point for stainless steel.

Table 9
TYPICAL ENDURANCE LIMITS OF ANNEALED CHROMIUM-NICKEL STAINLESS STEEL SHEET (2)

AISI Type	Endurance limit, ksi	MPa
301	35	241
302	34	234
303	35	241
304	35	241
316	39	269
321	38	262
347	39	269

Now the American Society of Civil Engineers (ASCE), in conjunction with the SSINA, has prepared a standard (ANSI/ASCE-8-90) "Specification for the Design of Cold-Formed Stainless Steel Structural Members." This standard covers four types of austenitic stainless steel, specificially Types 201, 301, 304 and 316, and three types of ferritic stainless steels (See Ferritic section below). This standard requires the use of structural quality stainless steel as defined in general by the provisions of the American Society for Testing and Materials (ASTM) specifications.

Some of the physical properties of austenitic stainless steels are similar to those of the martensitic and ferritic stainless steels. The modulus of elasticity, for example, is 28 x 10^6 psi (193 GPa) and density is 0.29 lb. per cu. in. (8060 Kg/m^3). The physical properties of annealed Type 304 are shown in Table 10.

Ferritic Stainless Steels

Ferritic stainless steels contain approximately 12% chromium (and up). The chemical composition of the standard grades are shown in Table 11 along with nominal mechanical properties. Also several proprietary grades (see Appendix A) have achieved relatively wide commercial acceptance.

Three ferritic stainless steels, namely Types 409, 430 and 439 are included in the ASCE "Specification for the Design of Cold-Formed Stainless Steel Structural Members." Designers should be aware of two notations in this specification:

(1) The maximum thickness for Type 409 ferritic stainless used in the standard is limited to 0.15 inches.

(2) The maximum thickness for Type 430 and 439 ferritic stainless steels is limited to 0.125 inches.

This is in recognition of concerns for the ductile to brittle transition temperature of the ferritic stainless steels in structural application. It should be noted that these alloys have been used in plate thickness for other applications.

Generally, toughness in the annealed condition decreases as the chromium content increases. Molybdenum tends to increase ductility, whereas carbon tends to decrease ductility. Ferritic stainless steels can be used for structural applications (as noted above), as well as such traditional applications as kitchen sinks, and automotive, appliance, and luggage trim, which require good resistance to corrosion and bright, highly polished finishes.

When compared to low-carbon steels, such as SAE 1010, the standard numbered AISI ferritic stainless steels, (such as Type 430) exhibit somewhat higher yield and tensile strengths, and low elongations. Thus, they are not as formable as the low-carbon steels. The proprietary ferritic stainless steels, on the other hand, with lower carbon levels have improved ductility and formability comparable with that of low-carbon steels. Because of the higher strength levels, the ferritic stainless steels require slightly more power to form.

Micro cleanliness is important to good formability of the ferritic types because inclusions can act as initiation sites for cracks during forming.

Type 405 stainless is used where the annealed mechanical properties and corrosion resistance of Type 410 are satisfactory but when better weldability is desired. Type 430 is used for formed products, such as sinks and decorative trim. Physical properties of Type 430 are shown in Table 10. Types 434 and 436 are used when better corrosion resistance is required and for relatively severe stretching.

For fasteners and other machined parts, Types 430F and 430F Se are often used, the latter being specified when forming is required in addition to machining.

Types 442 and 446 are heat resisting grades.

Type 409, which has the lowest chromium content of the stainless steels, is widely used for automotive exhaust systems.

Table 10 PHYSICAL PROPERTIES OF GENERAL-PURPOSE STAINLESS STEELS (ANNEALED) (1)	Type 304	Type 430	Type 410	S13800
Modulus of Elasticity in Tension psi x 10^6 (GPa)	28.0 (193)	29.0 (200)	29.0 (200)	29.4 (203)
Modulus of Elasticity in Torsion psi x 10^6 (GPa)	12.5 (86.2)	– –	– –	– –
Density, lbs/in^3 (kg/m^3)	0.29 (8060)	0.28 (7780)	0.28 (7780)	0.28 (7780)
Specific Heat, Btu/lb/F (J/kg•K) 32-212F (0-100ºC)	0.12 (503)	0.11 (460)	0.11 (460)	0.11 (460)
Thermal Conductivity, Btu/hr/ft/F (W/m•K)				
212ºF (100ºC)	9.4 (0.113)	15.1 (0.182)	14.4 (0.174)	8.1 (0.097)
932ºF (500ºC)	12.4 (0.149)	15.2 (0.183)	16.6 (0.201)	12.7 (0.152)
Mean Coefficient of Thermal Expansion x10^{-6}/F (x10^{-6}/ºC)				
32-212ºF (0-100ºC)	9.6 (17.3)	5.8 (10.4)	5.5 (9.9)	5.9 (10.6)
32-600ºF (0-315ºC)	9.9 (17.9)	6.1 (11.0)	6.3 (11.4)	6.2 (11.2)
32-1000ºF (0-538ºC)	10.2 (18.4)	6.3 (11.4)	6.4 (11.6)	6.6 (11.9)
32-1200ºF (0-648ºC)	10.4 (18.8)	6.6 (11.9)	6.5 (11.7)	– –
32-1800ºF (0.982ºC)	– –	6.9 (12.4) (32-1500ºF)	– –	– –
Melting Point Range ºF (ºC)	2550 to 2650 (1398 to 1454)	2600 to 2750 (1427 to 1510)	2700 to 2790 (1483 to 1532)	2560 to 2625 (1404 to 1440)

II

Table 11 FERRITIC STAINLESS STEELS (1)
Chemical Analysis % (Max. unless noted otherwise)

Type	C	Mn	P	S	Si	Cr	Ni	Mo	Other
405	0.08	1.00	0.040	0.030	1.00	11.50/14.50	0.60		0.10/0.30 Al
409	0.08	1.00	0.045	0.045	1.00	10.50/11.75	0.50		6xC/0.75 Ti
429	0.12	1.00	0.040	0.030	1.00	14.00/16.00	0.75		
430	0.12	1.00	0.040	0.030	1.00	16.00/18.00	0.75		
430F	0.12	1.25	0.060	0.15 (min)	1.00	16.00/18.00		0.60*	
430F Se	0.12	1.25	0.060	0.060	1.00	16.00/18.00			0.15 Se (min.)
434	0.12	1.00	0.040	0.030	1.00	16.00/18.00		0.75/1.25	
436	0.12	1.00	0.040	0.030	1.00	16.00/18.00		0.75/1.25	5xC/0.70Cb+Ta
442	0.20	1.00	0.040	0.030	1.00	18.00/23.00	0.60		
446	0.20	1.50	0.040	0.030	1.00	23.00/27.00	0.75		0.25N

* May be added at manufacturer's option.

Nominal Mechanical Properties
(Annealed sheet unless noted otherwise)

Type	Tensile Strength		Yield Strength (to .2% offset)		Elongation in 2" (50.80 mm) %	Hardness (Rockwell)	Product Form
	ksi	MPa	ksi	MPa			
405	65	448	40	276	25	B75	
409	65	448	35	241	25	B75	
429	70	483	40	276	30	B80	(Plate)
430	75	517	50	345	25	B85	
430F	95	655	85	586	10	B92	
430F Se	95	655	85	586	10	B92	(Wire)
434	77	531	53	365	23	B83	
436	77	531	53	365	23	B83	
442	80	552	45	310	20	B90	(Bar)
446	80	552	50	345	20	B83	

Martensitic Stainless Steels

The martensitic grades are so named because when heated above their critical temperature (1600ºF or 870ºC) and cooled rapidly, a metallurgical structure known as martensite is obtained. In the hardened condition the steel has very high strength and hardness, but to obtain optimum corrosion resistance, ductility, and impact strength, the steel is given a stress-relieving or tempering treatment (usually in the range 300-700ºF (149-371ºC)).

Tables 12, 13 and 14 give the chemical compositions and mechanical properties of martensitic grades in the annealed and hardened conditions.

II

The martensitic stainless steels fall into two main groups that are associated with two ranges of mechanical properties: low-carbon compositions with a maximum hardness of about Rockwell C45 and the higher-carbon compositions, which can be hardened up to Rockwell C60. (The maximum hardness of both groups in the annealed condition is about Rockwell C24.) The dividing line between the two groups is a carbon content of approximately 0.15%.

In the low-carbon class are Types 410, 416 (a free-machining grade) and 403 (a ''turbine-quality'' grade). The properties, performance, heat treatment, and fabrication of these three stainless steels are similar except for the better machinability of Type 416.

On the high-carbon side are Types 440A, B, and C.

Types 420, 414, and 431, however, do not fit into either category. Type 420 has a minimum carbon content of 0.15% and is usually produced to a carbon specification of 0.3-0.4%. While it will not harden to such high values as the 440 types, it can be tempered without substantial loss in corrosion resistance. Hence, a combination of hardness and adequate ductility (suitable for cutlery or plastic molds) is attained.

Types 414 and 431 contain 1.25–2.50% nickel, which is enough to increase hardenability, but not enough to make them austenitic at ambient temperature. The addition of nickel serves two purposes: (1) it improves corrosion resistance because it permits a higher chromium content, and (2) it enhances toughness.

Martensitic stainless steels are subject to temper embrittlement and should not be heat treated or used in the range of 800 to 1050ºF (427-566ºC) if toughness is important. The effect of tempering in this range is shown by the graph in Figure 18. Tempering is usually performed above this temperature range.

Impact tests on martensitic grades show that toughness tends to decrease with increasing hardness. High-strength (high-carbon) Type 440A exhibits lower toughness than Type 410. Nickel increases toughness, and Type 414 has a higher level of toughness than Type 410 at the same strength level.

Martensitic grades exhibit a ductile-brittle transition temperature at which notch ductility drops very suddenly. The transition temperature is near room temperature, and at low temperature about -300ºF (-184ºC) they become very brittle, as shown by the data in Figure 19. This effect depends on composition, heat treatment, and other variables.

Clearly, if notch ductility is critical at room temperature or below, and the steel is to be used in the hardened condition, careful evaluation is required. If the material is to be used much below room temperature, the chances are that quenched-and-tempered Type 410 will not be satisfactory. While its notch ductility is better in the annealed condition down to -100ºF (-73ºC), another type of stainless steel is probably more appropriate.

The fatigue properties of the martensitic stainless steels depend on heat treatment and design. A notch in a structure or the effect of a corrosive environment can do more to reduce fatigue limit than alloy content or heat treatment.

Figure 20 gives fatigue data for Type 403 turbine quality stainless at three test temperatures. The samples were smooth and polished, and the atmosphere was air.

Another important property is abrasion or wear resistance. Generally, the harder the material, the more resistance to abrasion it exhibits. In applications where corrosion occurs, however, such as in coal handling operations, this general rule may not hold, because the oxide film is continuously removed, resulting in a high apparent abrasion/corrosion rate.

Other mechanical properties of martensitic stainless steels, such as compressive yield shear strength, are generally similar to those of carbon and alloy steels at the same strength level.

Room-temperature physical properties of Type 410 are shown in Table 10. The property of most interest is modulus of elasticity. The moduli of the martensitic stainless steels (29×10^6 psi) (200 GPa) are slightly less than the modulus of carbon steel (30×10^6 psi) (207 GPa) but are markedly higher than the moduli of other engineering materials, such as aluminum (10×10^6 psi) (67 GPa).

The densities of the martensitic stainless steels (about 0.28 lb. per cu. in.) (7780 Kg/m^3) are slightly lower than those of the carbon and alloy steels. As a result, they have excellent vibration damping capacity.

The martensitic stainless steels are generally selected for moderate resistance to corrosion, relatively high strength, and good fatigue properties after suitable heat treatment. Type 410 is used for fasteners, machinery parts and press plates. If greater hardenability or higher toughness is required, Type 414 may be used, and for better machinability, Types 416 or 416 Se are used. Springs, flatware, knife blades, and hand tools are often made from Type 420, while Type 431 is frequently used for aircraft parts requiring high yield strength and resistance to shock. Cutlery consumes most of Types 440A and B, whereas Type 440C is frequently used for valve parts requiring good wear resistance.

High-carbon martensitic stainless steels are generally not recommended for welded applications, although Type 410 can be welded with relative ease. Hardening heat treatments should follow forming operations because of the poor forming qualities of the hardened steels.

Table 12 MARTENSITIC STAINLESS STEELS (1) Chemical Analysis % (Max. unless noted otherwise)									
Type	C	Mn	P	S	Si	Cr	Ni	Mo	Other
403	0.15	1.00	0.040	0.030	0.50	11.50/13.00			
410	0.15	1.00	0.040	0.030	1.00	11.50/13.50			
414	0.15	1.00	0.040	0.030	1.00	11.50/13.50	1.25/2.50		
416	0.15	1.25	0.060	0.15 (Min)	1.00	12.00/14.00		0.60*	
416 Se	0.15	1.25	0.060	0.060	1.00	12.00/14.00			0.15 Se (Min.)
420	0.15 (Min.)	1.00	0.040	0.030	1.00	12.00/14.00			
420 F	0.15 (Min.)	1.25	0.060	0.15 (Min.)	1.00	12.00/14.00		0.60*	
422	0.20/0.25	1.00	0.025	0.025	0.75	11.00/13.00	0.50/1.00	0.75/1.25	0.15/0.30 V 0.75/1.25 W
431	0.20	1.00	0.040	0.030	1.00	15.00/17.00	1.25/2.50		
440A	0.60/0.75	1.00	0.040	0.030	1.00	16.00/18.00		0.75	
440B	0.75/0.95	1.00	0.040	0.030	1.00	16.00/18.00		0.75	
440C	0.95/1.20	1.00	0.040	0.030	1.00	16.00/18.00		0.75	

*May be added at manufacturer's option

Table 13 Nominal Mechanical Properties (Annealed sheet unless noted otherwise)							
Type	Tensile Strength		Yield Strength (0.2% offset)		Elongation in 2" (50.80 mm)	Hardness (Rockwell)	Product Form
	ksi	MPa	ksi	MPa	%		
403	70	483	45	310	25	B80	
410	70	483	45	310	25	B80	
414	120	827	105	724	15	B98	
416	75	517	40	276	30	B82	(Bar)
416Se	75	517	40	276	30	B82	(Bar)
420	95	655	50	345	25	B92	(Bar)
420F	95	655	55	379	22	220 (Brinell)	(Bar)
422*	145	1000	125	862	18	320 (Brinell)	(Bar)
431	125	862	95	655	20	C24	(Bar)
440A	105	724	60	414	20	B95	(Bar)
440B	107	738	62	427	18	B96	(Bar)
440C	110	758	65	448	14	B97	(Bar)

*Hardened and Tempered

II

Precipitation Hardening Stainless Steels

The principle of precipitation hardening is that a supercooled solid solution (solution annealed material) changes its metallurgical structure on aging. The principal advantage is that products can be fabricated in the annealed condition and then strengthened by a relatively low-temperature 900-1150ºF (482-620ºC) treatment, minimizing the problems associated with high-temperature treatments. Strength levels of up to 260 ksi (1793 MPa) (tensile) can be achieved – exceeding even those of the martensitic stainless steels – while corrosion resistance is usually superior – nearly equal to that of Type 304 stainless. Ductility is similar to corresponding martensitic grades at the same strength level. Table 15 shows the chemical composition and the nominal mechanical properties of four AISI standard precipitation hardening stainless steels in solution treated and age hardened conditions.

Precipitation hardening stainless steels have high strength, relatively good ductility, and good corrosion resistance at moderate temperatures. They are utilized for aerospace structural components, fuel tanks, landing gear covers, pump parts, shafting, bolts, saws, knives, and flexible bellows-type expansion joints.

Physical properties of UNS S13800 are shown in Table 10.

II

Table 14
NOMINAL MECHANICAL PROPERTIES
As Quenched Hardness and Properties After Hardening and Tempering 1 in. (25.4 mm) Diameter Bars

Type	UNS	Hardening Temp. °F (°C)	As Quenched Hardness HB	As Quenched Hardness HR	Tempering Temp. °F (°C)	Tensile Strength, ksi (MPa)	Yield Str. 0.2% Offset ksi (MPa)	Elon. in. 2 in. (50.80 mm) %	Red. of Area %	Izod Impact V-Notch Ft. Lbs. (J)	Tempered Hardness HB	Tempered Hardness HR
403 and 410	S40300 S41000	1800 (981)	410	C43	400 (204)	190 (1310)	145 (1000)	15	55	35 (47)	390	C41
					600 (315)	180 (1241)	140 (965)	15	55	35 (47)	375	C39
					800* (426)	195 (1344)	150 (1034)	17	55		390	C41
					1000* (538)	145 (1000)	115 (793)	20	65		300	C31
					1200 (648)	110 (758)	85 (586)	23	65	75 (102)	225	B97
					1400 (760)	90 (621)	60 (414)	30	70	100 (136)	180	B89
416 and 416 Se	S41600 S41623	1800 (981)	410	C43	400 (204)	190 (1310)	145 (1000)	12	45	20 (27)	390	C41
					600 (315)	180 (1241)	140 (965)	13	45	20 (27)	375	C39
					800* (426)	195 (1344)	150 (1034)	13	50		390	C41
					1000* (538)	145 (1000)	115 (793)	15	50		300	C31
					1200 (648)	110 (758)	85 (586)	18	55	30 (41)	225	B97
					1400 (760)	90 (621)	60 (414)	25	60	60 (81)	180	B89
414	S41400	1800 (981)	425	C44	400 (204)	100 (1379)	150 (1034)	15	55	45 (61)	410	C43
					600 (315)	190 (1310)	145 (1000)	15	55	45 (61)	400	C41
					800 (426)	200 (1379)	150 (1034)	16	58		415	C43
					1000* (538)	145 (1000)	120 (827)	20	60		290	C30
					1200 (760)	120 (827)	105 (724)	20	65	50 (68)	250	C22
431	S43100	1900 (1036)	440	C45	400 (204)	205 (1413)	155 (1069)	15	55	30 (41)	415	C43
					600 (315)	195 (1344)	150 (1034)	15	55	45 (61)	400	C41
					800* (426)	205 (1413)	155 (1069)	15	60		415	C43
					1000* (538)	150 (1034)	130 (896)	18	60		325	C34
					1200 (760)	125 (862)	95 (655)	20	60	50 (68)	260	C24
420	S42000	1900 (1036)	540	C54	600 (315)	230 (1586)	195 (1344)	8	25	10 (14)	500	C50
440A	S44002	1900 (1036)	570	C56	600 (315)	260 (1793)	240 (1655)	5	20	4 (5)	510	C51
440B	S44003	1900 (1036)	590	C58	600 (315)	280 (1931)	270 (1862)	3	15	3 (4)	555	C55
440C	S44004	1900 (1036)	610	C60	600 (315)	285 (1965)	275 (1896)	2	10	2 (3)	580	C57

*Tempering within the range of 750 to 1050 ºF (399 to 565 ºC) is not recommended because such treatment will result in low and erratic impact properties and loss of corrosion resistance. Note. Variations in chemical composition within the individual type ranges may affect the mechanical properties.

Table 15
PRECIPITATION HARDENING STAINLESS STEELS (1)
Chemical Analysis % (Max. unless noted otherwise)

Type	C	Mn	P	S	Si	Cr	Ni	Mo	Other
S13800	0.05	0.10	0.010	0.008	0.10	12.25/13.25	7.50/8.50	2.00/2.50	0.90/1.35 Al 0.010 N
S15500	0.07	1.00	0.040	0.030	1.00	14.00/15.50	3.50/5.50		2.50/4.50 Cu 0.15/0.45 Cb + Ta
S17400	0.07	1.00	0.040	0.030	1.00	15.50/17.50	3.00/5.00		3.00/5.00 Cu 0.15/0.45 Cb + Ta
S17700	0.09	1.00	0.040	0.040	0.040	16.00/18.00	6.50/7.75		0.75/1.50 Al

Nominal Mechanical Properties
(Solution Treated Bar)

Type	Tensile Strength ksi	Tensile Strength MPa	Yield Strength (0.2%offset) ksi	Yield Strength (0.2%offset) MPa	Elongation in 2" (50.80 mm) %	Hardness (Rockwell)
S13800	160	1103	120	827	17	C33
S15500	160	1103	145	1000	15	C35
S17400	160	1103	145	1000	15	C35
S17700	130	896	40	276	10	B90

HIGH TEMPERATURE MECHANICAL PROPERTIES

Stainless steels are used at temperatures up to about 2000ºF (1093ºC) because they have good strength at elevated temperature and good resistance to corrosion and oxidation.

In steam power generation, for example, high allowable design stresses permit the use of thin sections and high operating temperatures. In aircraft and spacecraft design, the AISI numbered stainless steels are used for parts in which hot strength is crucial. Stainless steels are used extensively in heat exchangers in which there is need for both corrosion resistance and hot strength, especially for pressure service. The nuclear power industry represents many high-temperature applications for stainless steels, such as superheaters, boilers, feed-water heaters, valves, and main steam lines.

At steam temperatures over 1050ºF (566ºC), the austenitic stainless steels are preferred. This is illustrated by Table 16, which shows allowable stresses for Type 304 seamless pipe in unfired vessels, as compared with a low-alloy chromium-molybdenum steel.

In analyzing high-temperature properties, hot strength and thermal stability (from the standpoint of softening or embrittlement) are important. Physical properties are also significant.

Figure 21 gives a broad concept of the hot-strength advantages of stainless steels in comparison to low-carbon unalloyed steel. Precipitation hardening stainless steels also have excellent hot strength at moderate temperatures, but their strength declines sharply as they overage at high temperature.

Figure 22 compares the effect of temperature on the strength and ductility of annealed vs. cold worked Type 301. Above 1000ºF (537ºC), design will be based on creep or rupture strength.

Figure 23 shows short-time tensile and yield strengths of various stainless steels.

Table 17 shows creep and rupture strengths of annealed 400 Series stainless steels exposed to temperatures up to 1500ºF (816ºC), while Table 18 shows data for five stainless steels at 1600-2000ºF (871-1093ºC). Figures 24, 25, and 26 show comparative 100,000 hour stress-rupture data for Types 304, 321, and 347, respectively.

These data generally apply to stainless steels normally furnished by mills, but it should be recognized that processing variables can occur. To minimize such variables in materials for high-temperature service, ASTM has established an "H" modification of some AISI numbered stainless steels. This modification establishes a minimum carbon level in grades such as 304H and 321H when the intended application requires good high temperature properties.

Welding can affect high-temperature rupture and creep strength characteristics. Nevertheless, good welding practices result in reliable values.

Pressure vessels and other elevated-temperature equipment are designed to American Society of Mechanical Engineers Boiler and Pressure Vessel Codes. These represent an excellent compendium of minimum requirements for design, fabrication, inspection and construction. Designers should refer to the latest applicable revisions that reflect current technology.

THERMAL STABILITY

With time and temperature, changes in metallurgical structure can be expected for almost any steel or alloy. In stainless steels, the changes can be softening, carbide precipitation, or embrittlement.

Softening occurs in the martensitic stainless steels when exposed to temperatures approaching or exceeding the original tempering temperature. Type 440C, for example, can be held at 900ºF (482ºC) for only short periods if the high hardness is to be retained. Cold-worked austenitic stainless steels, as shown previously in Figure 22, may also soften at elevated temperature.

Embrittlement usually means the loss of room-temperature toughness. Embrittled equipment must be handled carefully to avoid impact when it is cooled down for maintenance. Table 19 shows how prolonged holding at temperatures of 900 to 1200ºF (482-649ºC) can affect room-temperature toughness of various stainless steels, while Figure 27 puts embrittlement in better perspective with respect to the three "general-purpose" types. Note that the transition temperature for Types 410 and 430 is near room temperature, while there is only minor loss of toughness in Type 304. Embrittlement is rarely of concern with austenitic stainless steels.

Ferritic and duplex stainless steels are subject to embrittlement when exposed to temperatures of 700-950ºF over an extended period of time. Martensitic grades with greater than 12% chromium also have been known to display brittle tendencies after extended periods in the same temperature range. This phenomenon is called 885F embrittlement because of the temperature at which embrittlement is most pronounced.

885F embrittlement results in low ductility and increased hardness and tensile strengths at room temperature, and the metal may fracture catastrophically if not handled carefully. The metal, however, retains its desirable mechanical properties at operating temperature (500ºF and higher). The effect of 885F embrittlement can be removed by heat treatment at 1100ºF (593ºC).

Ferritic and duplex grades are subject to sigma-phase embrittlement when exposed to temperatures of 1000-1800ºF (538-987ºC) over extended periods, which also results in loss of room-temperature ductility and corrosion resistance. Sigmaphase can be removed by heat treatment for ferritic grades at 1650ºF (899ºC) followed by air cooling, and at 1900ºF (1038ºC), and higher for duplex grades. II

Carbide precipitation occurs (see section on corrosion) in austenitic stainless steels in the temperature range of 800-1600ºF (427-871ºC). This causes a loss of toughness and may make the steel subject to intergranular corrosion in certain environments. It can be removed by heat treatment above 1900ºF (1038ºC).

Brittle failure under load is of concern, especially in welded fabrications. This type of embrittlement is largely a problem at temperatures of 1000-1500ºF (538-816ºC), since strength and not ductility is the limiting factor at higher temperatures. Because of difficulty in evaluating data, and the variable conditions involved, designers are encouraged to seek technical assistance from stainless steel producers.

Physical properties such as linear expansion and thermal conductivity are of interest. Figure 28 shows austenitic stainless steels to have greater thermal expansion than martensitic or ferritic grades. This should be considered when joining dissimilar metals.

Thermal conductivity is also different among different stainless steels. However, in heat exchange applications, film resistance, fouling, and other surface factors have a far greater effect on heat transfer than the alloy itself.

Fluctuating thermal stresses, resulting from periodic changes in temperature, can lead to fatigue problems. A rule of thumb has been used with apparent success: For a 20-year life, a figure of 7000 cycles – corresponding to one cycle a day – has been used in piping design, while 40,000 is the number of temperature swings for process equipment over the same period. The 300 Series stainless steels are more sensitive to thermal fatigue than the 400 Series types.

Table 16
ALLOWABLE STRESS AT MAXIMUM METAL TEMPERATURE (2)

ºF / ºC	900 / 482		950 / 510		1000 / 538		1050 / 566		1100 / 593		1150 / 621		1200 / 649	
Type	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa
304	10.0	68.9	9.8	67.2	9.5	65.2	9.0	62.1	8.3	56.9	6.9	47.6	5.5	37.9
2¼Cr-1 Mo	13.1	90.3	11.0	75.8	7.8	53.8	5.8	40.0	4.2	29.0	3.0	20.7	2.0	13.8

Table 17
RUPTURE AND CREEP CHARACTERISTICS OF
CHROMIUM STAINLESS STEELS IN ANNEALED CONDITION (2)

ºF / ºC	800 / 427		900 / 482		1000 / 538		1100 / 593		1200 / 649		1300 / 704		1400 / 760		1500 / 861	
Type	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa
Stress for rupture in 1000 hours																
405	–	–	25.0	172	16.0	110	6.8	47	3.8	27	2.2	15	1.2	8	0.8	6
410	54.0	372	34.0	234	19.0	131	10.0	69	4.9	34	2.5	18	1.2	8	–	–
430	–	–	30.0	207	17.5	120	9.1	63	5.0	34	2.8	2.0	1.7	12	0.9	6
446	–	–	–	–	17.9	123	5.6	39	4.0	28	2.7	19	1.8	13	1.2	8
Stress for rupture in 10,000 hours																
405	–	–	22.0	152	12.0	83	4.7	33	2.5	18	1.4	10	0.7	5	0.4	3
410	42.5	294	26.0	179	13.0	90	6.9	47	3.5	24	1.5	10	0.6	4	–	–
430	–	–	24.0	165	13.5	94	6.5	43	3.4	23	2.2	15	0.7	5	0.5	3
446	–	–	–	–	13.5	94	3.0	21	2.2	15	1.6	11	1.1	8	0.8	6
Stress for creep rate of 0.0001 % per hour																
405	–	–	43.0	296	8.0	55	2.0	14	–	–	–	–	–	–	–	–
410	43.0	296	29.0	200	9.2	63	4.2	29	2.0	14	1.0	7	0.8	6	–	–
430	23.0	159	15.4	106	8.6	59	4.3	30	1.2	8	1.4	10	0.9	6	0.6	4
446	31.0	214	16.4	113	6.1	42	2.8	20	1.4	10	0.7	5	0.3	2	0.1	1
Stress for creep rate of 0.00001 % per hour																
405	–	–	14.0	97	4.5	32	0.5	3	–	–	–	–	–	–	–	–
410	19.5	135	13.8	96	7.2	49	3.4	24	1.2	8	0.6	4	0.4	3	–	–
430	17.5	120	12.0	83	6.7	46	3.4	24	1.5	10	0.9	6	0.6	4	0.3	2
446	27.0	186	13.0	90	4.5	32	1.8	13	0.8	6	0.3	2	0.1	1	0.05	0.3

Table 18
RUPTURE AND CREEP CHARACTERISTICS OF
CHROMIUM-NICKEL STAINLESS STEELS (17)

	Testing temperature		Stress								Extrapolated elogation at rupture in 10,000 hr, %
			Rupture Time						Creep Rate		
			100 hr		1000 hr		10,000 hr		0.01 % hr.		
Type	°F	°C	ksi	MPa	ksi	MPa	ksi	MPa	ksi	MPa	
302	1600	871	4.70	33	2.80	19	1.75	12	2.50	17	150
	1800	982	2.45	17	1.55	11	0.96	7	1.30	9	30
	2000	1093	1.30	9	0.76	5	0.46	3	.62	4	18
309S	1600	871	5.80	40	3.20	22	–	–	3.50	24	–
	1800	982	2.60	18	1.65	11	1.00	7	1.00	7	105
	2000	1093	1.40	10	0.83	6	0.48	3	.76	5	42
310S	1600	871	6.60	45	4.00	28	2.50	17	4.00	28	30
	1800	982	3.20	22	2.10	15	1.35	9	1.75	12	60
	2000	1093	1.50	10	1.10	7	0.76	5	.80	6	60
314	1600	871	4.70	32	3.00	21	1.95	13	2.30	16	110
	1800	982	2.60	18	1.70	12	1.10	8	1.00	7	120
	2000	1093	1.50	10	1.12	7	0.85	6	.90	6	82
316	1600	871	5.00	34	2.70	19	1.40	10	2.60	18	30
	1800	982	2.65	18	1.25	9	0.60	4	1.20	8	35
	2000	1093	1.12	8	0.36	2	–	–	4.00	28	–

Table 19
EFFECT OF PROLONGED HOLDING AT 900-1200 ºF (482-649ºC) ON ROOM-TEMPERATURE TOUGHNESS AND HARDNESS (2)

	Room-Temperature Charpy keyhole impact strength														Room-temperature Brinell hardness							
			after 1000 hr at						after 10,000 hr at							after 1000 hr at			after 10,000 hr at			
	Unexposed		900(482)		1050(566)		1200ºF(649ºC)		900(428)		1050(566)		1200ºF(649ºC)		Unex-	900	1050	1200ºF	900	1050	1200ºF
Type	ft-lb	J	ft-lb	J	ft-lb	J	ft-lb	J	ft-lb	J	ft-lb	J	ft-lb	J	posed	(482)	(566)	(649ºC)	(482)	(566)	(649ºC)
304	91	123	87	118	75	102	60	81	79	107	62	84	47	64	141	145	142	143	143	132	143
304L	82	111	93	126	76	103	72	98	85	115	71	96	63	85	137	140	134	134	143	143	143
309	95	129	120	163	85	115	43	58	120	163	51	69	44	60	109	114	109	130	140	153	159
310	75	102	–	–	48	65	29	39	62	84	29	39	2	3	124	119	119	130	152	174	269
316	80	108	86	117	72	98	44	60	87	118	49	66	21	28	143	151	148	170	145	163	177
321	107	145	101	137	90	122	69	94	88	119	72	98	62	84	168	143	149	166	156	151	148
347	56	76	60	81	55	75	49	66	63	85	51	69	32	43	169	156	167	169	156	169	123
405	35	47	–	–	36	49	26	35	–	–	39	53	34	46	165	–	143	137	–	143	143
410	33	45	–	–	41	56	27	37	39	53	3	4	21	28	143	–	114	154	124	143	128
430	46	62	–	–	32	43	34	46	1	1	3	4	4	5	184	–	186	182	277	178	156
446	1	1	–	–	1	1	1	1	1	1	1	1	1	1	201	–	211	199	369	255	239

II

LOW TEMPERATURE MECHANICAL PROPERTIES

Alloys for low-temperature service must have suitable engineering properties, such as yield and tensile strengths and ductility. Experience with brittle fracture of steel ships during World War II demonstrated that many metals may have satisfactory "room-temperature" characteristics but not perform adequately at low temperatures. Low temperature brittle fracture can occur, sometimes with catastrophic failure, without any warning by stretching, sagging, bulging or other indication of plastic failure. Alloys that are ordinarily ductile may suddenly fail at very low levels of stress.

In the handling and storage of liquid gases at cryogenic temperatures, few steels can be used, Austenitic stainless steels are among these few because they exhibit good ductility and toughness at the most severe of cryogenic temperatures – minus 423ºF (253ºC) and lower.

Table 20 shows tensile properties of several stainless steels at cryogenic temperatures. Austenitic grades show not only good ductility down to -423ºF (253ºC), but they also show an increase in tensile and yield strengths.

Toughness is also excellent as indicated by the impact strength values – although there is some decrease as temperature decreases. Table 21 shows results of impact tests on four austenitic grades in different plate thicknesses, indicating that toughness is not markedly affected by section size. Impact tests show that Type 304 is very stable over long periods of exposure and does not exhibit any marked degradation of toughness. Properly made welds also have excellent low temperature properties.

Austenitic grades cold worked to high strength levels are also suitable for low temperature service. Type 310 can be cold worked as much as 85% and still exhibit a good notched-to-unnotched tensile ratio down to -423ºF (-253ºC). Tests indicate that toughness levels at cryogenic temperatures are higher in cold worked Type 310 than in cold worked Type 301.

Table 20
TYPICAL MECHANICAL PROPERTIES OF STAINLESS STEELS AT CRYOGENIC TEMPERATURES (2)

Type	Test Temperature		Yield Strength 0.2% Offset		Tensile Strength		Elongation in 2"	Izod Impact	
	ºF	ºC	ksi	MPa	ksi	MPa	%	ft. lbs.	J
304	– 40	– 40	34	234	155	1,069	47	110	149
	– 80	– 62	34	234	170	1,172	39	110	149
	–320	–196	39	269	221	1,524	40	110	149
	–423	–252	50	344	243	1,675	40	110	149
310	– 40	– 40	39	269	95	655	57	110	149
	– 80	– 62	40	276	100	689	55	110	149
	–320	–196	74	510	152	1,048	54	85	115
	–423	–252	108	745	176	1,213	56		
316	– 40	– 40	41	283	104	717	59	110	149
	– 80	– 62	44	303	118	814	57	110	149
	–320	–196	75	517	185	1,276	59		
	–423	–252	84	579	210	1,448	52		
347	– 40	– 40	44	303	117	807	63	110	149
	– 80	– 62	45	310	130	896	57	110	149
	–320	–196	47	324	200	1,379	43	95	129
	–423	–252	55	379	228	1,572	39	60	81
410	– 40	– 40	90	621	122	841	23	25	34
	– 80	– 62	94	648	128	883	22	25	34
	–320	–196	148	1,020	158	1,089	10	5	7
430	– 40	– 40	41	283	76	524	36	10	14
	– 80	– 62	44	303	81	558	36	8	11
	–320	–196	87	607	92	634	2	2	3

HEAT TRANSFER PROPERTIES

Stainless steels are used extensively for heat exchangers because their ability to remain clean enhances heat transfer efficiency. For example, Figure 29 illustrates that films and scale on exchanger surfaces impair heat transfer to a far greater extent than the metal wall, which accounts for only 2% of the total resistance to heat flow. Table 22 supports this contention by showing that thermal conductivity of a metal has only a minor effect on the "U" valve, or the overall heat-transfer coefficient.

The degree to which other factors affect heat transfer are dependent on the type of fluid involved, its velocity, and the nature of scale or fouling buildup on the surface. Since corrosion and scale accumulation is minimal with stainless steels, there would be less difference in service performance among various metals than would be indicated by thermal conductivity data. The power generation industry, for instance, has very carefully analyzed transfer characteristics of heat exchanger materials and has conclusively demonstrated that stainless steels behave in a manner far superior to other materials.

Figure 30 compares two condenser tubing materials exposed simultaneously to identical operating conditions. In the early stages of the test the relative performance of both materials corresponded to published thermal conductivity figures. However, in only 240 days, the overall heat transfer rate of the stainless steel was found to surpass that of the Admiralty brass. The heat transfer rate for both materials decreased with time, but that of the Admiralty brass was more rapid due to fouling and corrosion, while the stainless steel was affected only by fouling. Similar results were observed in desalination tests conducted in Freeport, Texas. Two booklets on heat exchangers are available from the Nickel Development Institute. One is “A Discussion of Stainless Steels for Surface Condenser and Feedwater Heater Tubing,” and the other, “The Role of Stainless Steels in Industrial Heat Exchangers.”

Table 21
LOW-TEMPERATURE IMPACT STRENGTH OF SEVERAL ANNEALED AUSTENITIC STAINLESS STEELS (2)

AISI Type	Testing Temp °F	Testing Temp °C	Specimen orientation	Type of notch	Product size	Energy absorbed, ft-lb	Energy absorbed, J
304	– 320	– 196	Longitudinal	Keyhole	3-in. plate	80	108
304	– 320	– 196	Transverse	Keyhole	3-in. plate	80	108
304	– 320	– 196	Transverse	Keyhole	2½-in. plate	70	95
304	– 423	– 252	Longitudinal	Keyhole	½-in. plate	80	108
304	– 423	– 252	Longitudinal	V-notch	3½-in. plate	91.5	124
304	– 423	– 252	Transverse	V-notch	3½-in. plate	85	115
304L	– 320	– 196	Longitudinal	Keyhole	½-in. plate	73	99
304L	– 320	– 196	Transverse	Keyhole	½-in. plate	43	58
304L	– 320	– 196	Longitudinal	V-notch	3½-in. plate	67	91
304L	– 423	– 252	Longitudinal	V-notch	3½-in. plate	66	90
310	– 320	– 196	Longitudinal	V-notch	3½-in. plate	90	122
310	– 320	– 196	Transverse	V-notch	3½-in, plate	87	118
310	– 423	– 252	Longitudinal	V-notch	3½-in. plate	86.5	117
310	– 423	– 252	Transverse	V-notch	3½-in. plate	85	115
347	– 320	– 196	Longitudinal	Keyhole	½-in. plate	60	81
347	– 320	– 196	Transverse	Keyhole	½-in. plate	47	64
347	– 423	– 252	Longitudinal	V-notch	3½-in. plate	59	80
347	– 423	– 252	Transverse	V-notch	3½-in. plate	53	72
347	– 300	– 184	Longitudinal	V-notch	6½-in. plate	77	104
347	– 300	– 184	Transverse	V-notch	6½-in. plate	58	79

Table 22
EFFECT OF METAL CONDUCTIVITY ON "U" VALUES (18)

Application	Material	Film Coefficients Btu/hr/ft²/°F (W/m²●K) h_o	Film Coefficients h_i	Thermal Conductivity of Metal Btu/hr/ft²/°F/in. (W/m●K)	"U" Value Btu/hr/ft²/°F (W/m²●K)
Heating water with saturated steam	Copper	300 (1704)	1000 (5678)	2680 (387)	229 (1300)
	Aluminum	300 (1704)	1000 (5678)	1570 (226)	228 (1295)
	Carbon Steel	300 (1704)	1000 (5678)	460 (66)	223 (1266)
	Stainless Steel	300 (1704)	1000 (5678)	105 (15)	198 (1124)
Heating air with saturated steam	Copper	5 (28)	1000 (5678)	2680 (387)	4.98 (28)
	Aluminum	5 (28)	1000 (5678)	570 (226)	4.97 (28)
	Carbon Steel	5 (28)	1000 (5678)	460 (66)	4.97 (28)
	Stainless Steel	5 (28)	1000 (5678)	105 (15)	4.96 (28)

Where h_o=outside fluid film heat-transfer coefficient
h_i=inside fluid film heat-transfer coefficient
Stainless steel is 300 Series Type

$$\text{“U”} = \frac{1}{\frac{1}{h_o} + \frac{\text{thickness of metal wall}}{\text{thermal conductivity}} + \frac{1}{h_i}}$$

SHAPES, SIZES, AND FINISHES

Exhibit 1 illustrates mill processes for making various stainless steel products. Because alloy composition must be very carefully controlled, various refining steps are used in conjunction with electric furnace (or vacuum furnace) melting and at the Argon Oxygen Decarburization (AOD) vessel. Other refining steps are vacuum arc, partial pressure inert gas arc, electron beam, and electroslag consumable arc remelting practices. During these remelting steps, certain impurities are reduced to minimum levels, and inclusion levels are lowered.

During the final stages of producing basic mill forms – sheet, strip, plate, and bar – and bringing these forms to specific sizes and tolerances, the materials are subjected to hot reduction with or without subsequent cold rolling operations, annealing, and cleaning. Further steps are required to produce other mill forms, such as wire and tubing.

Table 23 shows how the mill forms are classified by size, and Tables 24, 25, and 26 identify finishes and conditions in which sheet, bar, and plate are available.

Finishes are produced by three basic methods. These are (1) rolling between polished or textured rolls, (2) polishing and/or buffing with abrasive wheels, belts, or pads, and (3) blasting with abrasive grit or glass beads. The resulting surface textures vary from the "natural" appearance produced by hot or cold rolling (or by extrusion) to mirror-bright surfaces.

Rolled finishes result from the initial forming of the metal at the mill. These are the simplest and usually the lowest in cost, and they include a wide range of appearance depending on the character of the rolls themselves, which can be highly polished or etched to produce a dull matte finish.

Patterned finishes are also made by rolling, and they are available in a wide variety of sculptural designs and textures, all of which are proprietary in nature and not included in AISI numbered finish designations. These are produced either by passing mill-rolled finished sheet between two mating rolls of specific design or by impressing different patterns on each side of the sheet. These finishes usually have the added advantage of stiffness. They are supplied by some mills and by specialty processors.

Polished finishes are produced by mechanical polishing, and sometimes by buffing, and are characterized by fine parallel grit lines – the fineness being determined by the grit size used in the final step. These grit lines can impart a directional character to the finish, which when present should be considered in final product design. Some polished and buffed finishes are produced by some mill and specialty processors.

Blast finishes are applied by conventional blasting techniques using glass beads.

While finishes are usually selected for appearance, selection cannot be made independently of fabrication considerations. For example, if evidence of exposed welds is to be removed, rolled nondirectional finishes are generally not specified because they cannot be blended or refinished. A polished finish such as No. 4, on the other hand, can be blended by matching grit size to that used for the original polish. It is also important to consider the effects of various fabrication methods used in the manufacture of the stainless steel products. Severe forming, for example, can distort or locally remove grit lines, so it may be necessary to refinish the surface after fabrication, such as in the manufacture of pots and pans.

It should also be recognized that each finish can have variations in appearance depending upon composition, thickness, method of application, and supplier. In rolled finishes, the thinner the sheet, the smoother the surface. The 200 and 300 Series stainless steels have a characteristically different appearance than 400 Series types. Color variations may occur among the different types within the same metallurgical category. The practicability of describing any of the finishes in terms of measurable limits, such as for smoothness or reflectance, has not been established, so designers are encouraged to provide samples showing the final finish desired.

Tables 24, 25, and 26, as mentioned previously, show AISI numbered finishes and conditions for sheet, bar and plate. While there are no specific designations for polished finishes on bar or plate, the sheet finish designations are often used to describe the desired effect. This also applies to finishes on ornamental tubing.

There are three standard finishes for strip, which are broadly described by the finishing operations employed:

No. 1 Strip Finish is approximately the same as No. 2D Sheet Finish. It varies in appearance from dull gray matte to a fairly reflective surface, depending largely on alloy composition and amount of cold reduction.

No. 2 Strip Finish is approximately the same as a No. 2B sheet finish. It is smoother, more reflective than No. 1, and likewise varies with alloy composition.

Bright Annealed Finish is a highly reflective finish that is retained by final annealing in a controlled atmosphere furnace.

Mill-Buffed Finish is a bright cold rolled, highly reflective finish obtained on either No. 2 or on bright annealed strip by continuously buffing in coil form. The purpose of mill-buffing is to provide a uniform finish with regard to color and reflectivity. It can also provide a surface receptive to chromium plating. The finish has wide use in automotive trim, household trim, tableware, utensils, fire extinguishers, plumbing fixtures, etc.

Because of the wide variety of standard and nonstandard finishes, designers are encouraged to examine samples before selecting a finish.

FABRICATION

Stainless steels are generally selected, first on the basis of corrosion resistance and, second, do the basis of strength or other mechanical properties. A third-level consideration is fabrication. While the three general-purpose stainless types predominate, namely Types 304, 430, and 410, there are variations of these types that are better suited to certain manufacturing operators. (Service requirements may preclude the use of these variations, so it is well to know that all stainless steels can be readily fabricated by conventional manufacturing methods.)

The handbook "Stainless Steel Fabrication" is available from the SSINA and describes these alloys and various fabrication methods.

HOT FORMING

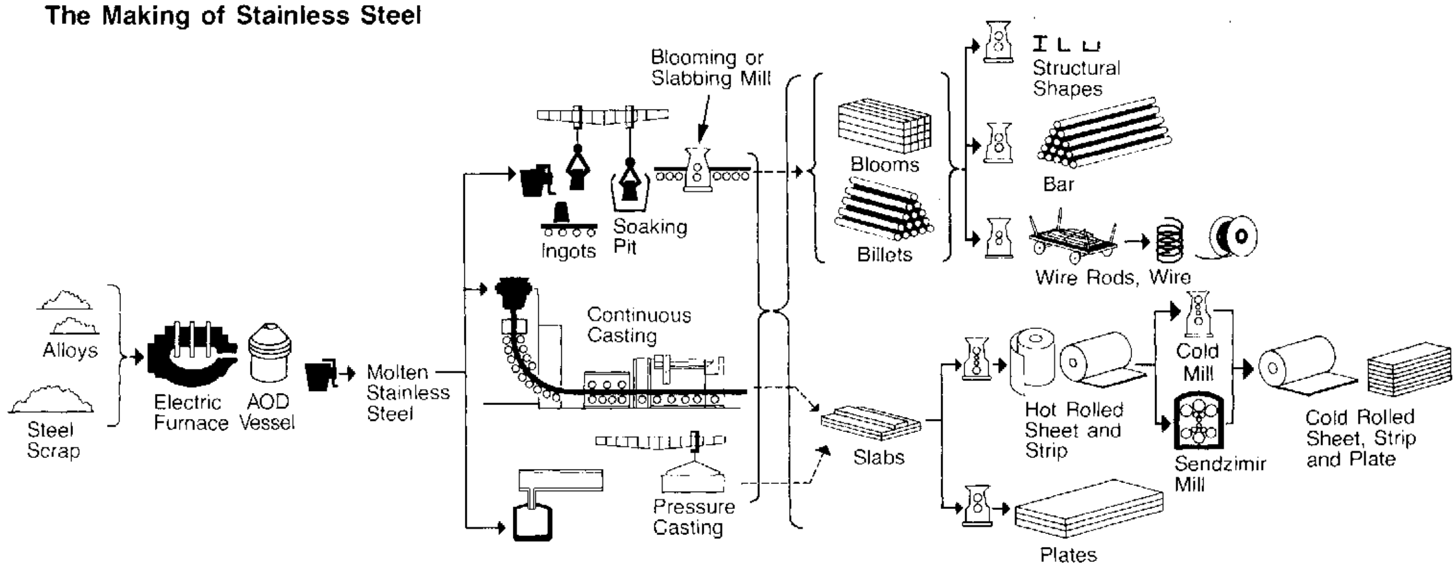
Stainless steels are readily formed by hot operations such as rolling, extrusion, and forging – methods that result in finished or semifinished parts.

Hot rolling is generally a steel mill operation for producing standard mill forms and special shapes. Exhibit 2 illustrates the variety of hot-rolled, cold-rolled, and cold-drawn shapes available in stainless steel bar.

Exhibit 3 provides some design guidelines for extrusion.

Forging is used extensively for stainless steels of all types (Exhibit 4).

Exhibit 1
The Making of Stainless Steel



II

Table 23
CLASSIFICATION OF STAINLESS STEEL PRODUCT FORMS (2)

Item	Description	Dimensions		
		Thickness	Width	Diameter or Size
Sheet	Coils and cut lengths: Mill finishes Nos. 1, 2D & 2B Pol. finishes Nos. 3, 4, 6, 7 & 8	under 3/16" (4.76 mm) " " "	24" (609.6 mm) & over all widths	-
Strip	Cold finished, coils or cut lengths Pol. finishes Nos. 3, 4, 6, 7 & 8	under 3/16" (4.76 mm) " " "	under 24" (609.6 mm) all widths	-
Plate	Flat rolled or forged	3/16" (4.76 mm) & over	over 10" (254 mm)	-
Bar	Hot finished rounds, squares, octagons and hexagons	-	-	¼" (6.35 mm) & over
	Hot finished flats	1/8" (3.18 mm) to 8"(203 mm) incl.	¼" (6.35 mm) to 10" (254 mm) incl.	-
	Cold finished rounds, squares, octagons and hexagons	-	-	over 1/8" (3.18 mm)
	Cold finished flats	1/8" (3.18 mm) to 4½" (114 mm)	3/8" (9.53 mm) to 4½" (114 mm)	-
Wire	Cold finishes only: (in coil) Round, square, octagon, hexagon, and flat wire	under 3/16" (4.76 mm)	under 3/8" (9.53 mm)	-
Pipe & Tubing	Several different classifications, with differing specifications, are available, For information on standard sizes consult your local Steel Service Center or the SSINA.			
Extrusions	Not considered "standard" shapes, but of potentially wide interest. Currently limited in size to approximately 6½" (165.1 mm) diameter, or structurals.			

Table 24

STANDARD MECHANICAL SHEET FINISHES (2)

Unpolished or Rolled Finishes:

No. 1 A rough, dull surface which results from hot rolling to the specified thickness followed by annealing and descaling.

No. 2D A dull finish which results from cold rolling followed by annealing and descaling, and may perhaps get a final light roll pass through unpolished rolls. A 2D finish is used where appearance is of no concern.

No. 2B A bright, cold-rolled finish resulting in the same manner as No. 2D finish, except that the annealed and descaled sheet receives a final light roll pass through polished rolls. This is the general-purpose cold-rolled finish that can be used as is, or as a preliminary step to polishing.

Polished Finishes;

No. 3 An intermediate polish surface obtained by finishing with a 100-grit abrasive. Generally used where a semifinished polished surface is required. A No. 3 finish usually receives additional polishing during fabrication.

No. 4 A polished surface obtained by finishing with a 120-150 mesh abrasive, following initial grinding with coarser abrasives. This is a general-purpose bright finish with a visible "grain" which prevents mirror reflection.

No. 6 A dull satin finish having lower reflectivity than No. 4 finish. It is produced by Tampico brushing the No. 4 finish in a medium of abrasive and oil. It is used for architectural applications and ornamentation where a high luster is undesirable, and to contrast with brighter finishes.

No. 7 A highly reflective finish that is obtained by buffing finely ground surfaces but not to the extent of completely removing the "grit" lines. It is used chiefly for architectural and ornamental purposes.

No. 8 The most reflective surface; which is obtained by polishing with successively finer abrasives and buffing extensively until all grit lines from preliminary grinding operations are removed. It is used for applications such as mirrors and reflectors.

II

Exhibit 2

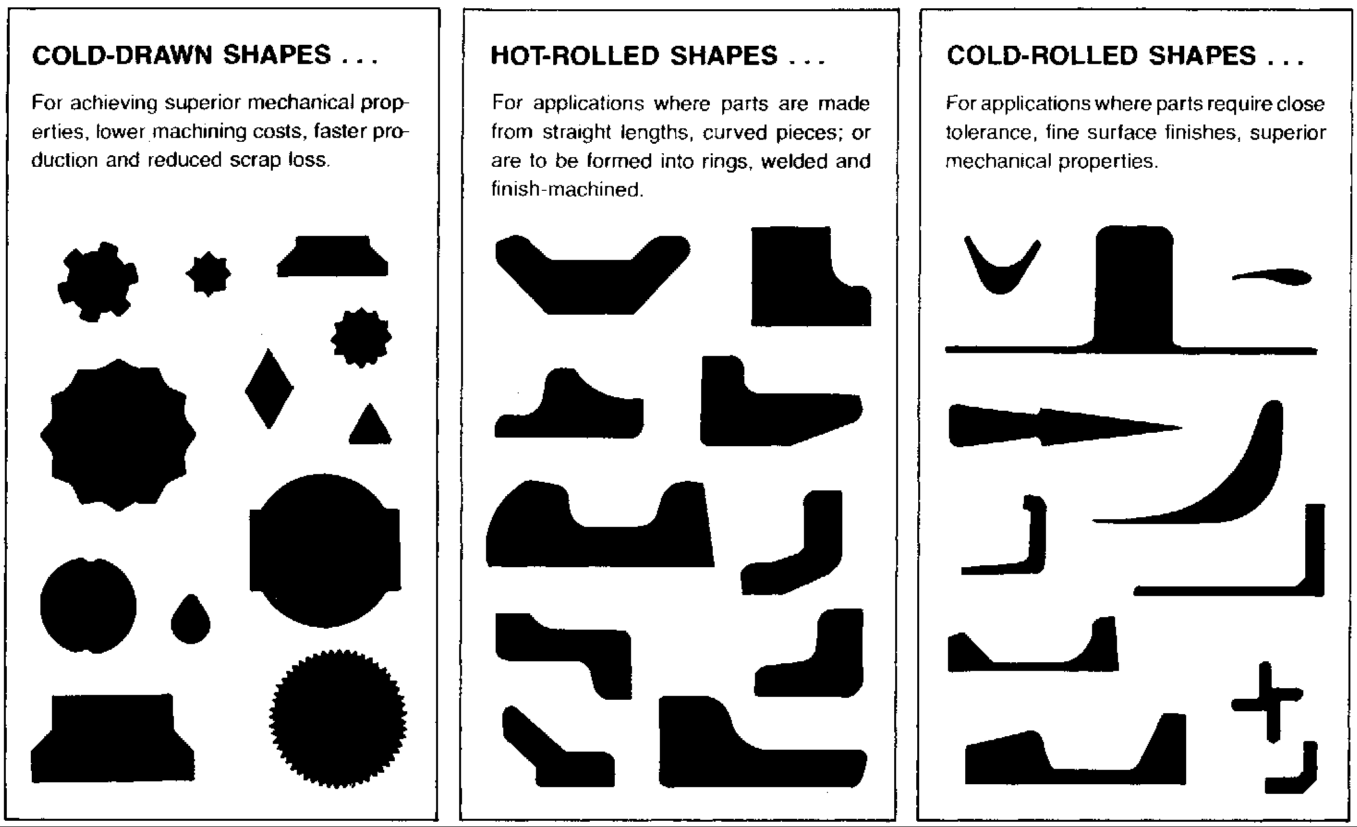


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Exhibit 3
Extrusion Guidelines

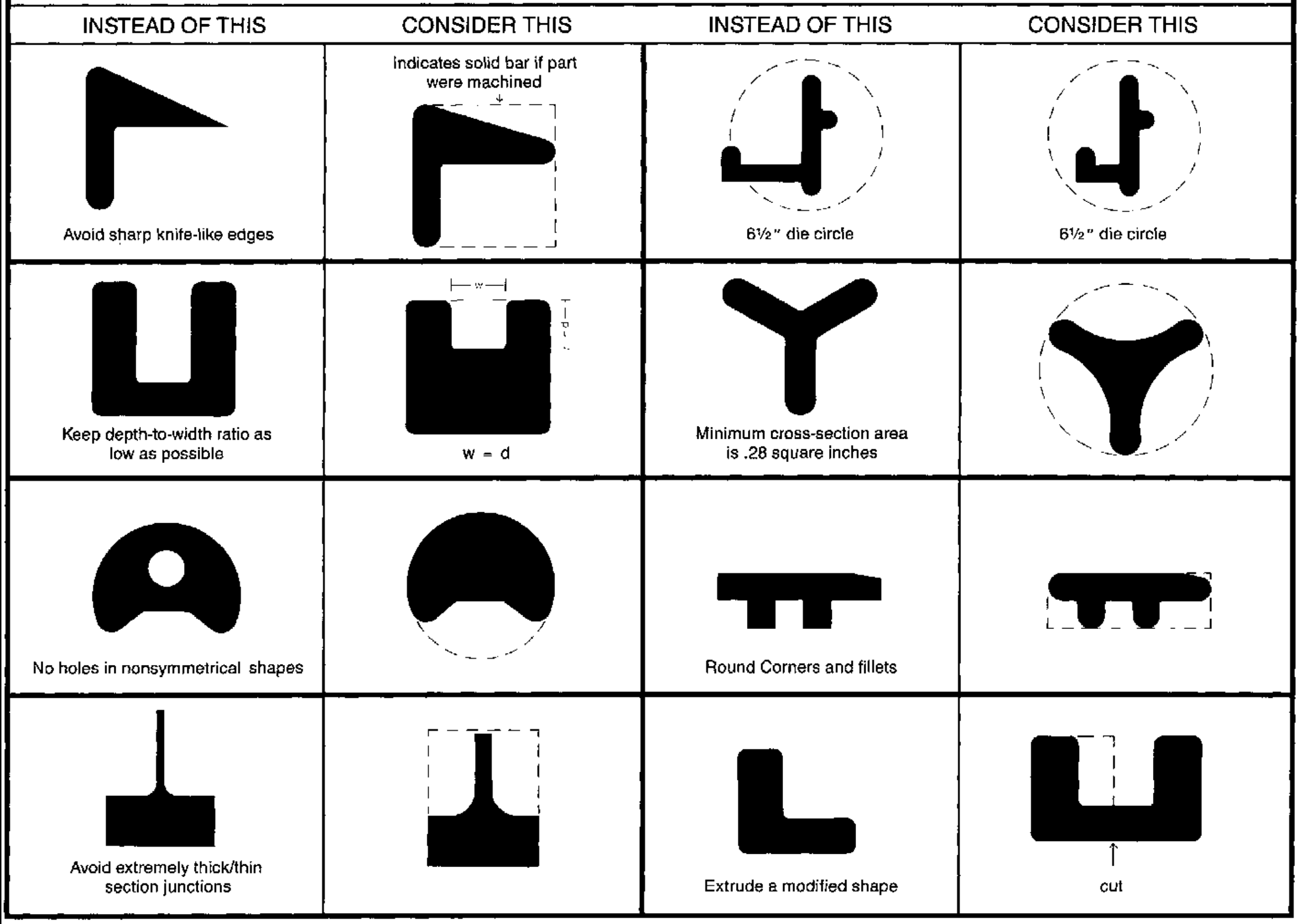
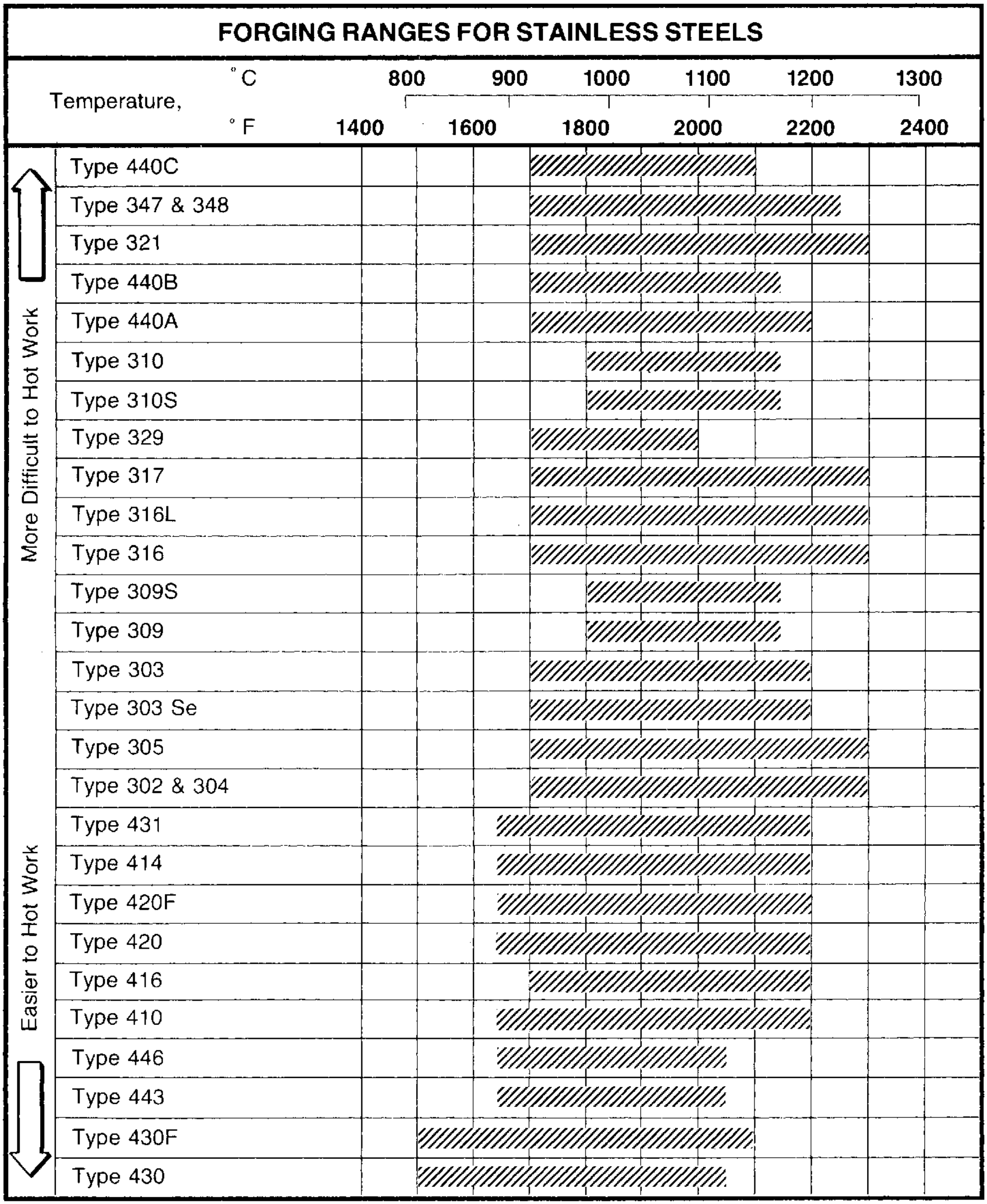


Exhibit 4


FORGING RANGES FOR STAINLESS STEELS
Temperature,
°C
800
900
1000
1100
1200
1300
°F
1400
1600
1800
2000
2200
2400
More Difficult to Hot Work
Easier to Hot Work
Type 440C
Type 347 & 348
Type 321
Type 440B
Type 440A
Type 310
Type 310S
Type 329
Type 317
Type 316L
Type 316
Type 309S
Type 309
Type 303
Type 303 Se
Type 305
Type 302 & 304
Type 431
Type 414
Type 420F
Type 420
Type 416
Type 410
Type 446
Type 443
Type 430F
Type 430

II

Table 25

CONDITIONS & FINISHES FOR BAR (2)

Conditions		Surface Finishes*
1. Hot worked only	(a)	Scale not removed (excluding spot conditioning)
	(b)	Rough turned**
	(c)	Pickled or blast cleaned and pickled
2. Annealed or otherwise heat treated	(a)	Scale not removed (excluding spot conditioning)
	(b)	Rough turned
	(c)	Pickled or blast cleaned and pickled
	(d)	Cold drawn or cold rolled
	(e)	Centerless ground
	(f)	Polished
3. Annealed and cold worked to high tensile strength***	(d)	Cold drawn or cold rolled
	(e)	Centerless ground
	(f)	Polished

* Surface finishes (b), (e) and (f) are applicable to round bars only.

** Bars of the 4xx series stainless steels which are highly hardenable, such as Types 414, 420, 420F, 431, 440A, 440B and 440C, are annealed before rough turning. Other hardenable grades, such as Types 403, 410, 416 and 416Se, may also require annealing depending on their composition and size.

*** Produced in Types 302, 303Se, 304 and 316.

Table 26

CONDITIONS & FINISHES FOR PLATE (2)

Condition and Finish	Description and Remarks
Hot rolled	Scale not removed. Not heat treated. Plates not recommended for final use in this condition.*
Hot rolled, annealed or heat treated	Scale not removed. Use of plates in this condition is generally confined to heat resisting applications. Scale impairs corrosion resistance.*
Hot rolled, annealed or heat treated, blast cleaned or pickled	Condition and finish commonly preferred for corrosion resisting and most heat resisting applications.
Hot rolled, annealed, descaled and temper passed	Smoother finish for specialized applications.
Hot rolled, annealed, descaled cold rolled, annealed, descaled, optionally temper passed	Smooth finish with greater freedom from surface imperfections than the above.
Hot rolled, annealed or heat treated, surface cleaned and polished	Polished finishes: refer to Table 24.

*Surface inspection is not practicable on plates which have not been pickled or otherwise descaled.

II

Table 27
RELATIVE FORMING CHARACTERISTICS OF AISI 200 AND 300 SERIES (2)
(Not Hardenable by Heat Treatment)

Forming Method	201	202	301	302	302B	303, 303Se	304	304L	305	308	309	309S	310	310S	314	316	316L	317	321	347, 348	384
Blanking	B	B	B	B	B	B	B	B	B	B	B	B	B	B	B	B	B	B	B	B	B
Brake forming	B	A	B	A	B	D	A	A	A	B	A	A	A	A	A	A	A	A	A	A	B
Coining	B-C	B	B-C	B	C	C-D	B	B	A-B	D	B	B	B	B	B	B	B	B	B	B	A
Deep drawing	A-B	A	A-B	A	B-C	D	A	A	B	D	B	B	B	B	B-C	B	B	B	B	B	B
Embossing	B-C	B	B-C	B	B-C	C	B	B	A-B	D	B	B	B	B	B-C	B	B	B	B	B	A-B
Forging, cold	C	B	C	B	B	D	B	B	A-B	D	B-C	B-C	B	B	B-C	B	B	B	B	B	A
Forging, hot	B	B	B	B	B	B-C	B	B	B	B	B-C	B-C	B-C	B-C	B-C	B	B	B-C	B	B	B
Heading, cold	C-D	C	C-D	C	D	D-C	C	C	B-A	D	C	C	C	C	C-D	C	C	C	C	C	A
Heading, hot	B	B	B	B	B	C	B	B	B	B	C	C	C	C	C	B	B	B-C	C-B	C-B	B
Punching	C	B	C	B	B	B	B	B	B	–	B	B	B	B	B	B	B	B	B	B	B
Roll forming	B	A	B	A	–	D	A	A	A	–	B	B	A	A	B	A	A	B	B	B	A
Sawing	C	C	C	C	C	B	C	C	C	C	C	C	C	C	C	C	C	C	C	C	C
Spinning	C-D	B-C	C-D	B-C	C	D	B	B	A	D	C	C	B	B	C	B	B	B-C	B-C	B-C	A

A = Excellent B = Good C = Fair D = Not generally recommended
Note: Ratings are for making comparisons of alloys within their own metallurgical group. They should not be used to compare 300 Series with 400 Series types.

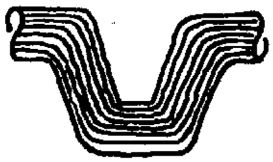
Table 28
RELATIVE FORMING CHARACTERISTICS OF AISI 400 SERIES (2)

	FERRITIC (Not Hardenable by Heat Treatment)								MARTENSITIC (Hardenable by Heat Treatment)									
Forming Method	430	430F, 430FSe	405	429	434	436	442	446	410	403	414	416, 416Se	420	420F	431	440A	440B	440C
Blanking	A	B	A	A	A	A	A	A	A	A	A	B	B	B	C-D	B-C	–	–
Brake forming	A*	B-C*	A*	A*	A*	A*	A"	A*	A*	A*	A*	C*	C*	C*	C*	C*	–	–
Coining	A	C-D	A	A	A	A	B	B	A	A	B	D	C-D	C-D	C-D	D	D	D
Deep drawing	A-B	D	A	A-B	B	B	B	B-C	A	A	B	D	C-D	C-D	C-D	C-D	–	–
Embossing	A	C	A	A	A	A	B	B	A	A	C	C	C	C	C-D	C	D	D
Forging, cold	B	D	B	B	B	B	B-C	C	B	B	C	D	C-D	C-D	C-D	C-D	D	D
Forging, hot	B	C	B	B	B	B	B-C	B-C	B	B	B	C	B	B	B	B	B	B
Hardening by cold work, typical tensile strength (1000 psi)																		
Annealed	73	75	–	–	–	–	–	–	90	90	–	70	–	–	–	–	–	–
25% reduction	96	95	–	–	–	–	–	–	120	120	–	90	–	–	–	–	–	–
50% reduction	115	110	–	–	–	–	–	–	130	130	–	105	–	–	–	–	–	–
Hardening by heat treatment	No	No	No	No	No	No	No	No	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
Heading, cold	A	D	A	A	A	A	B	C	A	A	D	D	C	C	C-D	C	C-D	C-D
Heading, hot	B	C	B	B	B	B	B-C	B-C	B	B	B	C	B	B	B	B	B	B
Punching	A-B	A-B	A-B	A-B	A-B	A	A-B	B	A-B	A-B	B	A-B	B-C	B-C	C-D	–	–	–
Roll forming	A	D	A	A	A	A	A	B	A	A	C	D	C-D	C-D	C-D	C-D	–	–
Sawing	B	A-B	B	B	B	B	B-C	B-C	B	B	C	B	C	C	C	C	C	C
Spinning	A	D	A	A	A-B	A-B	B-C	C	A	A	C	D	D	D	D	D	D	D

A = Excellent B = Good C = Fair D = Not generally recommended *Severe sharp bends should be avoided

II

Exhibit 5



Forging
True grain flow

Casting
No grain flow

Bar Stock
Grain flow broken
by machining

II

A unique feature of forgings is that the continuous grain flow follows the contour of the part, as illustrated by the top drawing in Exhibit 5. In comparison is the random grain structure of a cast part (center) and the straight-line orientation of grain in a machined part (bottom). From this difference stem secondary advantages inherent in forged stainless steels as follows:

Strength where needed. Through grain refinement and flow, forging puts the strength where it's needed most.

Lighter weight. Higher strength-to-weight ratio permits the use of thinner, light weight sections without sacrificing safety.

Improved mechanical properties. Forging develops the full impact resistance, fatigue resistance, ductility, creep-rupture life, and other mechanical properties of stainless steels.

Repeatable dimensions. Tolerances of a few thousandths are routinely maintained from part to part, simplifying final fixturing and machining requirements.

Structural uniformity. Forgings are sound, nonporous, and uniform in metallurgical structure. The booklet, "Stainless Steel Forgings," available from NiDI discusses in greater detail the forgability of stainless steels, and it provides guidelines for designing stainless forgings.

COLD FORMING

The mechanical properties of stainless steels serve as an indication of their relative formability at ambient or room temperature. Annealed austenitic grades are typified as having low yield strengths, high tensile strengths, and high elongations. Some of these alloys work harden to a high degree, which further increases their strength properties. The ferritic alloys have much lower ductility than the austenitic types and are closer to carbon steel with respect to mechanical properties; and they do not work harden significantly during cold forming.

Because of their excellent mechanical properties, stainless steels have excellent cold-forming characteristics. Table 27 shows the relative fabrication characteristics of the austenitic stainless steels and Table 28 shows the relative fabrication characteristics of the martensitic and ferritic grades.

Sheet, Strip and Plate

The bending characteristics of annealed austenitic stainless steels, as indicated by Table 29, are considered excellent. Many types will withstand a free bend of 180 degrees with a radius equal to one-half the material thickness or less. In a controlled V-block, the bend angle limit is 135 degrees. As the hardness of the stainless increases, bending becomes more restrictive. This is indicated by the data in Table 30 which show, for example, the free-bend characteristics of Type 301 in the ¼, ½, ¾, and full-hard temper. The bend characteristics of the 400 Series types, shown in Table 31 are also good. However, they tend to be somewhat less ductile than the 300 Series types, so the minimum bend radius is equal to the material thickness.

Table 29
BENDING CHARACTERISTICS: Annealed Stainless Steel Sheet & Strip (2)

Type	Free Bend	V-Block
301, 302, 304, 305, 309, 310, 316, 321, 347	180° R=½T	135° R=½T

NOTE: R=radius of bend; T=thickness of material. All bends are parallel to direction of rolling.

Table 30
BENDING CHARACTERISTICS: Temper Rolled Stainless Steel Sheet & Strip (2)

Type	Temper	Gage 0.050 in. (1.27 mm) and under Free Bend	Gage 0.051 in.-0.187 in. (1.30-4.75 mm)
301	¼ hard	180° R=½T	90° R=T
301	½ hard	180° R=T	90° R=T
301	¾ hard	180° R=1½T	–
301	full hard	180° R=2T	–
302	¼ hard	180° R=½T	90° R=T
316	¼ hard	180° R=T	90° R=T
		V-Block	
301	¼ hard	135° R=T	135° R=1½T
301	½ hard	135° R=2T	135° R=2T
301	¾ hard	135° R=3T	–
301	full hard	135° R=3T	–
302	¼ hard	135° R=2T	135° R=2T
316	¼ hard	135° R=2½T	135° R=3T

NOTE: R=radius of bend; T=thickness of material. All bends are parallel to direction of rolling.

Table 31
Typical Bending Characteristics of Annealed Stainless Steel Sheet, Strip and Plate (2)

AISI Type	Gage to 0.374" (9.50 mm)		Gage 0.375" to 0.500" (9.53-12.7 mm)	
	Free Bend	V-Block	Free Bend	V-Block
405	180° R=T	135° R=T	180°R=T	135° R=2T
410	180° R=T	135° R=T	180°R=T	135° R=2T
430	180° R=T	135° R=T	180°R=T	135° R=2T
442	180° R=T	135° R=T	180°R=2T	135° R=2T
446	180° R=T	135° R=T	180°R=2T	135° R=2T

NOTE: R=radius of bend; T=thickness of material. All bends are parallel to direction of rolling.

In simple bending operations, there is little need to consider variations of the general-purpose alloys, since all stainless steels within a metallurgical group tend to behave in a similar manner. However, in the more complex forming operations in which the metal is pressed, drawn, or stretched, considerable latitude exists for alloy selection. This can be visualized somewhat when one considers the need for extensive work hardening when a part is made essentially by stretching.

All of the 300 Series alloys work harden considerably and can be stretched severely, but this property is exemplified by Type 301. During stretching, holddown pressures are applied to the flange areas to prevent metal from flowing into the die. During this stretch, severe metal thinning occurs. However, as the metal thins it work-hardens sufficiently to exceed the strength in the thicker (less strong) sections, thus preventing cracking or tearing. A good example of stretching and the need for Type 301 is the manufacture of automobile wheel-covers.

At the other extreme is Type 305 with a low work-hardening rate that prevents excessive strengthening. It has good ductility and is an excellent choice for deep drawing in which little hold-down pressure is used. The use of this material can minimize annealing for multiple draws.

Between Types 301 and 305 is Type 304, which is the preferred choice when the forming operation combines both drawing and stretching.

The ferritic stainless steels do not exhibit nearly as high ductility as the austenitic types, nor do they have significant work-hardening traits. Their formability is thus more like that of carbon steel in that they cannot be stretched without thinning and fracturing – and formability usually decreases with increasing chromium content. In addition, these grades can show brittle tendencies that become more pronounced with increasing chromium content. To offset this factor, moderate warming of the higher chromium types is often recommended prior to drawing.

Bar and Wire

Many components of stainless steel are made from bar or wire with cold heading being the most widely used forming method. Many types of stainless are available as cold heading wire.

However, for multiple-blow operations in which a number of forming steps are performed in rapid sequence, it is desirable to use a material with good ductility but with a low work-hardening rate. Three stainless steels predominate in this respect; Types 305, 384, and (UNS) S30430. These three are similar to Type 304 in terms of corrosion resistance and mechanical properties.

With a chromium-nickel ratio less than that of Type 304, Type 305 has less tendency to work harden. Accordingly, a greater amount of deformation is possible before annealing is necessary. It is also readily available as bar and wire for cold heading, cold extrusion, and other cold-forming processes. In terms of corrosion resistance, it is freely interchangeable with Type 304. Type 305 resists attack by nitric acid and is used in a wide range of organic and inorganic chemicals, foodstuffs, and sterilizing solutions. It also has good high-temperature scaling resistance, and is utilized for continuous service at 1600ºF (871ºC). Unlike Type 304, however, Type 305 remains nonmagnetic even after severe cold work.

Type 384 is widely used for fasteners, cold-headed bolts, screws, upset nuts, and instrument parts, also for severe coining, extrusion, and swaging. It is also ideally suited for thread rolling. Because Type 384 is similar in chromium content to Type 304, it generally can be used anywhere that Type 304 is used. Exhibit 6 truss square shoulder bolt warm-headed in Type 384.

Both Types 305 and 384 are subject to carbide precipitation if heated or cooled slowly in the range of 800-1650ºF (427-899ºC). This can lead to intergranular corrosion in aggressive environments. This condition can be corrected, however, by annealing and water quenching from at least 1900ºF (1038ºC). Type (UNS) S30430 is one of the most widely used cold-heading stainless steels that is often identified by a popular trade name 304 HQ. Its composition is similar to that of Type 304 except that it contains 3.00-4.00% copper. This eliminates cracking, especially in recessed heads, and results in improved tool performance. It becomes mildly magnetic after severe cold working.

The three types just described are the principal stainless steels used for cold-heading operations. However, many other stainless steel types are available as coldheading wire. They include Types 410, 430, 440C, (UNS) S17400 (a precipitation hardening type) and several proprietary stainless alloys.

It is well to keep in mind that the inherent high strength of stainless steels requires more power in forming than that for forming carbon steel. And since many of the alloys work harden rapidly in cold-forming operations, there is need for added power after the start of initial deformation. The booklet, "Cold Forming Stainless Steel Bar and Wire," which is available from NiDI, describes in greater depth the selection of materials and the design of products to be fabricated by cold forming methods.

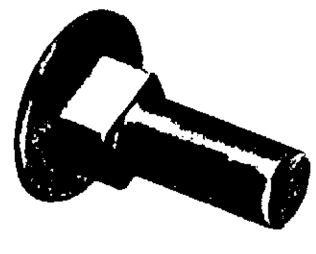


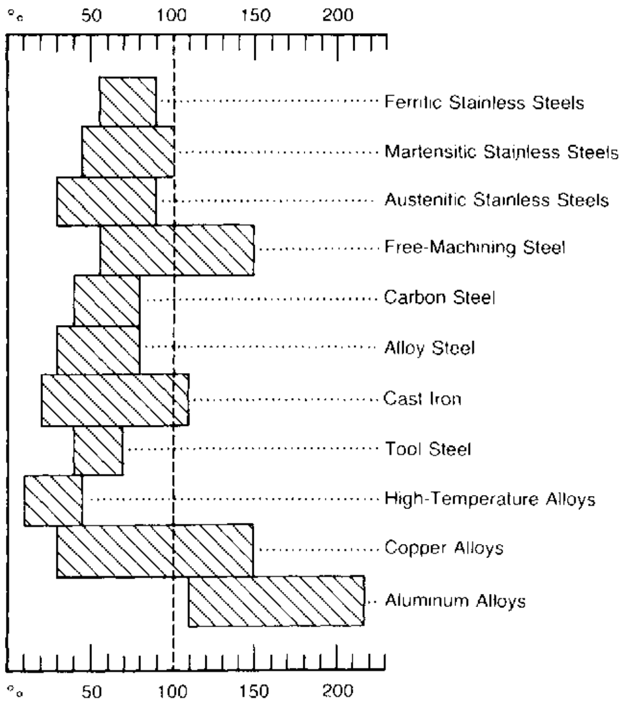
Exhibit 6

A warm-headed truss square shoulder bolt of Type 384 stainless. By using warm heading, less forming pressure was required on the die resulting in increased tool life.

II

Exhibit 7

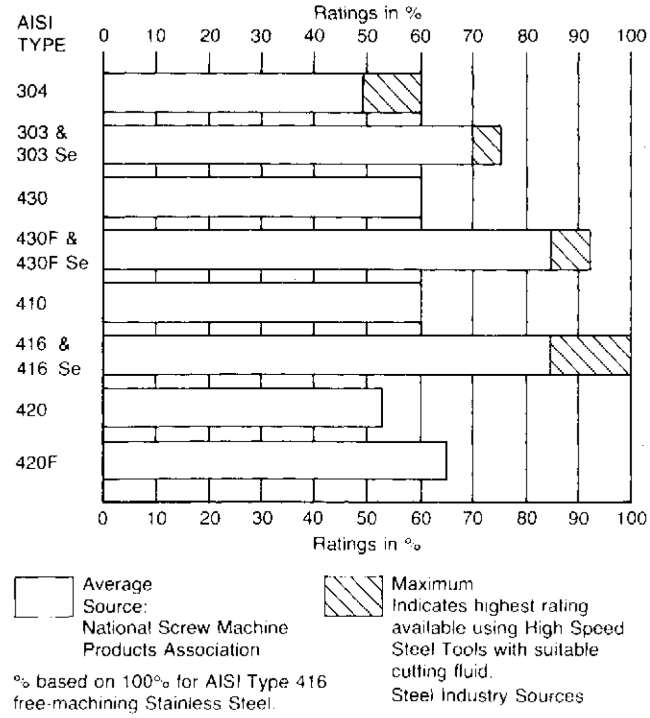
Comparative Machinability of Common Metals



100% Type 416 Stainless Steel

Exhibit 8

Comparative Machinability of Frequently Used Stainless Steels and Their Free-Machining Counterparts



It has been traditional in machining literature to compare the machinability of various metals to AISIB-1112 which is a free-machining carbon steel. However, since Type 416 stainless steel has a machining rating equal to that of B-1112, and since B-1112 is no longer on the market, comparisons in this booklet will be made with Type 416 as the base at 100%.

MACHINING

The machining characteristics of stainless steels are substantially different from those of carbon or alloy steels and other metals, as illustrated in Exhibit 7, "Comparative Machinability of Common Metals." In varying degree, most stainless steels without composition modification are tough, rather gummy, and they tend to seize and gall.

While the 400 Series stainless steels are the easiest to machine, a stringy chip produced during the machining, can slow productivity. The 200 and 300 Series, on the other hand, have the most difficult machining characteristics, primarily because of their propensity to work harden at a very rapid rate.

An experienced machine shop production engineer can work around these conditions and achieve good productivity with any of the stainless steels. However, wherever conditions permit, the design engineer can help minimize problems and get maximum machining productivity. Here are three suggestions: (1) specify a free-machining stainless steel, (2) suggest to the production engineer that he use a special analysis stainless steel that is "more suited for machining," or (3) specify stainless steel bar for machining that is in a slightly hardened condition.

Free-Machining Stainless Steels

Some stainless steel compositions contain sulfur, selenium, lead, copper, aluminum, or phosphorus – either separately or in combination – in sufficient quantity to improve the machining characteristics of the metal. These alloying elements reduce the friction between the workpiece and the tool, thereby minimizing the tendency of the chip to weld to the tool. Also, sulfur and selenium form inclusions that reduce the friction forces and transverse ductility of the chips, causing them to break off more readily. The improvement in machinability in the free-machining stainless steels – namely Types 303, 303 Se, 430F, 430F Se, 416, 416 Se, and 420F – is clearly evident in Exhibit 8, "Comparative Machinability of Frequently Used Stainless Steels." (Also, excellent are several proprietary free-machining alloys.)

Suppose, for example, that Type 304 is being considered on the basis of corrosion resistance and strength, but the machine shop needs the best possible machining rate. Type 303 could be specified as an alternate, provided its properties meet end-use requirements and 304 is not specifically requested. The chromium, nickel, and sulfur contents of Type 303 are slightly different than those of Type 304, and as a result Type 303 can be machined at speeds about 25-30% faster than Type 304.

Type 303 Se is another free-machining stainless steel similar to Type 303 except that it contains selenium as the ingredient to enhance the machining characteristics. It is used for better surface finishes or when cold working may be involved, such as staking, swaging, spinning, or severe thread rolling, in addition to machining.

If end-use conditions call for Type 430 stainless, the designer can specify free-machining Types 430F or 430F Se, which have similar properties although less corrosion resistance. Type 430F contains more sulfur, while 430F Se contains selenium instead of a high sulfur content.

The free-machining choices in lieu of Type 410 are Types 416 (higher sulfur) or 416 Se (selenium). And for Type 420, the shop might consider Type 420F.

It should be understood that the alloying elements used to improve the free-machining characteristics of stainless steels can adversely affect corrosion resistance, transverse ductility, and other qualities, such as weldability. These grades should be used only after careful consideration, but when used, they will machine at significantly higher production rates.

Special Analysis Stainless Steels

If, for example, end-use conditions are too restrictive to permit the use of Type 303 instead of Type 304, designers might suggest using Type 304 with a special analysis that has somewhat better machining qualities but with very little difference in corrosion resistance. In melting of special analysis stainless steels, minor modifications are made in the composition to enhance certain characteristics of the metal. A stainless steel producer should be consulted.

Hardened Stainless Steel Bar

When conditions require maximum resistance to corrosion in the alloy selected, and there is no room for compromise in the composition of the stainless steel, the machine shop can order bar stock in a slightly hardened condition that may result in a small improvement in machinability. Under any circumstances, and especially when corrosive environments are involved, it is always good practice to consult with a stainless steel producer.

Screw Machining Operations

Automatic screw machining is a fast and efficient method for machining that benefits greatly from the use of the free-machining stainless steels. In many typical screw machine applications, parts are turned out at rates as high as 300 to 400 pieces per hour. However, one should not have any misgivings about screw machining any of the stainless steels. With appropriate design and good shop practices, even the non free-machining types can be handled at relatively high rates. The machinability of stainless steels in general has improved significantly in the past few years, primarily through melting and refining practices that permit tighter analysis control.

JOINING

Welding

Nearly all of the stainless steels can be welded by most methods employed by industry today. Because of differences between these alloys and carbon or low-alloy steels, however, there are variations in welding techniques. First, it is important that procedures be followed to preserve corrosion resistance in the weld and in the area immediately adjacent to the weld, referred to as the heat-affected-zone (HAZ). Second, it is desirable to maintain optimum mechanical properties in the joint, and, third, certain steps are necessary to minimize problems of heat distortion. The principal difference between stainless and other steel types is alloy content, which provides corrosion resistance. In welding, it is necessary to select a weld rod that provides weld filler metal having corrosion resistant properties as nearly identical to the base metal as possible or better. This is not always as obvious as some might expect. For instance, a Type 308 weld rod is specified for welding Type 304, and a 300 Series rod is often used for joining 400 Series types. The best suggestion is to follow American Welding Society (AWS) practices for weld rod selection (and weld procedures as well) or to consult weld rod manufacturers. The latter have up-to-date tables for rod selection. Proper weld rod selection not only insures preservation of the corrosion resistant properties, but it is also important in achieving optimum mechanical properties.

Another principal difference between austenitic stainless and carbon or low-alloy steels is thermal conductivity, with stainless about half that of other steels. Hence, heat is not dissipated as rapidly.

There are four methods to accommodate this situation: lower weld current settings, skip-weld techniques to minimize heat concentration, use of back-up chill bars or other cooling techniques to dissipate heat, and proper joint design. The first three methods fall in the realm of welding shop procedures that are often adequately covered by AWS recommended practices and welding shop standard practices. It is always good policy, however, for designers to double check to see that the welding shop follows proper procedures. Also, it is often desirable to provide specimen welds for establishing quality. These precautions are important because corrosion problems often begin in weld areas. One problem, discussed earlier, is carbide precipitation (sensitization) that can lead to intergranular corrosion in corrosive environments. The lack of proper heat dissipation can also lead to heat distortion of the finished product that can be unacceptable for aesthetic reasons.

From the designer's viewpoint, joint configuration can also encourage heat dissipation. For this reason, the use of beveled joints is common in thinner gages, which in carbon steel might be welded as a square-edge butt. Beveling permits the use of several light passes, thus avoiding the high temperature that would be reached in a single, heavy pass.

Cleaning of the edges to be welded is also important. Contamination from grease and oil can lead to carburization in the weld area with subsequent reduction of corrosion resistance. Post weld clean-up is also important and should not be done with carbon steel files and brushes. Carbon steel cleaning tools, as well as grinding wheels that are used on carbon steel, can leave fine particles imbedded in the stainless steel surface that will later rust and stain if not removed by chemical cleaning.

Martensitic Stainless Steels

There is always the possibility of metallurgical change during cooling, which can lead to cracking. This can be offset by preheating and postheating to reduce the cooling rate. Filler metal for welds can be identical to the base metal or it can be an austenitic stainless steel composition.

Ferritic Stainless Steels

The three major difficulties associated with welding ferritic stainless steels are (1) excessive grain growth, (2) sensitization, and (3) lack of ductility. Heat treating after welding can minimize some of these problems, or one of the proprietary ferritic alloys with lower carbon and nitrogen contents can be specified. Filler metal can be of either a similar composition or an austenitic composition (Types 308, 309, 316L, or 310), which is helpful in improving ductility and toughness.

Austenitic Stainless Steels

II

The 200 and 300 Series are the most weldable of the stainless steels. The problems that arise relate mainly to sensitization in the heat-affected-zone, which can be minimized by using the low-carbon or stabilized grades.

Preheating is not required; postheating is necessary only to redissolve precipitated carbides and to stress relieve components that are to be used in environments that may cause stress corrosion cracking.

The coefficient of expansion of austenitic types is higher than that of carbon steels; hence thermal contraction is greater. Precautions are necessary to avoid bead cracking and minimize distortion, such as sound fixtures, tack welding, skip welding, copper chill bars, minimum heat input, and small weld passes.

Precipitation Hardening Stainless Steels

The precipitation hardening grades are suited to welding with little need for pre- or post-heat treatment except to restore or improve mechanical properties.

Free-Machining Stainless Steels

Problems of porosity and segregation arise when free-machining types are welded. However, special filler rods (Type 312) are available that, with careful exclusion of hydrogen from the weld, will assist welding.

Soldering

Stainless steels are readily soldered with relatively few problems arising from temperature. Aggressive fluxes, however, are necessary to prepare the surface for soldering. Phosphoric acid type fluxes are preferred because they are not corrosive at room temperature.

Brazing

All stainless steels can be brazed, but because the brazing alloys are usually composed of copper, silver, and zinc, high temperatures are required. Care must be taken that the brazing cycle does not cause such high-temperature problems as carbide precipitation and a lessening of corrosion resistance.

Fastening

Although fasteners are available in many materials, stainless fasteners are a good first choice, especially if the materials being joined are stainless. Stainless fasteners are easy to make, in both standard and special designs, and they are readily available.

Since corrosion resistance is an important aspect of product reliability, inherent in any attempt to prevent corrosion is the careful selection of fastener materials. A common practice in industry is to use fasteners made of metals or alloys that are equal to or more corrosion resistant than the materials they join. This practice is justified because the fasteners may have to withstand higher loads with greater unit stress than the parts being held together, and they are usually considerably smaller in surface area than the material being joined. Also, corrosion-weakened fasteners may lead to a more immediate failure with more serious consequences than the same amount of corrosive attack elsewhere in the assembly.

Corrosion protection for a fastened joint encompasses much more than a consideration of the corrosion resistance of the fastener itself. Required is an analysis of the entire assembled joint as a system. This system includes structural design, material stresses, product life expectancy and environmental conditions.

Where two dissimilar metals are in contact in the presence of an electrolyte, a battery effect is created, current flows, and one of the metals corrodes. In considering a bimetallic couple, it is important to know which of the two metals is more anodic (less noble). A guide to this is the arrangement of metals in the galvanic series chart shown in Exhibit 9. Any metal in this series will tend to have corrosion accelerated when it is coupled, in the presence of an electrolyte, with a metal in a lower position on the chart. The corrosion of this lower metal will tend to be reduced, or even avoided.

A very important factor to consider in evaluating the potential for galvanic corrosion is the relative surface area of the two different metals in contact. For example, carbon steel is located above stainless steel in the galvanic series and is accordingly subject to accelerated corrosion when a galvanic couple is established. But the extent of this galvanic action depends on the relative surface area of each material. For instance, if small steel fasteners, such as rivets, are used to join stainless steel plates, and the assembly is exposed to water, the steel rivets will corrode quickly. If, on the other hand, stainless rivets are used to join steel plates in water, both rivets and plates will suffer negligible galvanic attack, even in the immediate vicinity of the rivets. Aircraft designers, for instance, who specify stainless steel fasteners in aluminum structures depend upon this area-relationship principle.

When the designer has determined candidate fastener materials on the basis of their corrosion-resistant properties, his next concern probably will be the mechanical and physical properties of these materials. Once again, the group of stainless steels covers a wide choice. The choice need not be difficult if the designer uses the guidelines available to him, such as the specifications published by the Industrial Fasteners Institute (IFI).

Data on stainless steel fasteners are available from NiDI in the booklet, "Stainless Steel Fasteners, A Systematic Approach to Their Selection."

Exhibit 9

Galvanic Series of Metals and Alloys in Sea Water (14)

Magnesium	Anodic ↑
Zinc	More Likely to Be
Alclad 3003	Attacked
Aluminum 3003	
Aluminum 6061	
Aluminum 6063	
Aluminum 5052	
Mild Steel	
Low Alloy Steel	
Cast Iron	
Stainless Steels (Active)	
Muntz Metal	
Yellow Brass	
Red Brass	
Copper	
Aluminum Bronze	
Silver	
Stainless Type 430 (Passive)	
Stainless Type 304 (Passive)	
Stainless Type 316 (Passive)	
Monel	More Noble ↓
Gold	Cathodic

II

SURFACE PROTECTION AND CLEANING

Stainless steels must have clean surfaces to offer optimum corrosion resistance. Design engineers should take steps to see that fabricators either protect the metal surface from contamination during forming or other manufacturing steps or restore the surface by mechanical or chemical cleaning.

Metal particles from steel dies can become imbedded in the surface of the stainless at pressure points. This pickup will rust and stain the surface when exposed to moisture. Chromium plated dies, or a paper or plastic protective covering on the stainless steel being formed, can prevent such problems during many fabrication steps, except for the more severe operations, such as forging, machining, heading, coining, drawing, welding, or spinning.

In the case of protective coverings, a number of materials are available. Such coverings should be selected on the basis of their ability to remain intact during layout and fabrication, and on the basis of their ability to be removed readily.

When stainless cannot be protected by covering, procedures should be employed to keep the material clean. Rusty water drips, dirt from overhead cranes, unclean handling equipment, even dust from open doors, can be sources of staining. Perhaps the most severe problems arise in shops that work on carbon steel as well as stainless. Using grinding tools on stainless that were previously used on carbon steel can leave particles on the stainless surface that will later rust and stain. In such cases, the best procedure is to chemically clean the stainless after fabrication, in solutions that will dissolve the carbon steel particles, such as solutions of nitric acid and water.

From a practical standpoint, the composition of the cleaning bath is not important as long as it serves the function of chemically cleaning the surface without harm or discoloration. Procedures and makeup of solutions are widely published or are available from companies listed on the back cover of this booklet.

It is also important to see that stainless components are thoroughly clean prior to heat treatment. Lubricants, grease, or foreign particles can burn into the steel surface during heating, which can increase the cost of final cleaning or, in extreme cases, render the parts unusable, if appearance is a vital factor.

When stainless steels are heated for forging, annealing, or hardening, an oxidized scale forms on the surface. If not removed, the scale lowers corrosion resistance and can interfere with subsequent operations. Scale can be removed mechanically by glass bead blasting or tumbling, or by chemical pickling. The type or degree of scaling determines the method of cleaning. Frequently, both glass bead blasting and chemical clearing are used. Composition of pickling baths vary widely so fabricators are encouraged to seek advice from stainless steel producers.

Note: ASTM A380 is a good reference on cleaning procedures and surface treatments for stainless steel.

Pitting occurs when the protective film breaks down in small isolated spots, such as when halide salts contact the surface. Once started, the attack may accelerate because of differences in electric potential between the large area of passive surface vs. the active pit.

Pitting is avoided in many environments by using Types 316 and 317, which contain molybdenum. The four alloys in Table I, plus Type 317, performed well in desalination environments during a three-year test conducted by the Committee of Stainless Steel Producers in Freeport, Texas.

Table I

ALLOY	UNS
Alloy 6X (20Cr-23Ni-6Mo)	N08366
Alloy 216 (19.75Cr-6Ni-8.25Mn - 2.5Mo-0.37N)	S21600
Nitronic 50® (22Cr-13Ni-5Mn-2.25Mo)	S20910
®Registered Trademark of Armco Steel Corporation	
Alloy 20Cb-3® (20Cr-33Ni-2.5Mo-3.5Cu)	N08020
®Registered Trademark of Carpenter Technology	

Crevice Corrosion results from local differences in oxygen concentration associated with deposits on the metal surface, gaskets, lap joints, or crevices under bolt or rivet heads where small amounts of liquid can collect and become stagnant. The material responsible for the formation of a crevice need not be metallic. Wood, plastics, rubber, glass, concrete, asbestos, wax, and living organisms have all been reported to cause crevice corrosion. Once attack begins within the crevice, its progress is very rapid, especially in chloride environments. For this reason, the stainless steels containing molybdenum are often used to minimize the problem. Notwithstanding, the best solution to crevice corrosion is a design that eliminates crevices.

Stress Corrosion Cracking is caused by the combined effects of tensile stress and corrosion. Many alloys systems have been known to experience stress corrosion cracking – brass in ammonia, carbon steel in nitrate solutions, titanium in methanol, aluminum in sea water, and gold in acetic acid. Stainless steels are susceptible to stress corrosion cracking in chloride environments. It is necessary for tensile stress, chlorides and elevated temperature all to be present for stress corrosion cracking to occur. Wet-dry or heat transfer conditions, which promote the concentration of chlorides, are particularly aggressive with respect to initiating stress corrosion cracking. While the mechanism of stress corrosion cracking is not fully understood, laboratory tests and service experience have resulted in methods to minimize the problem. For instance, alloy 2205 (a duplex stainless containing 21-23% chromium, 4.5-6.5% nickel, and 2.5-3.5% molybdenum plus nitrogen) exhibits superior resistance to chloride stress corrosion cracking; plus it has a general corrosion and pitting resistance similar to Type 317. Studies by Climax Molybdenum Company indicate that Type 317 with 3.5% (minimum) molybdenum has excellent resistance, and it has been shown to perform well in a flue-gas desulfurization environment. Several proprietary austenitic stainless steels also have shown resistance to stress cracking in hot chloride environments.

The ferritic stainless steels, such as Types 405 and 430, should also be considered when the potential exists for stress-corrosion cracking.

The corrosion resistance of ferritic stainless steels is improved by increased chromium and molybdenum contents, while ductility, toughness, and weldability are improved by reducing carbon and nitrogen contents. The commercialization of new melting and refining processes has resulted in several new ferritic stainless steels with improved characteristics, which can be classified as follows; those with about 18% chromium having corrosion resistance similar to Type 304, and those with more than 18% chromium with resistance to corrosion comparable or superior to Type 316 in many media. With two exceptions (439, 444), these ferritic stainless steels are not AISI numbered grades. Some of these stainless steels are listed in Table II.

The high-chromium ferritic types have resistance to chlorides previously obtainable only in high-nickel and titanium alloys.

Table II

ALLOY	ASTM	UNS
18 Cr and Ti	439	S43035
18 Cr - 2 Mo	444	S44400
18 Cr - 2 Mo + S	XM-34	S18200
26 Cr - 1 Mo	XM-27	S44625
26 Cr - 1 Mo - Ti	XM-33	S44626
29 Cr - 4 Mo		S44700
29 Cr - 4 Mo - Ti/Cb		S44735
29 Cr - 4 Mo - 2 Ni		S44800

Other Corrosion Types should be considered when using stainless steels, such as corrosion fatigue, delayed brittle fracture and hydrogen stress cracking. Corrosion fatigue is encountered in cyclic loading in a corrosive environment. Brittle fracture is caused by hydrogen impregnation of an alloy during processing, which leads to brittle failure when subsequently loaded. Hydrogen stress cracking results from a cathodic reaction in service.

The austenitic stainless steels resist hydrogen effects, but martensitic and precipitation-hardening alloys may be susceptible to both hydrogen stress cracking and chloride stress-corrosion cracking.

Sulfide ions, selenium, phosphorus and arsenic compounds increase the propensity for hydrogen to enter hardenable stainless steels and cause hydrogen stress cracking. Their presence should warn of a failure possibility. Cathodic protection can also cause hydrogen stress cracking of high-strength alloys in service, if “overprotected.” Therefore, cathodic protection, or coupling hardenable stainless steels to less-noble materials in corrosive environments should be used with caution.

Excessive exposure of duplex stainless steels to 1300 to 1750ºF (700 to 955ºC) tends to form intermetallic compounds such as sigma phase or chi phase. These compounds of iron, chromium, and molybdenum are highly detrimental to corrosion resistance and toughness.

Intergranular Corrosion

When austenitic stainless steels are heated or cooled through the temperature range of about 800-1650ºF (427-899ºC), the chromium along grain boundaries tends to combine with carbon to form chromium carbides. Called carbide precipitation, or sensitization, the effect is a depletion of chromium and the lowering of corrosion resistance in areas adjacent to the grain boundary. This is a time temperature dependent phenomenon, as indicated in Figure 4 (See Appendix B).

Sensitization may result from slow cooling from annealing temperatures, stress-relieving in the sensitization range, or welding. Due to the longer time at temperature of annealing or stress-relieving, it is possible that the entire piece of material will be sensitized, whereas the shorter time at temperature characteristic of welding can result in sensitization of a band, usually 1/8 to 1/4 inch wide, adjacent to but slightly removed from the weld. This region is known as the Heat-Affected-Zone or HAZ.

Intergranular corrosion depends upon the magnitude of the sensitization and the aggressiveness of the environment to which the sensitized material is exposed. Many environments do not cause intergranular corrosion in sensitized austenitic stainless steel. For example, glacial acetic acid at room temperature or fresh clean water do not; strong nitric acids do. Carbide precipitation and subsequent intergranular corrosion in austenitic stainless steels have been thoroughly investigated; the causes are understood and methods of prevention have been devised. These methods include:

1. Use of stainless steel in the annealed condition.

2. Selection of the low-carbon (0.030% maximum) stainless steels for weld fabrication. Low-carbon grades are Types 304L, 316L, and 317L. The less carbon available to combine with the chromium, the less likely is carbide precipitation to occur. However, the low-carbon grades may become sensitized at extremely long exposures to temperatures in the sensitization range.

3. Selection of a stabilized grade, such as Type 321 (titanium stabilized) or Type 347 (columbium stabilized), for service in the 800-1650ºF (427-899ºC) range. The protection obtained with these grades is based upon the greater affinity of titanium and columbium for carbon as compared to chromium.

Columbium stabilization is preferred because its carbides are more readily retained in welds and it is easier to add in the steelmaking process. However, the use of columbium stabilized steel requires additional care in welding.

4. Redissolving carbides by annealing parts after fabrication, although this is not always practical.

It should be understood that the above steps are necessary only if the service environment is known to be capable of causing intergranular corrosion.

Although sensitization can also occur in the ferritic stainless steels (heated to 1700ºF (927ºC) and water quenched or air cooled) it is far less likely to occur than with austenitic grades, and intergranular corrosion has not been a problem in these steels–except for a narrow band in the heat-affected-zone close to welds. (Most of the proprietary ferritic stainless steels mentioned earlier are stabilized to prevent sensitization during welding.) Galvanic corrosion is discussed briefly in the section on fasteners, page 29.

HIGH TEMPERATURE CORROSION RESISTANCE

Stainless steels have been widely used for elevated-temperature service, so fundamental and practical data concerning their resistance to corrosion are available.

When stainless steels are exposed at elevated temperatures, changes can occur in the nature of the surface film. For example, at mildly elevated temperatures in an oxidizing gas, a protective oxide film is formed.

In more aggressive environments, with temperatures above 1600ºF (871ºC), the surface film may break down with sudden increase in scaling. Depending on alloy content and environment, the film may be self healing for a period of time followed by another breakdown.

Under extreme conditions of high temperature and corrosion, the surface film may not be protective at all.

For these reasons, the following data should serve only as a starting point for material selection, not as a substitute for service tests.

Oxidation

In nonfluctuating-temperature service, the oxidation resistance (or scaling resistance) of stainless steels depends on chromium content, as indicated by the curve in Figure 5. Steels with less than 18% chromium (ferritic grades primarily) are limited to temperatures below 1500ºF (816ºC). Those containing 18-20% chromium are useful to temperatures of 1800ºF (982ºC), while adequate resistance to scaling at temperatures up to 2000ºF (1093ºC) requires a chromium content of at least 22%, such as Types 309, 310 or 446.

The maximum service temperature based on a rate of oxidation of 10 mg. per sq. cm. in 1000 hours is given for several stainless steels in Table III for nonfluctuating-temperature. The corrosion resistance of several stainless steels in steam and oxidizing flue gases, compared with their corrosion resistance in air, is shown in Figure 6.

In many processes, isothermal (constant temperature) conditions are not maintained and process temperatures vary. The temperature limits under variable conditions are shown in Table III in the column “Intermittent Service.” Expansion and contraction differences between the base metal and the protective film (or scale) during heating and cooling cause cracking and spalling of the protective scale. This allows the oxidizing media to attack the exposed metal surface.

The spalling resistance of the austenitic stainless steels is greatly improved at higher nickel levels, as illustrated in Figure 7. Nickel reduces the thermal expansion differential between alloy and oxide film and thereby reducing stresses at the alloy-oxide interface during cooling. Also, Type 446 and the proprietary ferritic chromium-molybdenum stainless steels have a fairly low coefficient of thermal expansion, which tends to enhance spalling resistance.

A number of proprietary austenitic stainless steels that rely on silicon, aluminum, or cerium additions for improved oxidation resistance are listed in ASTM A240 and other product specifications.

Effect of Atmosphere

Much attention has been given to the compatibility of stainless steels with air or oxygen. However, trends in the design of steam and other forms of power generation have resulted in a growing interest in oxidation in such environments as carbon monoxide, carbon dioxide, and water vapor. Exposure to mild conditions in these environments leads to the formation of the protective oxide film described earlier, but when conditions become too severe, film breakdown can occur. The onset of this transition is unpredictable and sensitive to alloy composition.

Although the reaction mechanisms are probably similar in air, oxygen, water vapor, and carbon dioxide, reaction rates may vary considerably. For example, similar scaling behavior has been observed in air and oxygen except that scale breakdown occurs more rapidly in oxygen. For this reason, results obtained in air should be applied with care when considering service in pure oxygen.

An increase in corrosion rates can be expected in the presence of water vapor. Figure 8 illustrates the effect of moist air on the oxidation of Types 302 and 330. Type 302 undergoes rapid corrosion in wet air at 2000ºF (1093ºC), whereas a protective film is formed in dry air. The higher nickel Type 330 is less sensitive to the effects of moisture, so it is assumed that increased chromium and nickel permits

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higher operating temperatures in moist air. Types 309 and 310 are superior at temperatures greater than 1800ºF (982ºC), and Type 446 is usable at temperatures approaching 2000ºF (1093ºC). The addition of moisture to oxygen significantly increases the corrosion rates of Types 304, 321, 316 and 347, and for the other grades listed in Table III, the temperature limits should be adjusted downwards.

Table III
Suggested Maximum Service Temperatures in Air

AISI Type	Intermittent Service		Continuous Service	
	ºC	ºF	ºC	ºF
201	815	1500	845	1550
202	815	1500	845	1550
301	840	1550	900	1650
302	870	1600	925	1700
304	870	1600	925	1700
308	925	1700	980	1800
309	980	1800	1095	2000
310	1035	1900	1150	2100
316	870	1600	925	1700
317	870	1600	925	1700
321	870	1600	925	1700
330	1035	1900	1150	2100
347	870	1600	925	1700
410	815	1500	705	1300
416	760	1400	675	1250
420	735	1350	620	1150
440	815	1500	760	1400
405	815	1500	705	1300
430	870	1600	815	1500
442	1035	1900	980	1800
446	1175	2150	1095	2000

It is difficult to indicate maximum service temperatures for steam service, one reason being the sensitivity of corrosion rate to surface condition. (Cold worked surfaces tend to reduce corrosion effects in steam service.) Most austenitic stainless steels can be used at temperatures up to 1600ºF (871ºC), and Types 309, 310, and 446 at higher temperatures. Types 304, 321, and 347 are being used in low-pressure steam systems at temperatures approaching 1400ºF (760ºC). Scale on Types 304, 347, and 316 tends to exfoliate at higher temperatures.

The oxidation of stainless steels in carbon dioxide and carbon dioxide-carbon monoxide atmospheres at 1100-1800ºF (593-982ºC) is of interest because of their use in gas-cooled nuclear reactors. Type 304 is serviceable in this environment, although some proprietary stainless steels offer better resistance.

A note of caution about stainless steels at high temperatures in *stagnant* oxidizing environments: The protective film breaks down in the presence of certain metal oxides, causing accelerated attack. For instance, austenitic types are susceptible to attack in the presence of lead oxide at temperatures as low as 1300ºF (704ºC). Vanadium oxide, found in fuel ash, may cause failure of Types 309 and 310 at 1900ºF (1038ºC) when water vapor is present. Molybdenum oxide behaves in a similar manner.

Sulfidation

Sulfur in various forms and even in relatively small quantities accelerates corrosion in many environments. Sulfur dioxide, hydrogen sulfide, and sulfur vapor are among the most corrosive forms. Sulfur vapor and hydrogen sulfide are considerably more aggressive than sulfur dioxide.

Sulfur attack, although closely related to oxidation, is more severe. Metal sulfides melt at lower temperatures than comparable oxides, and they may fuse to metal surfaces. Also, sulfides are less likely to form tenacious, continuous, protective films. Fusion and lack of adherence result in accelerated corrosion.

The resistance of stainless steels to sulfidation depends on chromium content.

Sulfur Dioxide

Type 316, in a series of 24-hour laboratory tests, was subjected to mixtures of oxygen and sulfur dioxide (varying from 100% oxygen to 100% sulfur dioxide) at temperatures between 1100 and 1600ºF (593 and 871ºC). Results indicated that the rate of attack was largely independent of the gas composition, and no scale developed–only a heavy tarnish.

Hydrogen Sulfide

Low chromium steels are adequate to resist attack in relatively low hydrogen sulfide levels, but hydrogen sulfide under high pressure results in rapid corrosion. Then a minimum of about 17% chromium is required to obtain satisfactory resistance. Type 304 has been used extensively for this service. The iso-corrosion curves shown in Figure 9 show the effects of hydrogen sulfide and temperature on the austenitic stainless steels.

Sulfur Vapor

Sulfur vapor readily attacks the austenitic grades. In tests, relatively high corrosion rates were encountered in flowing sulfur vapor at 1060ºF (571ºC), although it has been reported that Type 310 has been successfully used for a sulfur vapor line at 900ºF (482ºC).

In liquid sulfur, most austenitic grades are resistant up to 400ºF (204ºC), with the stabilized Types 321 and 347 showing satisfactory service to 832ºF (444ºC).

Flue Gas

The corrosivity of flue gas containing sulfur dioxide or hydrogen sulfide is similar to that of most sulfur-bearing gases. Accordingly, the corrosion resistance of stainless steels in flue gas environments is improved by increased chromium content, as shown in Figure 10. Table IV indicates the effect of chromium content on corrosion in various fuel sources. Corrosion rates of 1 to 2 mils per year have been reported for Types 304, 321, 347 and 316 in the temperature range 1200-1400ºF (649-760ºC).

For reducing flue-gas environments, satisfactory material selection requires service tests.

Other High-Temperature Environments

Data are available on the corrosion resistance of stainless steels in other high-temperature environments, such as their use for liquid-metal environments. Designers are referred to the following publications for additional data on high-temperature applications: *Selection of Stainless Steels,* ASM Engineering Bookshelf and *Corrosion Resistance of the Austenitic Stainless Steels in High-Temperature Environments,* by The Nickel Development Institute.

Table IV
CORROSION RATES OF STAINLESS STEELS IN FLUE GASES
(EXPOSURE 3 MONTHS) (16)

Material AISI Type	Corrosion Rate					
	Coke Oven Gas (1500 ºF) (816 ºC)		Coke Oven Gas (1800 ºF) (982 ºC)		Natural Gas (1500 ºF) (816 ºC)	
	mpy	mmpy	mpy	mmpy	mpy	mmpy
430	91	2.31	236†	6.00	12	0.30
446(26 Cr)	30	0.76	40	1.02	4	0.10
446(28 Cr)	27	0.69	14	0.36	3	0.08
302B	104	2.64	225†	6.00	–	–
309S	37*	0.94	45	1.14	3	0.08
310S	38*	0.97	25	0.64	3	0.08
314	23*	0.58	94	2.39	3	0.08

*Pitted specimens–average pit depth. † Specimens destroyed.

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Figure 1
Effect of Chromium Content on Corrosion Rate (2)

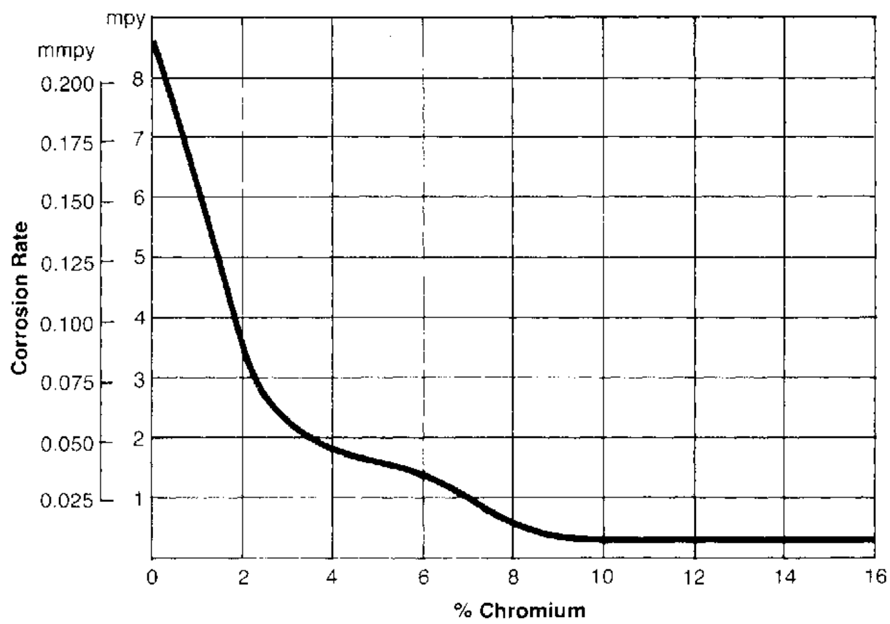
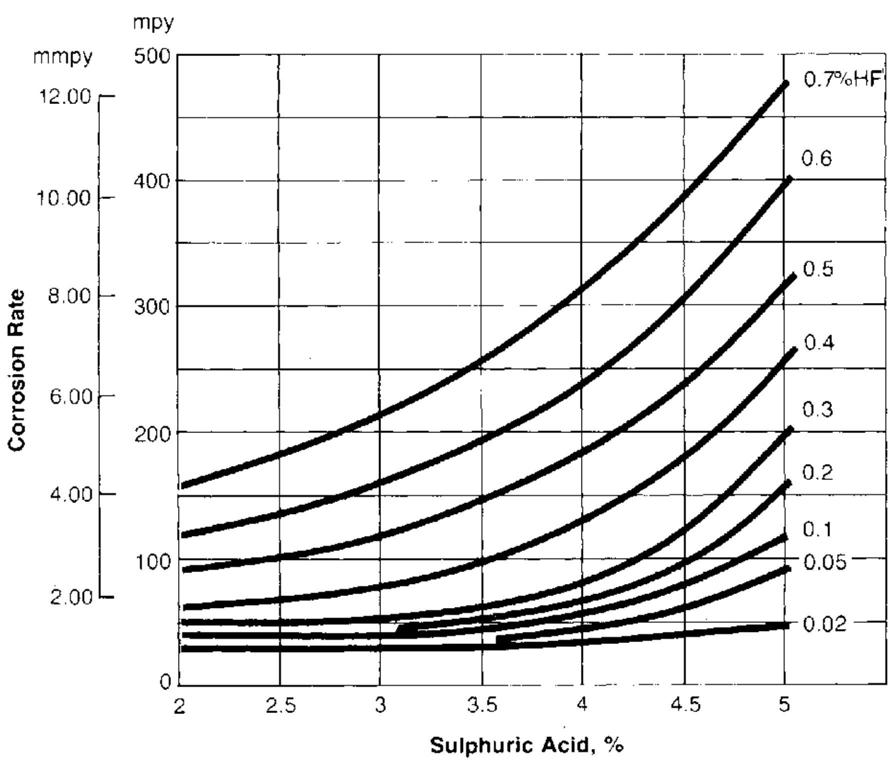


Figure 2
Effect of Chemical Content on Corrosion Rate (3)



Small quantities of hydrofluoric and sulfuric acids can have a serious effect on Type 316 stainless in 25% P_2O_5 phosphoric acid with 1.5% F as H_2SiF_6 at 190°F (90°C).

Figure 3
Effect of Temperature on Corrosion Rate (4)

93% H_2SO_4 with Velocity of 0.1 foot/second

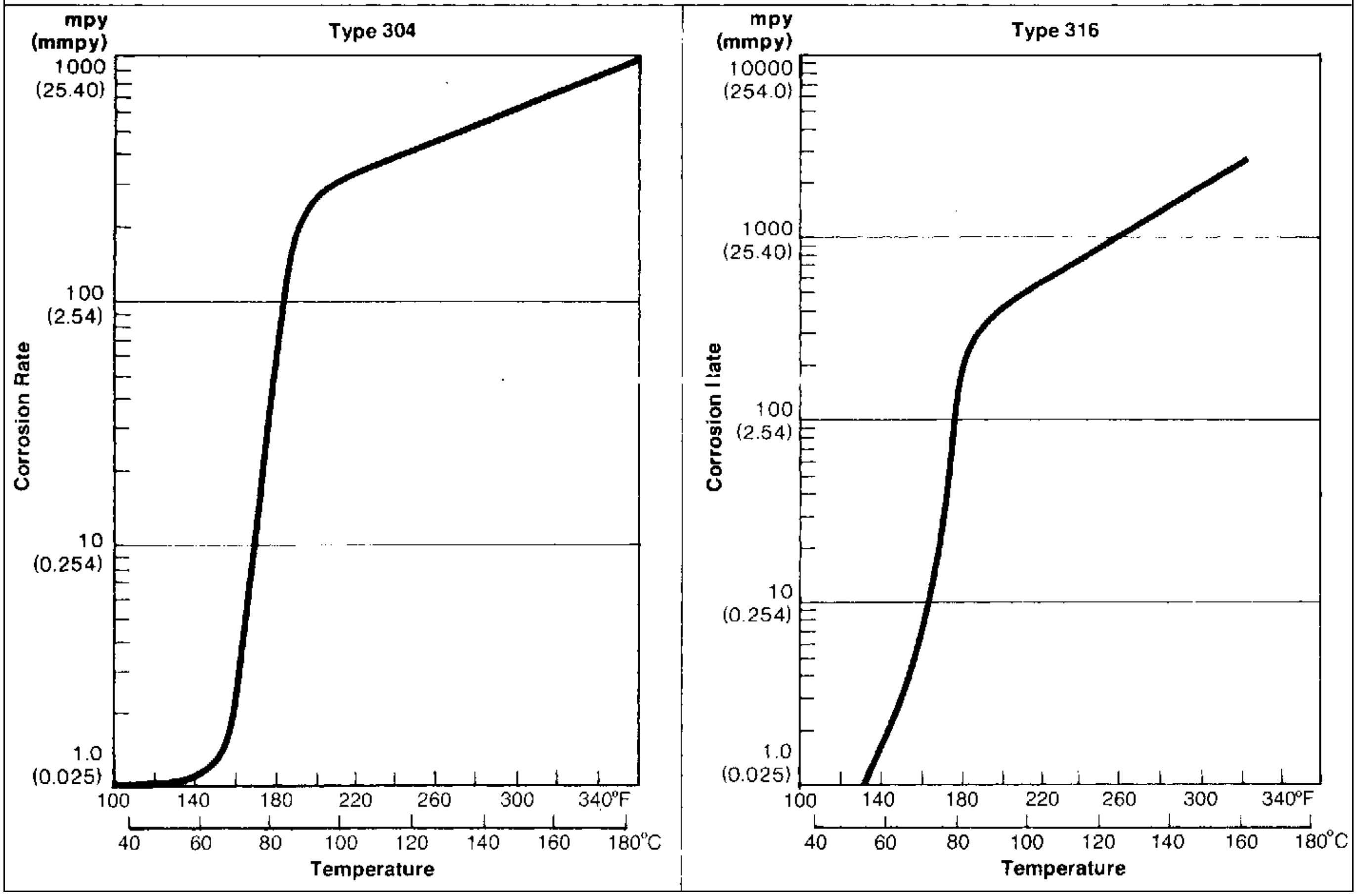


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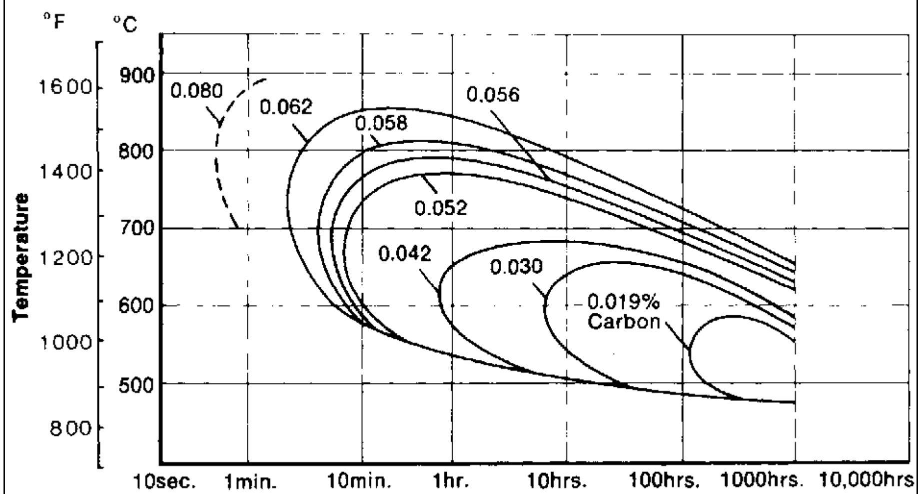
Figure 4

Effect of Carbon Content on Carbide Precipitation (5)



Time-Temperature-Sensitization Curves

Time required for formation of carbide precipitation in stainless steels with various carbon contents. Carbide precipitation forms in the areas to the right of the various carbon-content curves. Within time-periods applicable to welding, chromium-nickel stainless steels with 0.05% carbon would be quite free from grain boundary precipitation. (7)

Figure 5

Effect of Chromium Content on Scaling Resistance (At 1800°F or 982°C) (2)

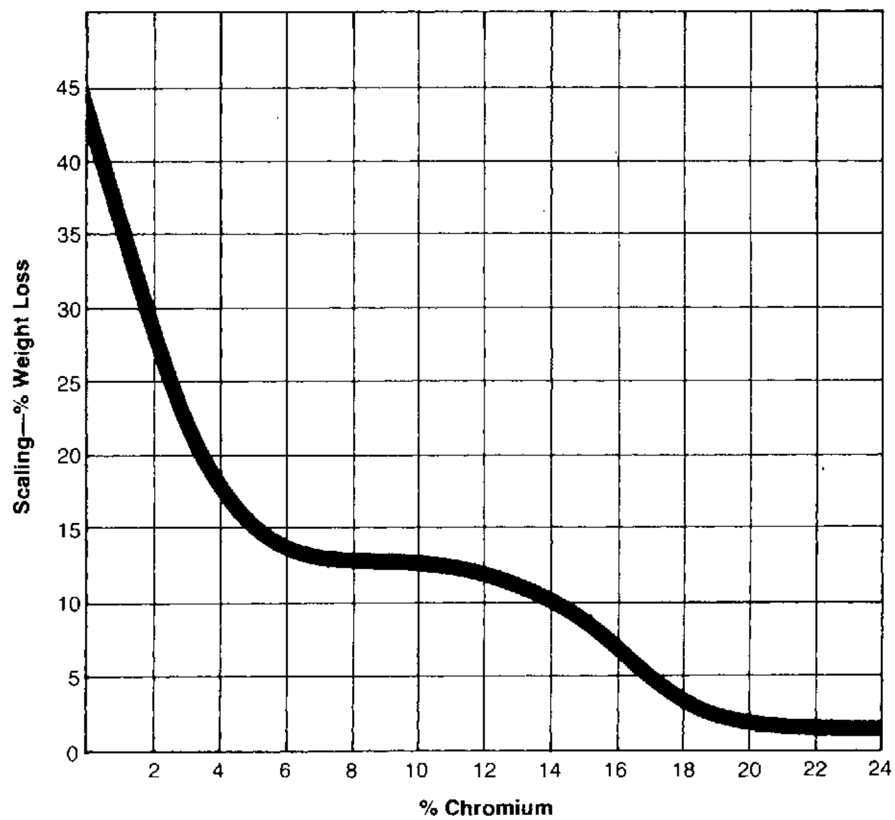
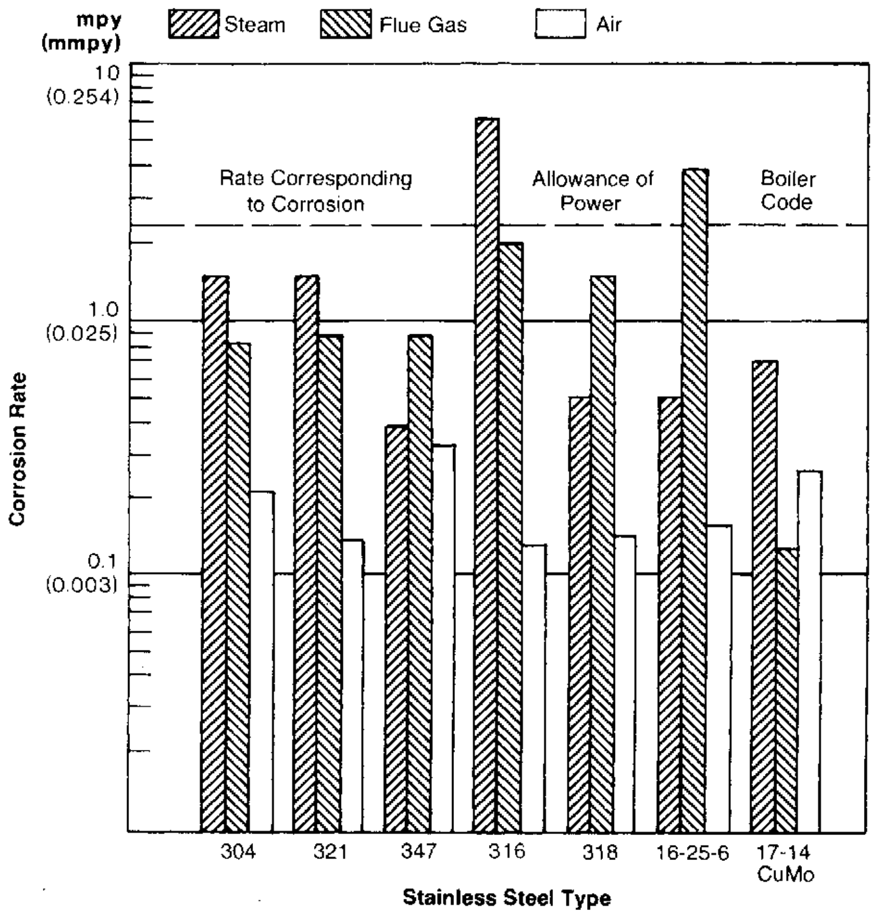


Figure 6

Corrosion Rates in Various Gases (7)



Comparative corrosion rates of stainless steels in steam at 1250°F (677°C), flue gas at 1200 to 1400°F (649 to 760°C), and air at 1400°F (760°C). (Exposure time was 6950 hours for steam and flue gas, 1260 hours for air.)

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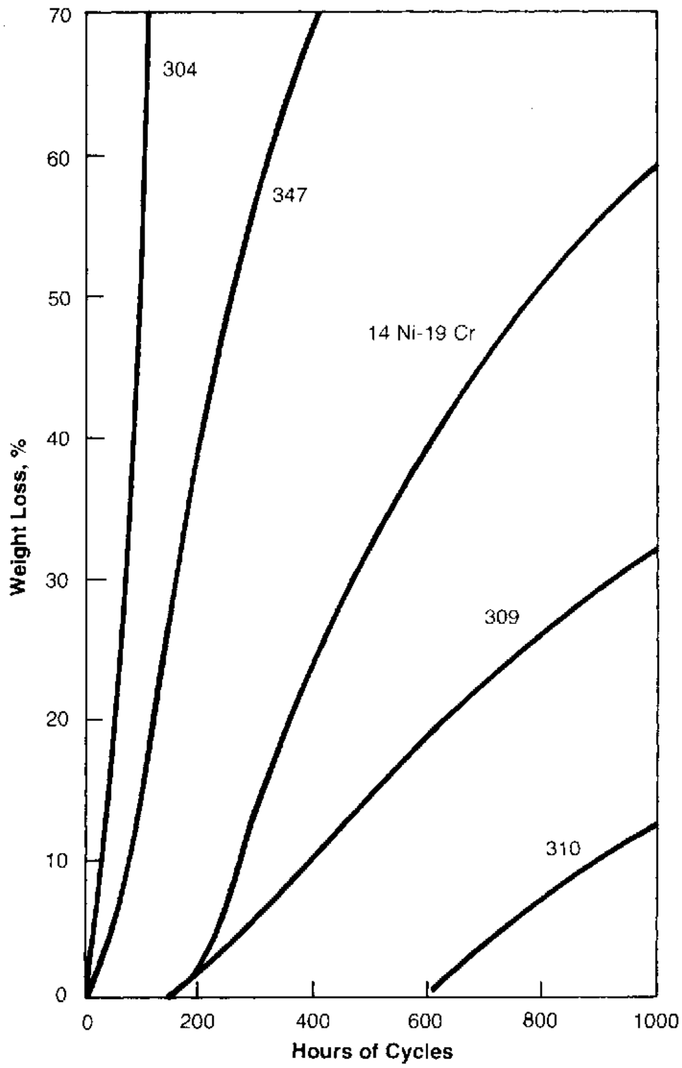
Figure 7

Effect of Nickel (Cr, Cb) on Scaling Resistance (2)



Scaling resistance of some iron-chromium-nickel alloys in cycling-temperature conditions at 1800°F (982°C). Cycle consisted of 15 min. in the furnace and 5 min. in air. Sheet specimens 0.031 in. (0.787 mm) thick were exposed on both sides.

Figure 8

Oxidation of Types 302 and 330 in Wet and Dry Air (8)

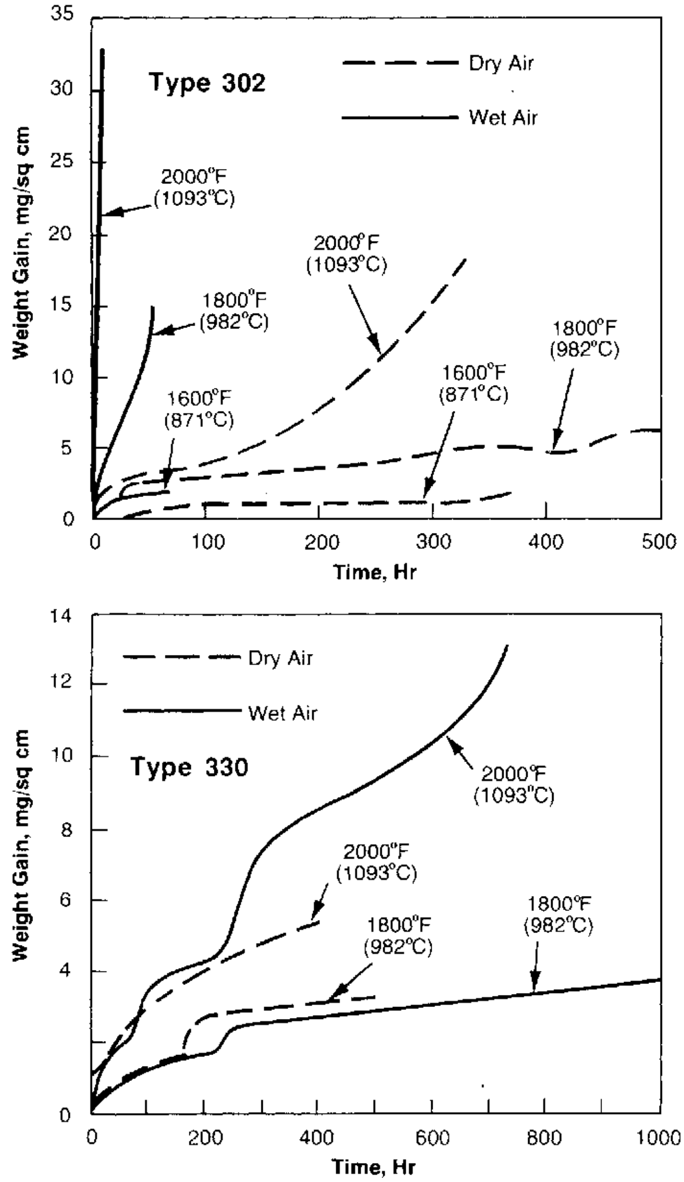
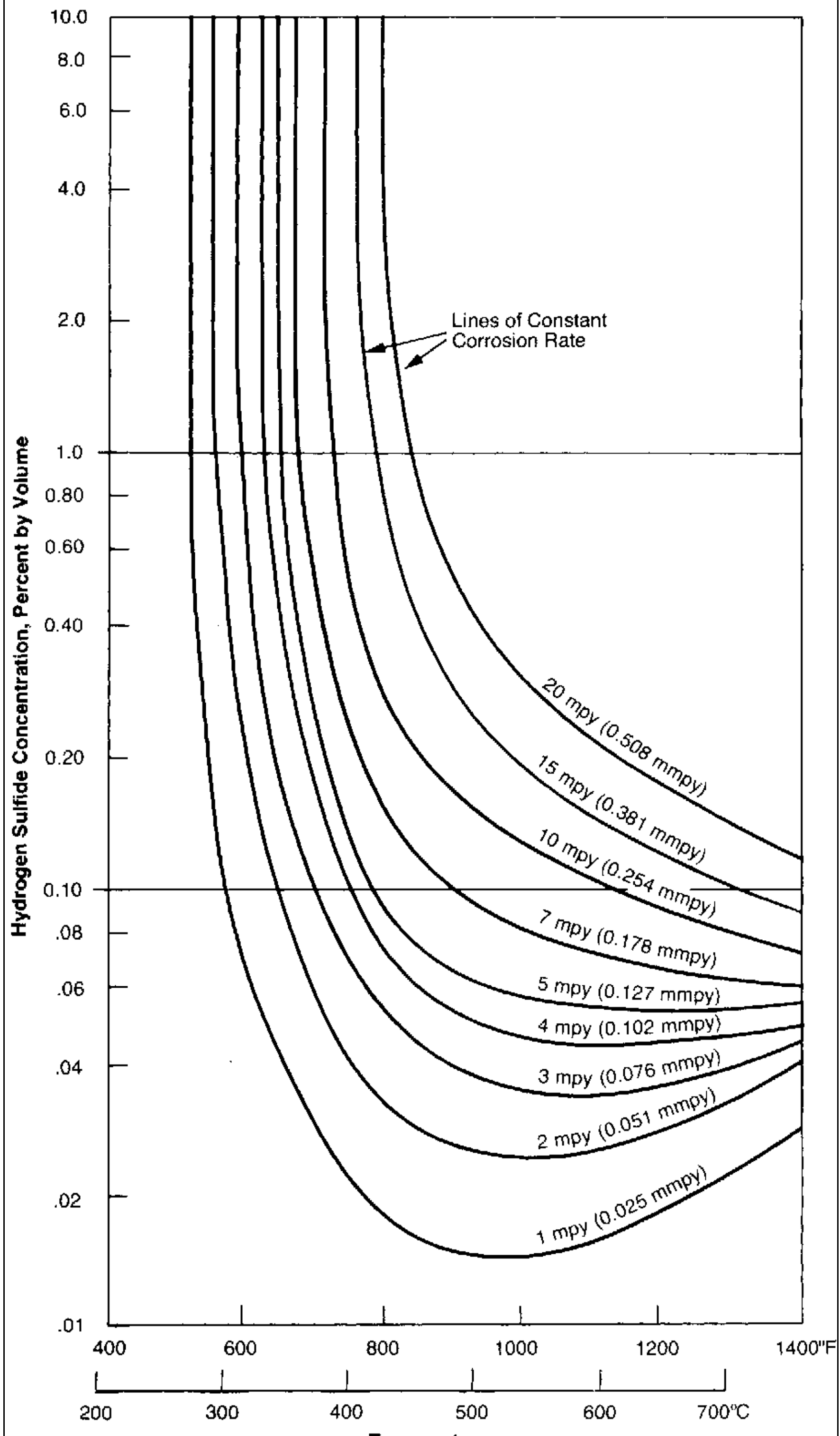


Figure 9

Effect of Temperature and H_2S on Corrosion Rate (9)



Effect of temperature and hydrogen sulfide concentration on corrosion rate of chromium-nickel austenitic stainless steels in hydrogen atmospheres at 175 to 500 psig (1.21-3.45 MPa). (Exposure time greater than 150 hr.)

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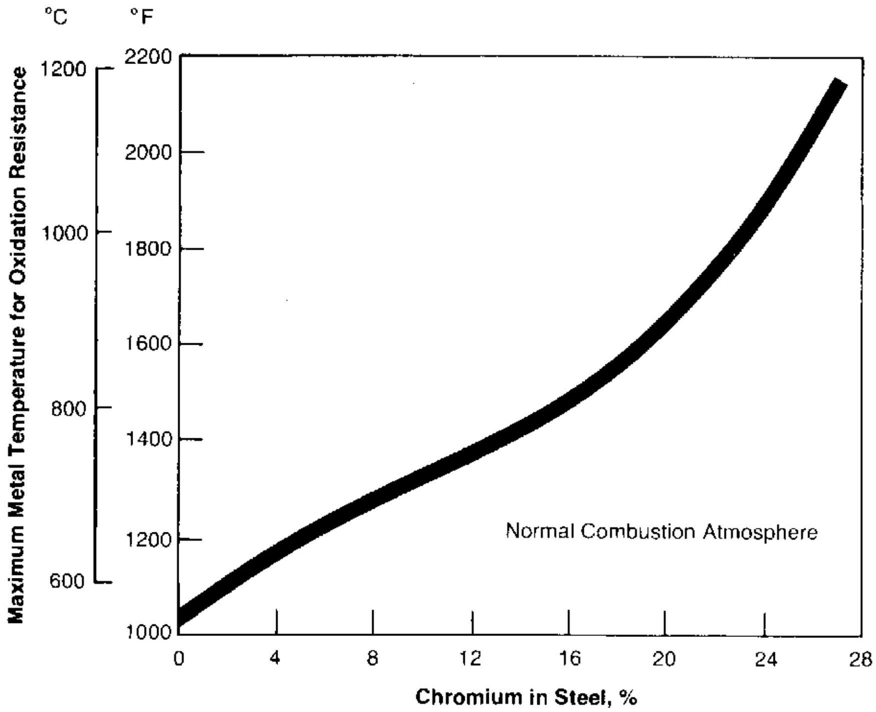
Figure 10
Effect of Chromium in Normal Combustion Atmosphere (10)



Effect of chromium on the oxidation resistance of steel in a normal combustion atmosphere

Figure 11
Stress Strain Curves for Type 304 and Type 301 (2)

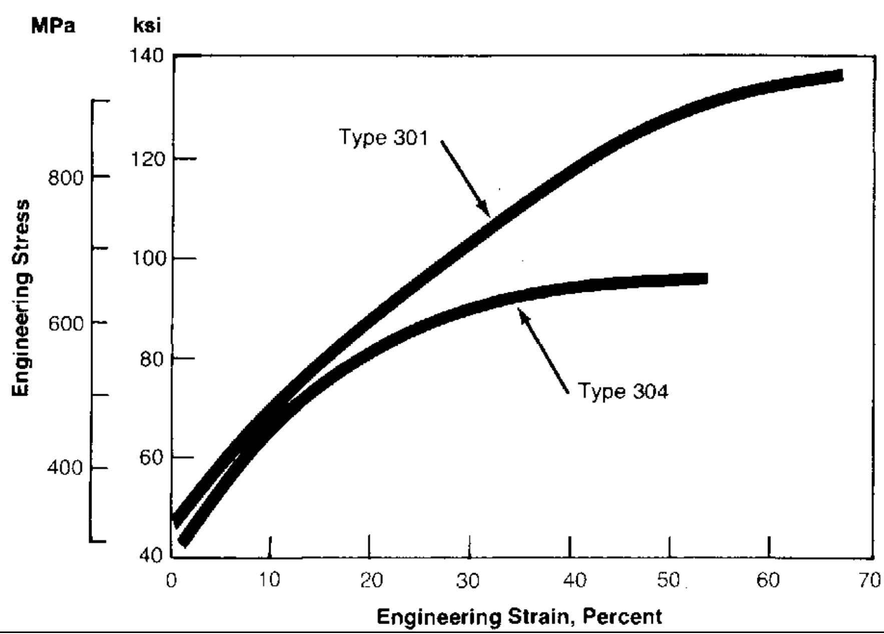


Figure 12
Effect of Cold Work on Mechanical Properties of Type 202 (2)

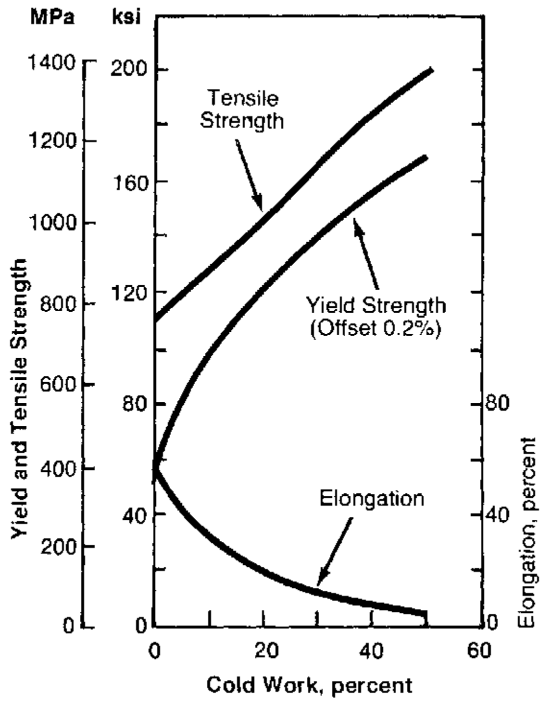


Figure 13
Effect of Cold Work on Mechanical Properties of Type 301 (2)

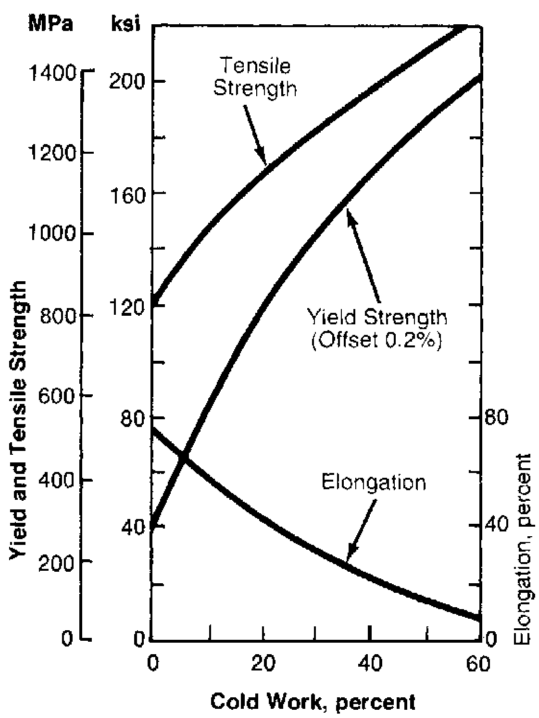


Figure 14
Effect of Cold Work on Mechanical Properties of Type 305 (2)

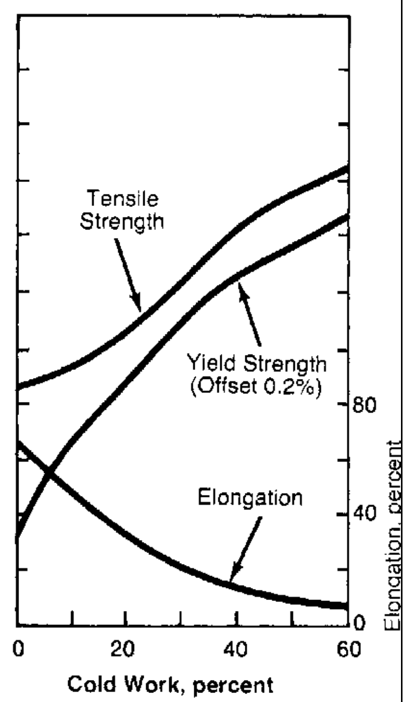


Figure 15
Effect of Cold Work on Mechanical Properties of Type 310 (2)

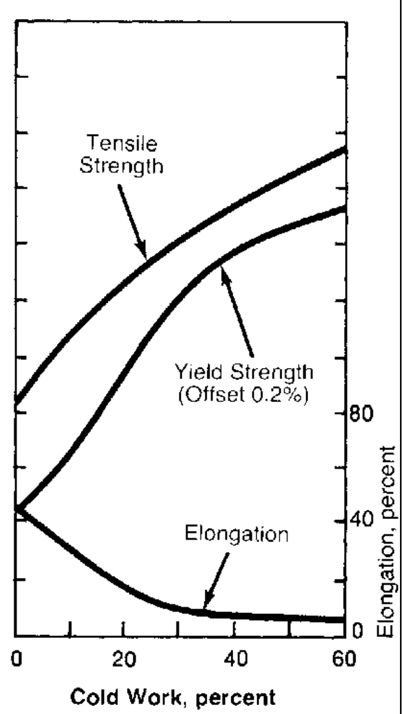


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Figure 16

Effect of Cold Rolling and Test Direction on Notch Strength of Type 301 Sheet (2)

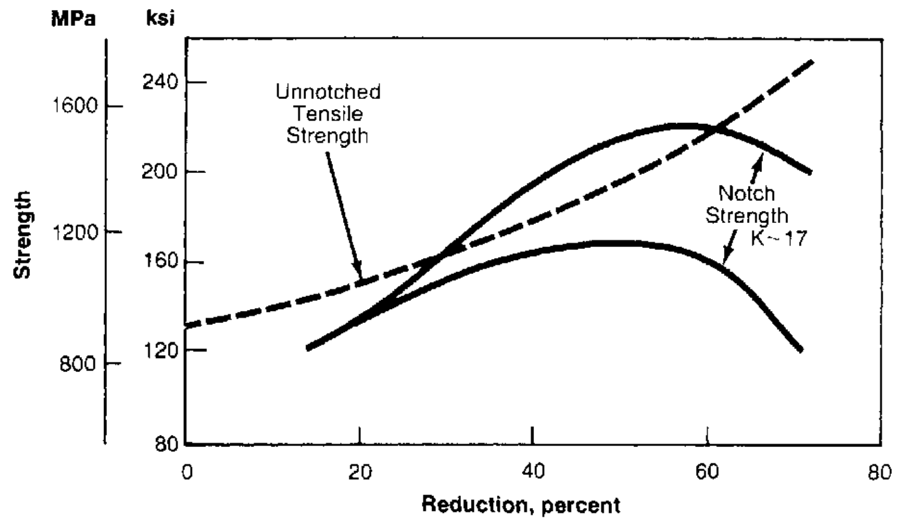

Figure 17

Representative Stress-Strain Curve for Half-Hard Stainless Steel and Mild Steel (2)

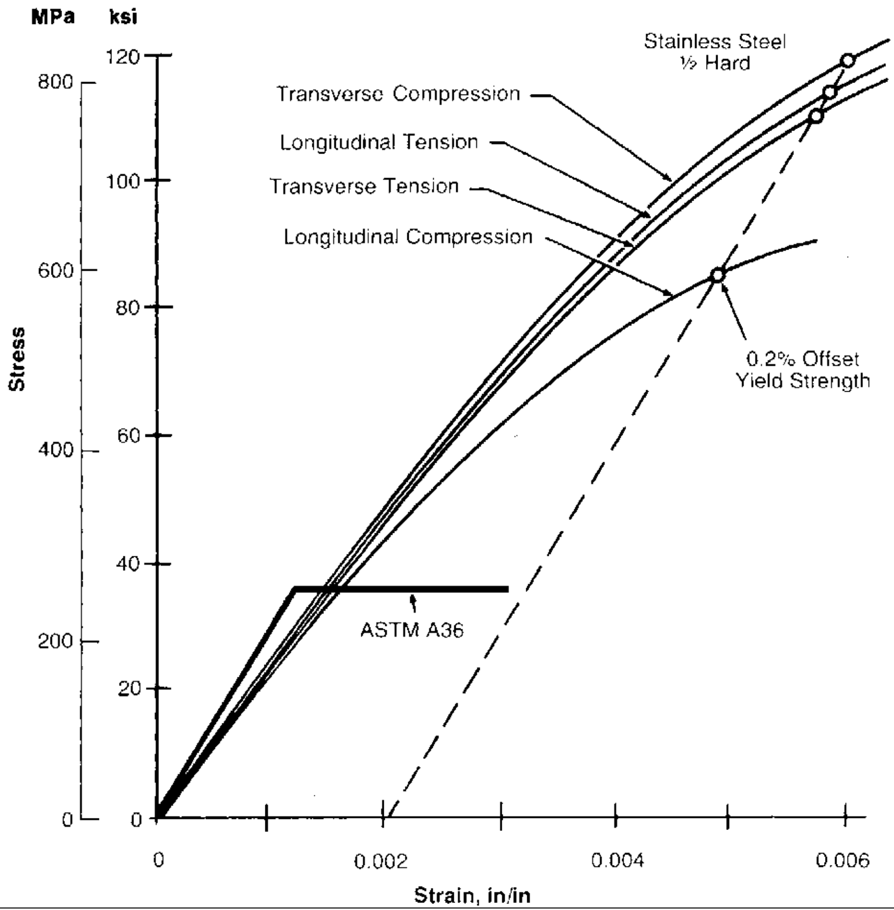

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Figure 18

Izod Impact Values for Type 410 Quenched from 1800°F (982°C) and Tempered for 3 Hours at Indicated Temperature (2)

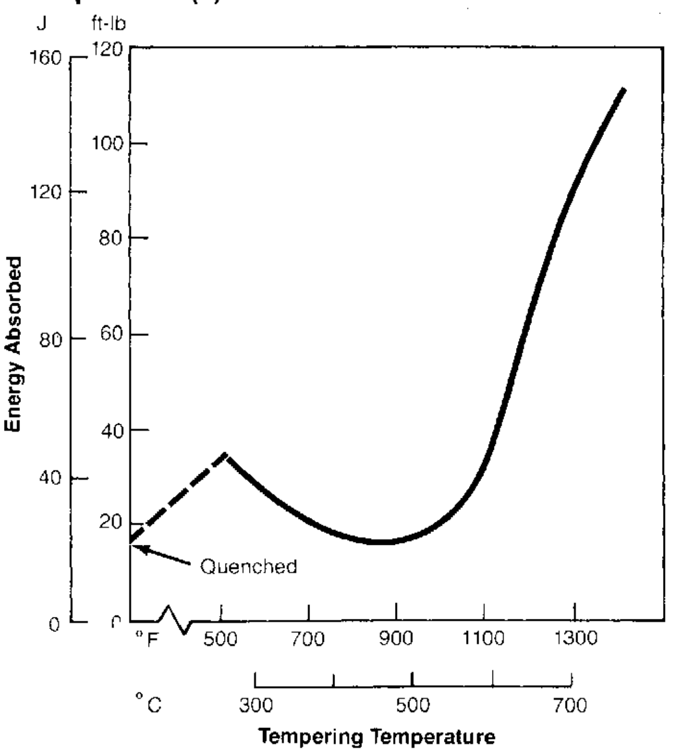

Figure 19

Izod Data for 410 After Quenching from 1800°F (982°C) and Tempering at 1150°F (621°C). (Bhn 228) (2)

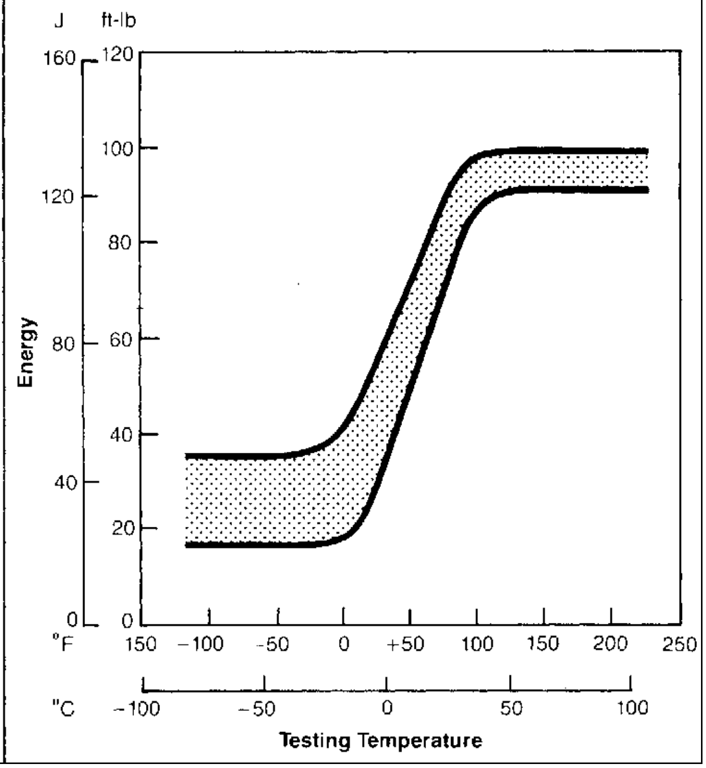

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Figure 20

Fatigue Data for Quenched-and-Tempered Type 403 (Rockwell C 24 to 26) (2)

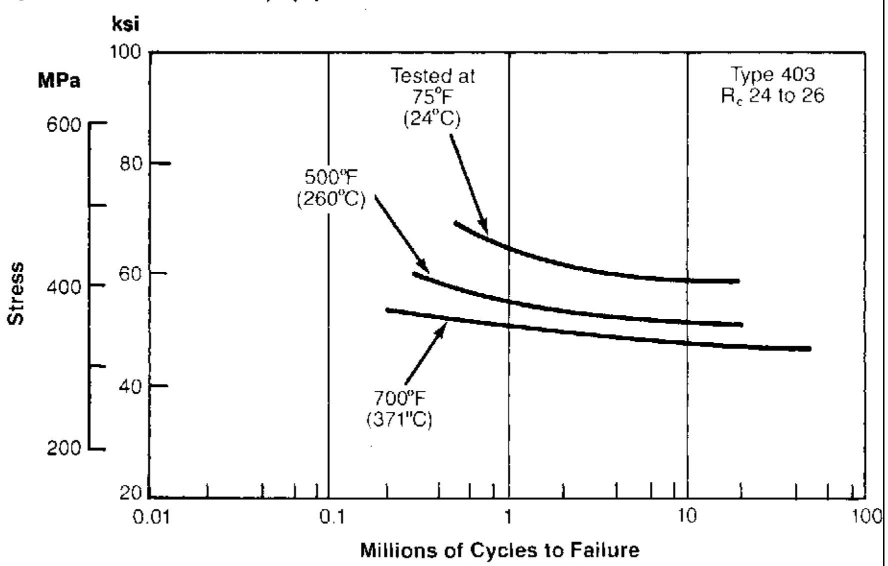
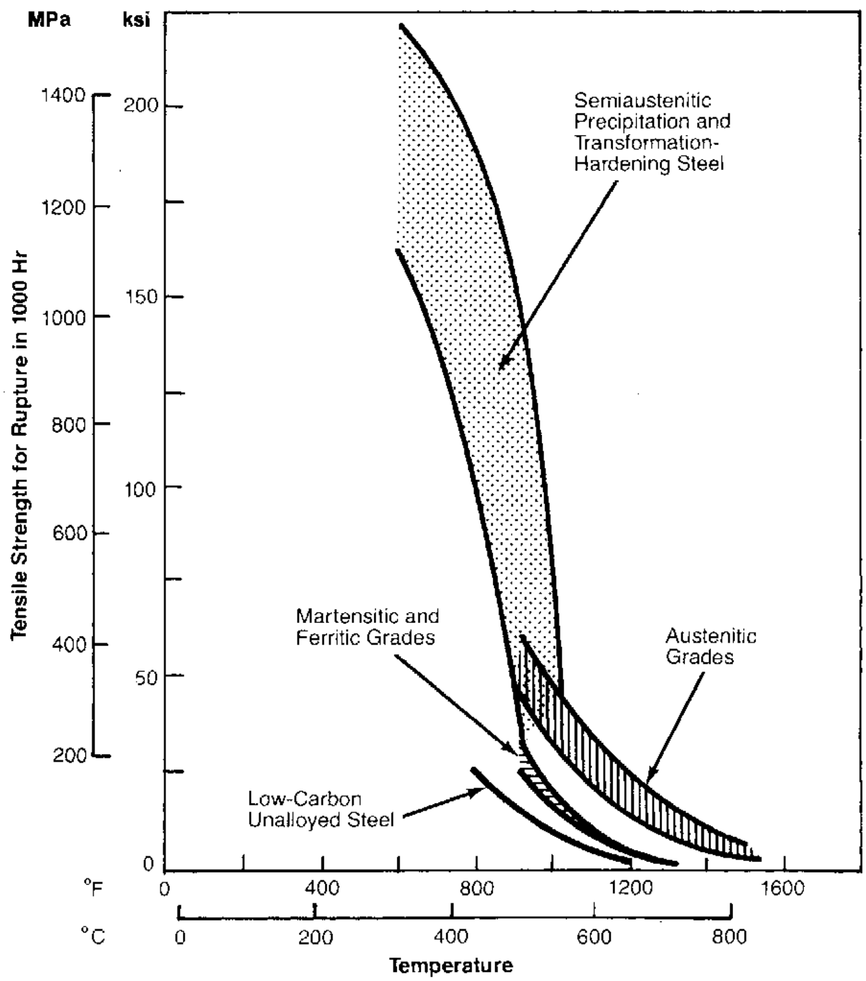


Figure 21

Hot-Strength Characteristics (2)



General comparison of the hot-strength characteristics of austenitic, martensitic and ferritic stainless steels with those of low-carbon unalloyed steel and semi-austenitic precipitation and transformation-hardening steels.

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Figure 22

Effect of Cold Work on the Short-Time Tensile Properties of Type 301 at Elevated Temperature (2)

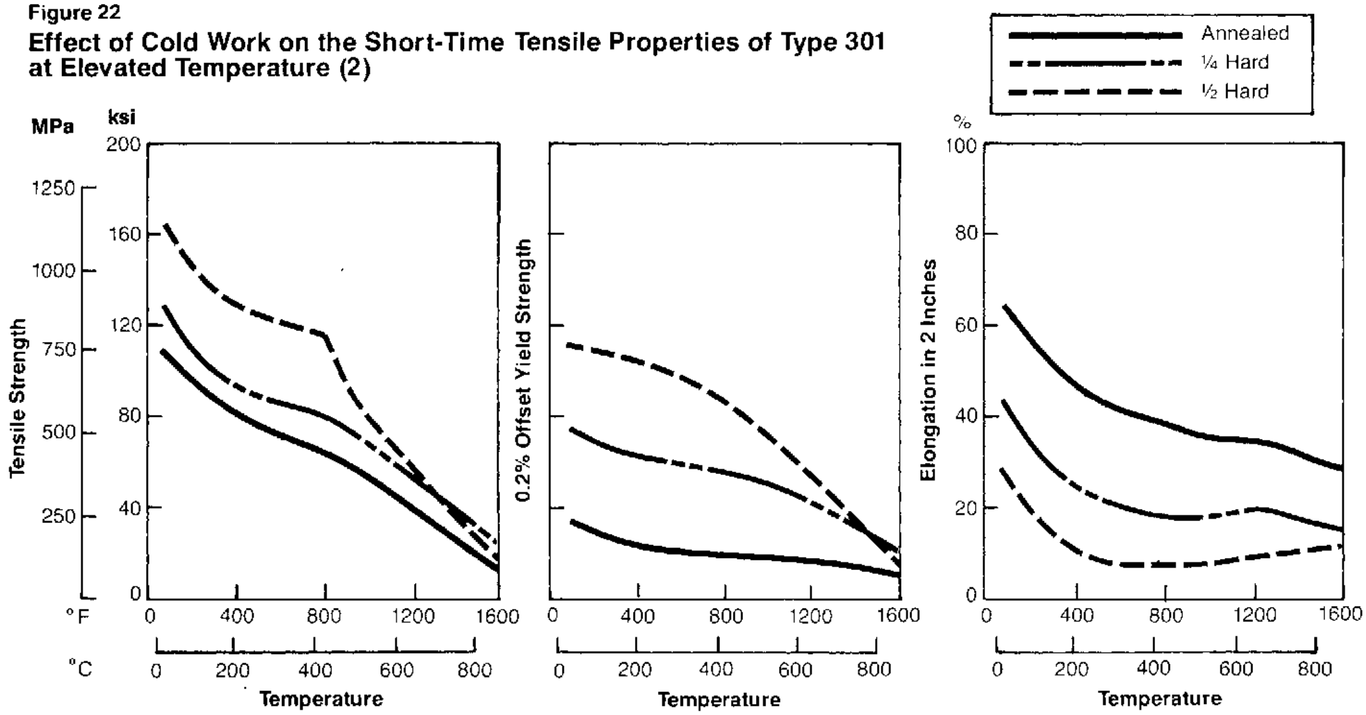

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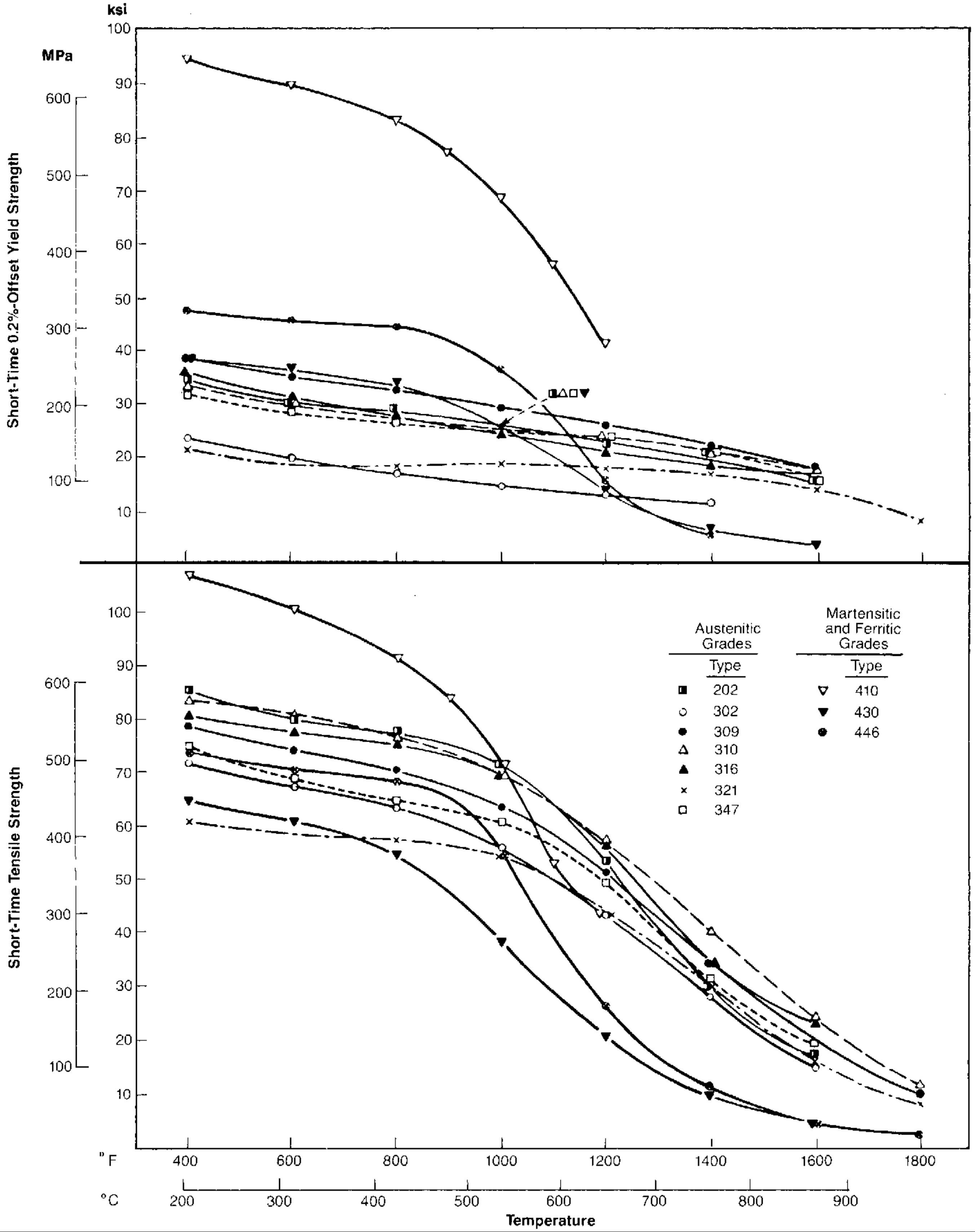
Figure 23 Short-Time Tensile Strengths (2)



Typical short-time tensile strengths of various standard stainless steels at elevated temperature. All steels were tested in the annealed condition except for the martensitic Type 410, which was heat treated by oil quenching from 1800 ºF (982 ºC) and tempering at 1200 ºF (649 ºC).

Figure 24, 25, 26

Comparative 100,000-hr Stress-Rupture Data for Types 316 and 347 Tube and Pipe and on Type 304 Bar. (2)

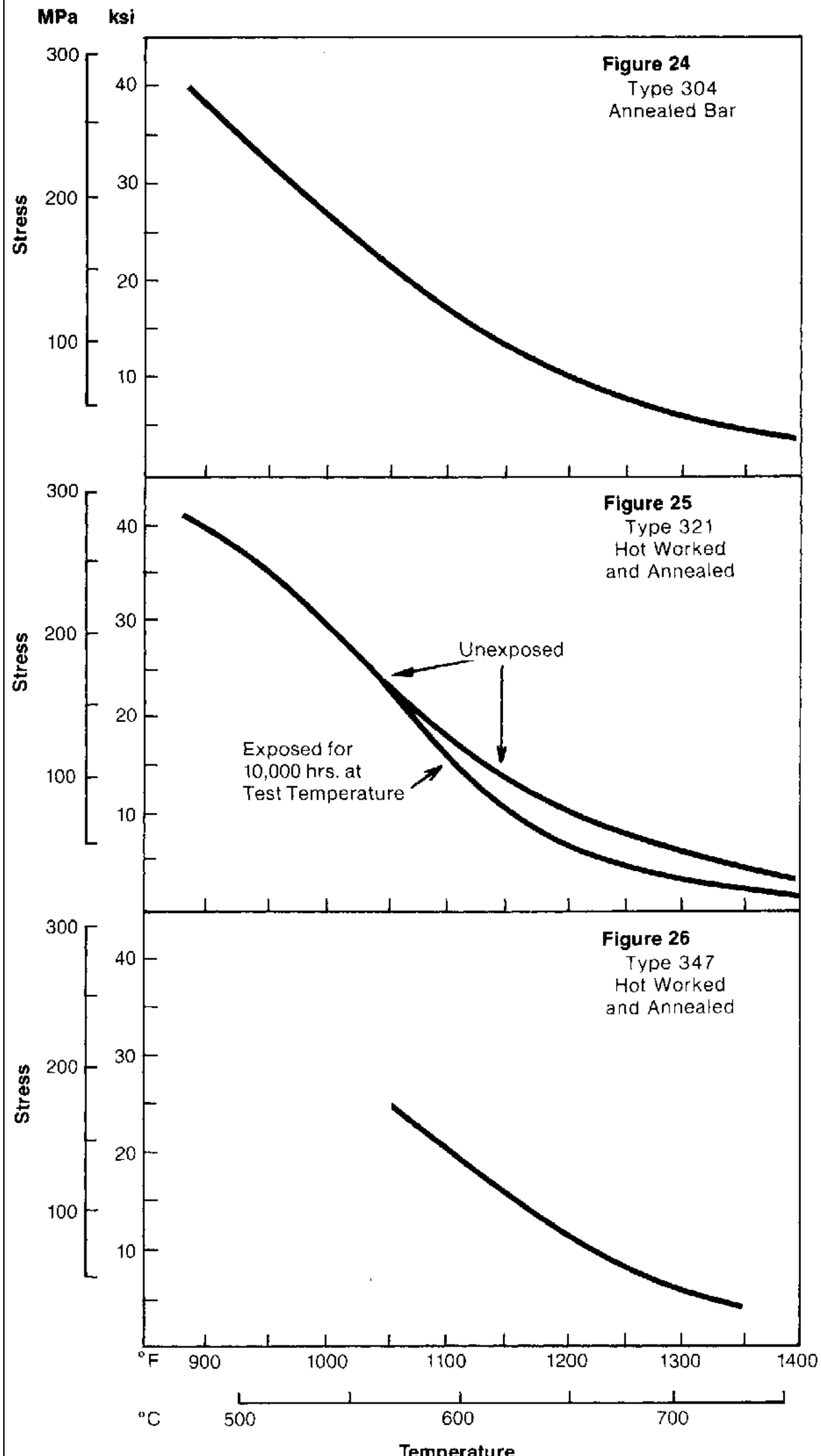


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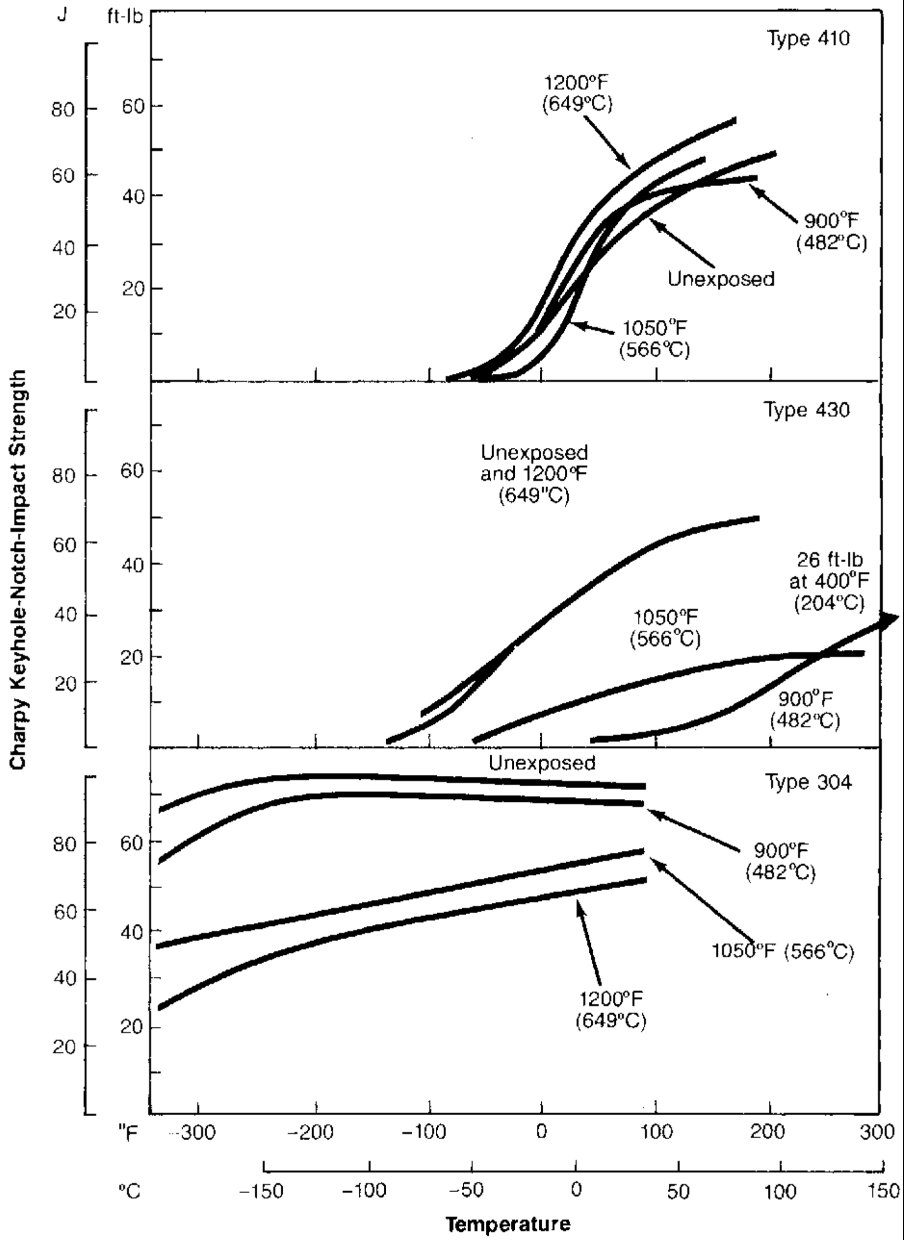
Figure 27

Effect of Holding 10,000 Hr at 900, 1050 and 1200°F (482, 566, and 649°C) on the Impact Characteristics of Type 410, 430 and 304 (11)



Hardness Values Were as Follows

Type	DPN Hardness Unexposed	After Exposure for 10,000 hr at 900°F (482°C)	1050°F (566°C)	1200°F (649°C)
410	125	125	124	123
430	185	274	198	169
304	138	140	147	141

Figure 28

Linear Thermal Expansion of the Three Main Classes of Stainless Steel

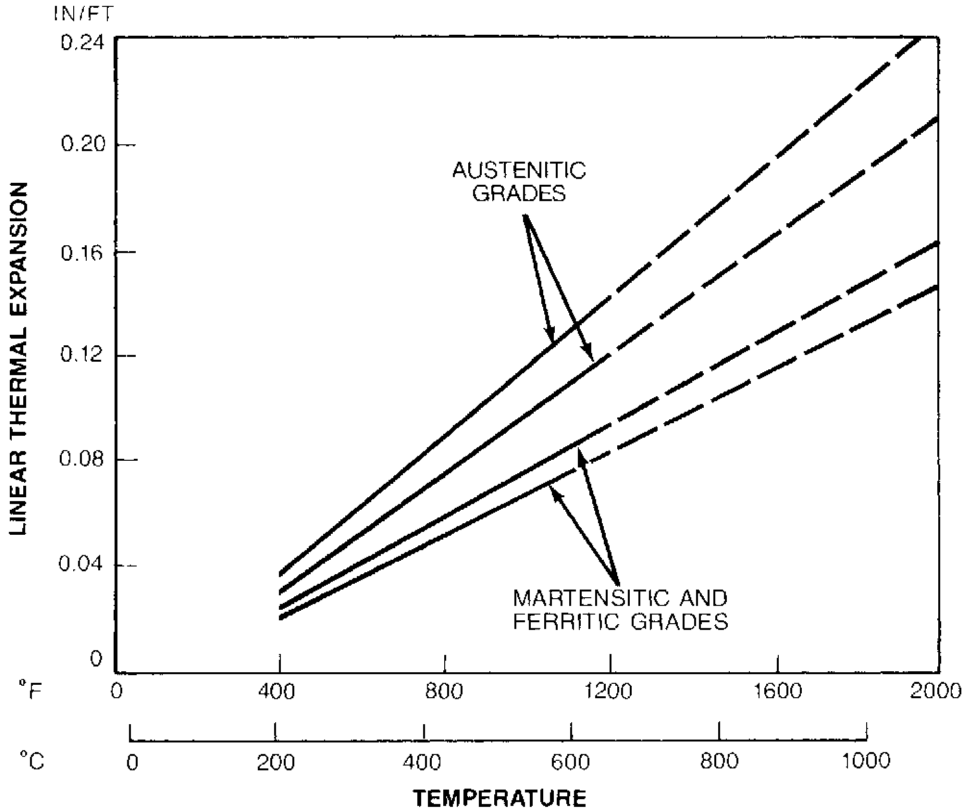


Figure 29

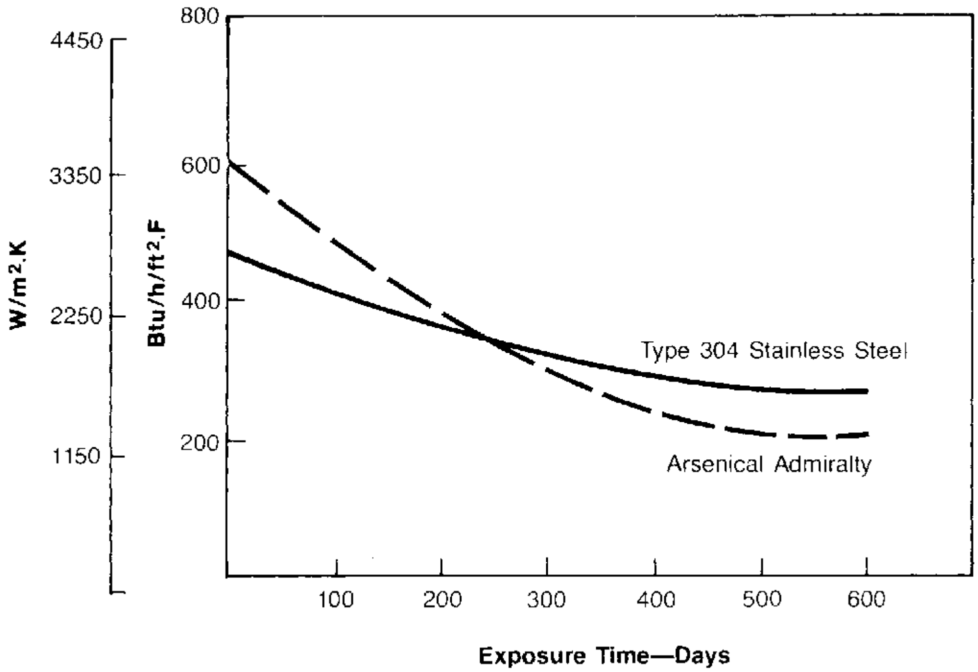
Factors Affecting Heat Transfer (12)

Steam Side Water Film	18%
Steam Side Fouling	8%
Tube Wall	2%
Water Side Fouling	33%
Water Side Film	39%

II

Figure 30

Overall Heat Transfer vs. Exposure Time (13)



II

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18. Tranter Mfg., Inc.

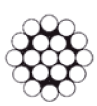
II

II

MIDLAND WIRE CORDAGE
COMPANY LIMITED

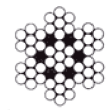
Data kindly supplied by: Midland Wire Cordage Company Ltd., Wire Rope House, Eagle Road, North Moonsmoat, Redditch, Worcester. B98 9HF, UK.
Signifcant elongation can occur before breaking, normal maximum working load 50% of breaking load.

Galvanised Wire Rope



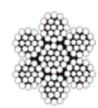
1 X 19 (12/6/1) WIRE STRAND

Construction	Dia mm	Grade	Min. Load Breaking /tonnes	~ weight kgs/100m
1 X 19 (12/6/1) WIRE STRAND	1.0	180	0.09	0.49
	1.14	180	0.12	0.63
	1.2	180	0.13	0.70
	1.25	180	0.14	0.76
	1.3	180	0.15	0.82
	1.5	180	0.20	1.10
	1.6	180	0.23	1.25
	1.8	180	0.30	1.58
	1.9	180	0.33	1.76
	1.93	180	0.34	1.82
	2.0	180	0.37	1.95
	2.1	180	0.40	2.15
	2.2	180	0.44	2.36
	2.3	180	0.49	2.58
	2.5	180	0.57	3.05
	2.66	180	0.65	3.45
	3.0	180	0.83	4.39
	3.5	180	1.13	5.97
	4.0	180	1.47	7.75
	4.75	180	2.08	11.00
	5.0	180	2.31	12.20
	6.0	180	3.32	17.50
	8.0	180	5.92	31.20
	10.0	180	9.27	48.80



7 X 7 (6/1) WIRE STRAND CORE

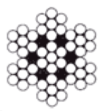
Construction	Dia mm	Grade	Min. Load Breaking /tonnes	~ weight kgs/100m
7 X 7 (6/1) WIRE STRAND CORE	1.2	180	0.14	0.53
	1.5	180	0.15	0.90
	1.8	180	0.21	1.23
	2.0	180	0.26	1.52
	2.5	180	0.40	2.20
	3.0	180	0.58	3.43
	3.5	180	0.84	4.63
	4.0	180	1.04	6.10
	5.0	180	1.62	9.53
	6.0	180	2.33	13.70
	7.0	180	3.17	18.70
	8.0	180	4.15	24.40
	9.0	180	5.25	30.90
	10.0	180	6.47	38.50
	12.0	180	9.33	54.40



7 X 19 (12/6/1) WIRE STRAND CORE

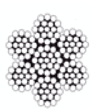
Construction	Dia mm	Grade	Min. Load Breaking /tonnes	~ weight kgs/100m
7 X 19 (12/6/1) WIRE STRAND CORE	2.0	180	0.24	1.45
	2.5	180	0.37	2.40
	3.0	180	0.54	3.43
	3.2	180	0.82	3.80
	3.8	180	0.86	5.36
	4.0	180	0.96	6.10
	4.57	180	1.27	7.76
	5.0	180	1.50	9.53
	5.3	180	1.77	10.40
	5.5	180	1.81	11.20
	6.0	180	2.16	13.70
	6.3	180	2.37	14.70
	7.0	180	3.15	19.50
	9.0	180	5.20	32.20
	10.0	180	6.42	39.80
	11.0	180	7.77	48.20
	12.0	180	9.25	57.30

Stainless Steel Wire Rope



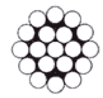
7 X 7 (6/1) WIRE STRAND CORE

Construction	Dia mm	Grade	Min. Load Breaking /tonnes	~ weight kgs/100m
7 X 7 (6/1) WIRE STRAND CORE	.63	304	.033	0.16
	.81	304	.054	0.26
	.90	304	.066	0.32
	1.0	304	.080	0.39
	1.2	304	.115	0.50
	1.5	316	.136	0.90
	2.0	316	.242	1.51
	2.5	316	.364	2.36
	3.0	316	.545	3.40
	4.0	316	.968	6.05
	5.0	316	1.51	9.46
	6.0	316	2.18	13.60
	7.0	316	2.97	18.50
	8.0	316	3.87	24.20
	9.0	316	5.23	31.20
	10.0	316	6.05	37.80



7 X 19 (12/6/1) WIRE STRAND CORE

Construction	Dia mm	Grade	Min. Load Breaking /tonnes	~ weight kgs/100m
7 X 19 (12/6/1) WIRE STRAND CORE	1.5	304	1.70	0.90
	1.7	304	2.15	1.07
	2.0	316	2.30	1.49
	2.5	316	3.50	2.40
	3.0	316	5.10	3.34
	4.0	316	9.06	5.94
	5.0	316	14.2	9.29
	6.0	316	20.4	13.40
	7.0	316	27.8	18.20
	8.0	316	36.3	23.80
	9.0	316	48.9	32.20
	10.0	316	56.7	37.20
	11.0	316	72.3	48.20
	12.0	316	81.6	53.50
	13.0	316	89.65	67.30
	14.0	316	11.10	72.80
	16.0	316	13.60	95.10



1 X 19 (12/6/1) WIRE STRAND

Construction	Dia mm	Grade	Min. Load Breaking /tonnes	~ weight kgs/100m
1 X 19 (12/6/1) WIRE STRAND	0.55	304	.033	0.14
	1.0	304	1.10	0.49
	1.2	304	1.50	0.70
	1.5	316	1.89	1.10
	1.6	316	2.77	1.25
	2.0	316	3.20	1.90
	2.3	316	4.90	2.58
	2.5	316	5.00	3.05
	3.0	316	7.20	4.39
	4.0	316	1.28	7.81
	5.0	316	2.00	12.20
	6.0	316	2.88	17.60
	7.0	316	3.55	23.90
	8.0	316	4.64	31.20
	9.0	316	5.87	39.50
	10.0	316	7.25	48.80

II

Part III

Electrical Information

RESISTOR & CAPACITOR DATA

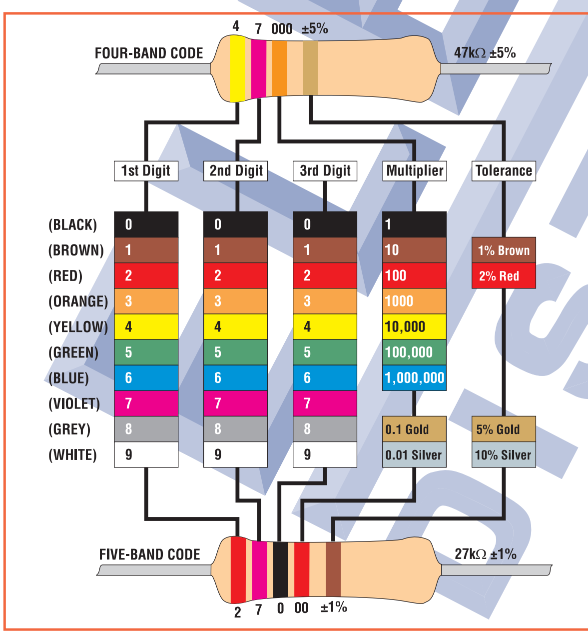
Resistors

Many resistors are so small that it would be difficult to print their value and % tolerance on their body in digits. To overcome this, a coding system based on bands of distinctive colours was developed to assist in identification. Learning this 'colour code' is not as necessary as it used to be (thanks to accurate, low cost digital multimeters!), but it's not hard to learn and it's quite useful knowledge anyway.

The first thing to know is that in each decade of resistance — i.e., from 10 - 100Ω, 100 - 1kΩ, 1k - 10kΩ, etc — there are only a finite number of different nominal values allowed. Most common resistors have values in the 'E12' series, which only has 12 allowed values per decade. Normalised these are 1.0, 1.2, 1.5, 1.8, 2.2, 2.7, 3.3, 3.9, 4.7, 5.6, 6.8 and 8.2. Multiples of these values are simply repeated in each decade — e.g., 10, 12, 15, 18 and so on. Note that the 'steps' between these values are always very close to 20%, because the E12 series dates from the days of resistors with ±10% tolerance.

To allow greater accuracy in circuit design, modern 1% tolerance resistors are made in a larger range of values: the 'E24' series, which has 12 *additional* allowed values per decade as shown in the table. As before, these nominal values are simply repeated in each decade. The table at right shows both the E12 and E24 allowed values for comparison.

The next thing to know is that there are **two** different resistor colour coding systems in use: one using a total of 4 colour bands, and the other 5. The 5-band system is generally used for 2% and closer tolerance resistors, even though the 4-band system is quite capable of handling any resistors with E12 or E24 values. Both systems use the same band colours to represent the various digits; the main difference is that 5-band resistors have an additional 'third band', which is almost always BLACK to represent a third digit of '0'. Here's how both systems work in practice:



4-band resistors will almost always have values in the E12 series, while 5-band resistors can have any value in the E24 series. This is worth remembering, because depending on the resistor's body colour, some of the band colours may not be easy to distinguish. Blue (6) and grey (8) sometimes look very similar, as do red (2), brown (1) and orange (3). So if you're in doubt, check the apparent coded value against the allowed E12 or E24 values to see if it's 'legal' — or check with a digital multimeter, just to make sure.

Preferred Resistor Values (within each decade)

E12 Series	E24 Series
10	10
	11
12	12
	13
15	15
	16
18	18
	20
22	22
	24
27	27
	30
33	33
	36
39	39
	43
47	47
	51
56	56
	62
68	68
	75
82	82
	91

Capacitors

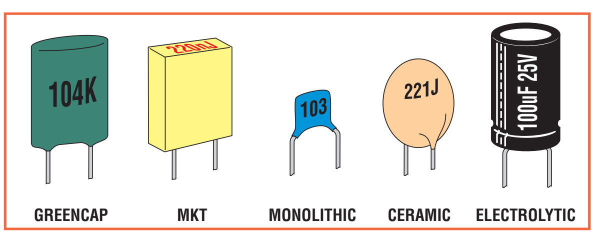
Virtually all of the capacitors stocked by Jaycar have their electrical values printed directly on their body, in digits and letters. However there's often still a coding system, which can make it a bit tricky to work out the capacitance, voltage rating, tolerance and so on until you know how it works. This is explained below.

Incidentally, so-called 'greencaps' (which can actually be brown, dark red or even blue!) are one type of metallised polyester film capacitor, like the 'MKT' type — which tends to be smaller, and in a more tightly controlled rectangular package. Similarly the 'monolithic' type is a type of multilayer ceramic capacitor, designed to combine high capacitance with very low self-inductance.

Plastic film, Ceramic & Monolithic Capacitors

Most of these types have their nominal value either printed directly on them or use the 'EIA' coding system, which is a bit like resistor colour coding, but in digits: the first two digits followed by a 'multiplier' showing the number of zeroes. With this code the value is generally given in picofarads (pF), which you'll need to divide by either one million or one thousand (respectively) if you want the value in microfarads (μF) or nanofarads (nF).

Hence a capacitor marked '104' has a value of 10 with 4 zeroes after it, or 100,000pF (which is the same as 100nF, or 0.1μF). Similarly '681' means 68 with a single zero, or 680pF, while '472' means 47 with two zeroes, or 4700pF (which is the same as 4.7nF).



Alternatively the value may be given directly in nanofarads, with three significant digits but the third generally '0'. In this case there's generally also a small 'n', which can be used in place of a decimal point. So '220n' means a 220nF capacitor, which is the same as 0.22µF, while '3n3' means 3.3nF (= 3300pF).

Many of these capacitors also have a capital letter to indicate their tolerance rating, according to the following coding system:

Capacitor Tolerance Marking Codes

F	G	J	K	M	Z
±1%	±2%	±5%	±10%	±20%	-20%, +80%
Examples: 104K = 0.1µF ± 10%; 4n7J = 4.7nF ±5%					

Material Codes for Plastic Film Capacitors

Capacitors which use a plastic film dielectric are identified using the following codes:

MKT	Metallised Polyester (PETP)
KS	Polystyrene film/foil
MKC	Metallised Polycarbonate
KP	Polypropylene film/foil
KT	Polyester film/foil
MKP	Metallised polypropylene

Ceramic Capacitor Colour coding for Temperature Coefficient

Capacitors which use a plastic dielectric have a very low *temperature coefficient* (tempco) — i.e., their capacitance scarcely varies with temperature, and can generally be regarded as 'stable'. However this isn't true with many ceramic-dielectric types. Many of the ceramic materials produce a negative tempco, where capacitance *decreases* with temperature, while a few give a positive tempco where capacitance increases with temperature.

By careful mixing of materials, manufacturers can produce a ceramic which gives a tempco very close to zero, but the resulting dielectric constant is also quite low. That is why such 'NP0' capacitors are normally only available in relatively low values — less than about 200pF, typically.

The following colour bands are used on ceramic capacitors to indicate their tempco. Note that 'P' indicates a positive tempco and 'N' a negative one, with the number indicating parts per million per degree C.

P100	Red/Violet	**NP0**	Black
N033	Brown	**N075**	Red
N150	Orange	**N220**	Yellow
N330	Green	**N470**	Blue
N750	Violet	**N1500**	Orange/Orange

Electrolytic Capacitors

Electrolytic capacitors take advantage of the ability of some metal oxides to act as an excellent insulator (at low voltages) and also form a dielectric material with a very high dielectric constant 'K'. Most common electrolytic capacitors use aluminium oxide as the dielectric, but special-purpose and low leakage types generally use tantalum oxide.

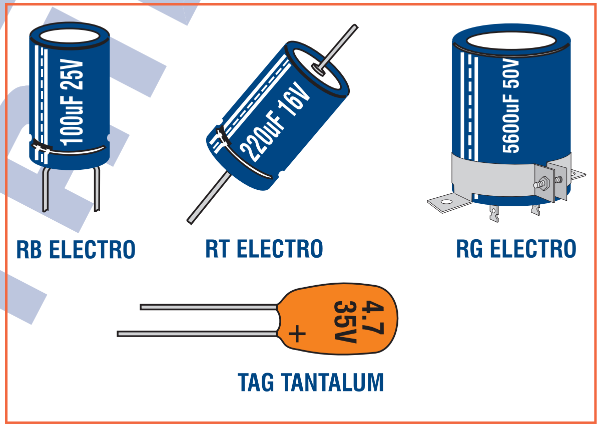
The main shortcoming of electrolytic capacitors is that the insulating and dielectric properties of the metallic oxides are polarity sensitive — so most electrolytic capacitors must be connected into circuit so that voltage is always applied to them with the correct polarity (which is marked on their body). The only exception is 'non polarised' or *bipolar* (BP) electrolytics, which are effectively two electrolytics in series back-to-back.

Because the oxide dielectric layer in electrolytic capacitors is extremely thin, these capacitors are more prone to breakdown at higher voltages. So all electrolytics are clearly marked in terms of their safe maximum operating voltage.

In most cases electrolytics also have their capacitance value shown directly on the case as well.

The three most common types of aluminium electrolytic in current use are the axial-lead or RT type, the radial-lead or RB type (for vertical mounting on PC boards) and the chassis-mounting or RG type. There's also a variation on the RB type called the RP, with a third lead for orientation and added support.

The most common type of tantalum electrolytic in current use is the solid or TAG tantalum type, where the tantalum oxide dielectric is formed on the surface of a solid block of sintered tantalum granules. These capacitors provide low leakage and very high capacitance in a very small volume, but are limited to quite low voltages — typically less than 33V.



III

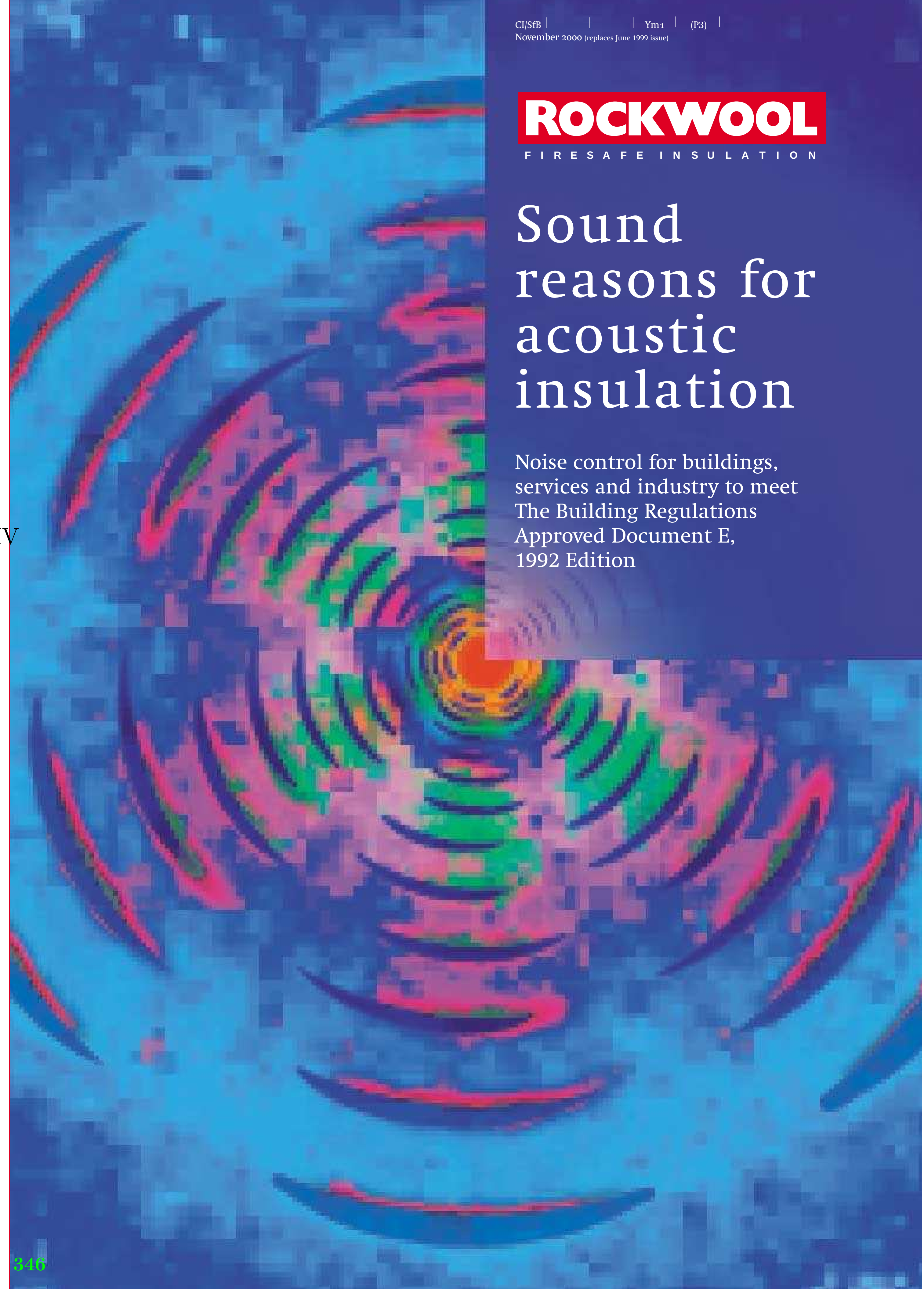
Part IV
Structural Information

CI/SfB | | | Ym1 | (P3) |
November 2000 (replaces June 1999 issue)

ROCKWOOL
FIRESAFE INSULATION

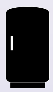
Sound reasons for acoustic insulation

Noise control for buildings, services and industry to meet The Building Regulations Approved Document E, 1992 Edition



IV

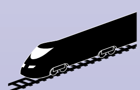
0 dB
Threshold of audibility



30 dB
The hum of a fridge

50 dB
Background noise in an office

60 dB
The sound of a vacuum cleaner



80 dB
An Intercity train from a station platform

90 dB
Pneumatic drill at 3 metres

100 dB
Jet plane taking off at 50 metres

140 dB
Threshold of pain

How is ‘noise’ defined and how is it measured?

Noise can be generally described as unwanted sound. Subjectively, sound is a vibration of the air that is perceived by the human ear. Objectively, sound is a pressure fluctuation which has an intensity (volume) and a wavelength (pitch). The intensity of sound depends on the pressure level, which is measured in decibels (dB) and can be accurately quantified by a meter. Because the perception of the human ear to sound intensity depends also on the pitch, the decibel scale is adjusted to take account of this fact. The adjusted scale is measured in dB(A) units and sound meters have their scales adjusted accordingly. Pitch, or frequency, is expressed in cycles per second. The unit of frequency is the hertz (Hz).

The benefits of using Rockwool

Rockwool products are manufactured to a higher density than other mineral wool slabs and provide improved acoustic control across a wide range of frequencies.

Effective sound insulation is an essential requirement for modern life styles. Excessive noise can increase stress, hinder speech and can cause its own form of environmental pollution.

Rockwool has been proven over many years to be the ideal insulation material for all applications where noise attenuation or noise absorption is needed - in domestic, commercial, manufacturing, industrial and environmental situations. In addition to its acoustic properties, its well known thermal insulation and fire protection performance are inherent benefits.

This sixth edition of the Rockwool Acoustic Manual gives an explanation of the principles of acoustics pertaining to buildings, services and industry. It also contains specific examples of the application of Rockwool products to constructions that will meet the requirements of the Regulations. Where there is noise pollution, there is a Rockwool solution.

Contents

IV

Sound reasons for acoustic insulation

Why do we need to insulate against noise?

Noise can be harmful to health

Exposure to high levels of noise can be dangerous in situations requiring awareness of potential dangers. For example when operating machinery, audible warnings may not be heard.

Prolonged exposure to high noise levels can induce hearing loss. It is for this reason that the Health and Safety at Work Act requires noise levels in a work area to be limited to less than 90 dB(A) during an 8 hour working day. At a noise level of 105 dB(A), exposure time is limited by the Act to 15 minutes. For this reason, ear protection is a statutory requirement in many industrial situations. Whilst the wearing of ear defenders may be a permitted solution, in practice they are often removed because of discomfort and isolation from general workplace communication.

Noise can cause inefficiency at work

Even at permitted sound levels, communication is difficult and productivity can be adversely affected. In offices, intermittent extraneous noise can be disturbing to jobs requiring concentration. Prolonged high pitched noise (such as that produced by air-conditioning ducts or extract fans) can be physically tiring and can affect personal performance.

Low frequency noise is equally disturbing where work requires concentration. Examples are office equipment, generators, traffic noise and general maintenance work in or near the workplace.

Noise can be a nuisance in the environment

Heavy traffic, low aircraft close to airports, road drills, noisy neighbours, are types of noise frequently experienced. Causing noise nuisance can be a legal offence. There are statutory requirements imposing noise limits in some cases, or standards for noise insulation to residential buildings close to motorways and airports.

In the home, noise between rooms can be a nuisance. Although not a statutory requirement, it is recommended for example that internal walls between bedrooms and toilets or bathrooms should have a minimum noise reduction value of 38 dB.

Statutory requirements

The Building Regulations 1995, Approved Document E (1992 edition) sets standards for the noise insulation between dwellings, which are detailed on pages 10 to 15.

Acceptable noise levels in the workplace are determined by the Health and Safety at Work Act and the Noise at Work Regulations 1989.

Noisy plant or machinery used in public places or close to occupied buildings has to be insulated against noise in accordance with current Government environmental policy directives.

External noise

The Department of the Environment, Transport and the Regions (DETR) requires residential buildings built in the vicinity of an existing motorway or other major traffic route to be shielded from noise nuisance. The current level is set at 70 dB(A) measured at the building facade.

In a brick built elevation, this reduction is usually achieved by the installation of double glazing. Consideration should also be given to insulation of roof spaces with Rockwool Rollbatt. This can be used in applications where one layer is placed between the joists and a second layer at right angles across the joists. This method has proved in practice to offer an effective reduction in external noise nuisance.

New roads or railways planned close to existing residential property are required to be provided with acoustic shielding.

Glazing

Double or triple glazing can greatly improve sound insulation where external noise or noise between internal rooms is a problem. It is usual to install different thicknesses of glass for each leaf, or to place them at a different angle, or both, to prevent the sympathetic vibration which occurs between two similar membranes. The sound reduction is further improved by placing a sound-absorbent lining in the reveals between the glazing.

What measures can be taken against noise?

There are two distinct ways in which acoustic insulation can be used to control noise; by transmission loss or by sound absorption.

Transmission loss

Transmission loss is the reduction in the amount of sound energy passing through a wall, floor, roof etc and is a property of the element as a whole. It is expressed in decibels (dB).

Noise may be due to airborne or impact sound and both must be taken into account where appropriate.

The traditional solution for reducing sound transmission is to use constructions with materials of high mass (eg concrete or brickwork). However, the use of heavyweight elements may not always be practicable. Lightweight alternative constructions using plasterboard on studding with a Rockwool core achieve a high level of attenuation.

Flanking transmission

In designing for acceptable levels of sound transmission, particular attention should be paid to ways in which sound may by-pass the element by penetrating other construction at the periphery, or via unsealed doors or windows. Known as 'flanking sound transmission', the subject is extensively covered in Approved Document E.

Neighbourhood noise

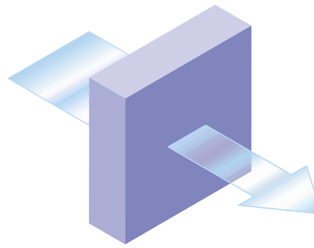
Where manufacturing industries or certain types of leisure and entertainments activities lie close to residential districts, acoustic control is often desirable to help reduce local noise nuisance. An acoustic envelope similar to that needed for overhead or external noise may be one solution, while some sources may need special acoustic enclosures.

In cases of neighbour noise between adjacent dwellings causing interference, nuisance or distress, the problem can be reduced by fitting acoustic insulation to party walls and floor structures.

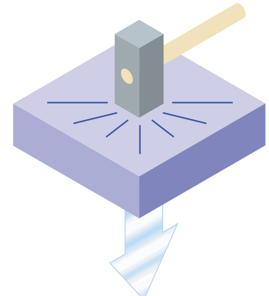
The exclusion of outside and overhead noise can be achieved by insulating the building structure at the time of construction, or in the case of existing buildings, as part of a refurbishment scheme.

Incidental improvement

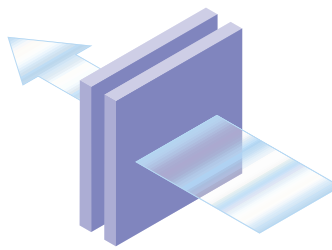
Where improvement schemes specify thermal upgrading using Rockwool insulation, such installations will often provide sufficient incidental acoustic insulation to reduce neighbourhood noise problems.



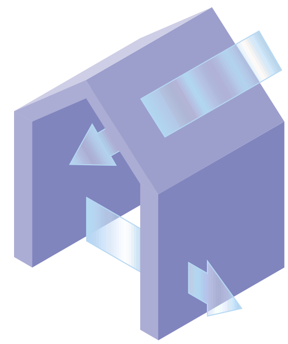
Transmission loss through wall element (airborne sound)



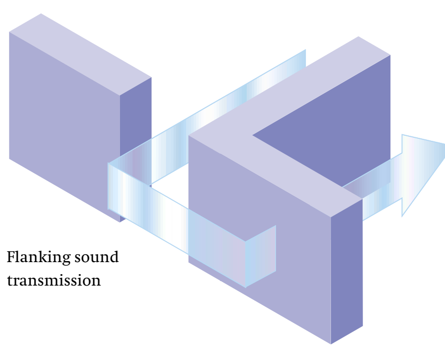
Structure borne sound through floors (impact sound)



Transmission loss through partitions (airborne sound)



Overhead noise from outside the building and preventing neighbourhood noise nuisance



Flanking sound transmission

IV

Sound absorption

Hard, smooth surfaces have the characteristic of reflecting sound, and a noise source can become amplified in a room lined with such materials. This process is called reverberation, and a 'reverberation room' is used to test the sound absorbency of acoustic materials.

The ability of certain materials to absorb sound, notably Rockwool insulation, can be utilised in reducing reverberation and reflected noise within rooms.

Sound absorption coefficient

This is the term used to describe how well (or badly) a particular material absorbs sound energy.

It is denoted by a and is defined as:

$$a = \frac{\text{Sound energy not reflected from material}}{\text{Sound energy incident upon material}}$$

A perfect sound absorber would have a coefficient of a = 1, and perfectly reflective material a coefficient of a = 0.

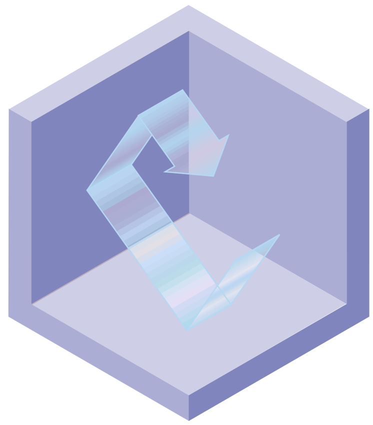
The absorption coefficient of materials varies with the sound frequency (Hertz) and also with the angle at which the sound strikes the material.

Some examples of sound absorption coefficients for Rockwool materials are shown below.

Reducing noise levels

Where there is a need to reduce ambient noise levels within a room, or to modify sound within a space, the answer is to install noise absorbing linings to interior surfaces. Specialist products, such as Rockwool Acoustic Elements and Rockwool Sound Absorption Board are most effective for this purpose. There is also a wide choice of ceiling linings, for example sound absorbent panels fixed to the soffit or suspended ceilings with lay-in boards.

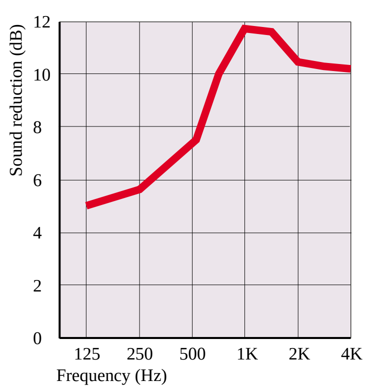
In addition, bespoke systems can be developed for individual situations with special acoustic requirements.



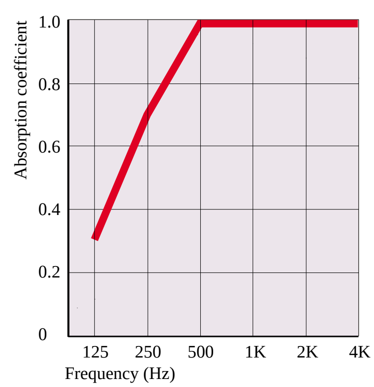
Absorption coefficients for Rockwool Acoustic products

Material	**Thickness (mm)**	**Mounting**	**Frequency (Hz)**					
			125	**250**	**500**	**1K**	**2K**	**4K**
Acoustic Slab	47	Direct	0.20	0.50	0.85	1.00	1.00	1.00
Acoustic Slab	67	Direct	0.30	0.70	1.00	1.00	1.00	1.00
Slab RW3	50	Direct	0.11	0.60	0.96	0.94	0.92	0.82
Slab RW3	75	Direct	0.34	0.95	1.00	0.82	0.87	0.86
Slab RW5	30	Direct	0.10	0.40	0.80	0.90	0.90	0.90
Slab RW5	30	300 mm gap	0.40	0.75	0.90	0.80	0.90	0.85
Slab RW5	75	Direct	0.40	0.75	0.90	0.80	0.90	0.85
Slab RW6	50	Direct	0.20	0.75	0.90	0.85	0.90	0.85
Slab RW6	50	300 mm gap	0.65	0.55	0.75	0.85	0.75	0.85

The absorption coefficients shown above are typical figures that can be achieved with Rockwool products. They have been obtained from a comprehensive range of measurements made over a number of years.



Typical noise reduction achieved through Rockwool absorbent lining to walls and ceilings enclosing noisy machinery



Typical absorption coefficients - Rockwool 67 mm Acoustic Slab

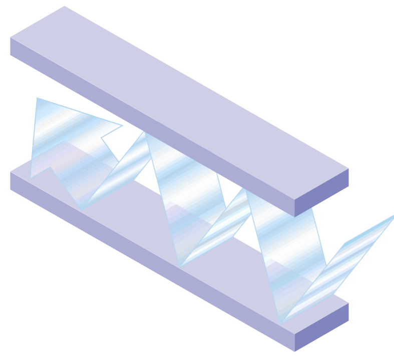
IV

Noise transmitted along ducts

Reduction of transmitted noise through pipes and air conditioning ductwork is achieved by the use of acoustic linings or claddings. Products such as Rockwool Ductslab, Ductwrap and Techwrap provide both acoustic and thermal protection.

Room to room noise via service penetrations

Airborne and structure borne noise from adjacent rooms via pipes, ducts and services can be reduced by the installation of acoustic sleeves at wall penetration points. Rockwool mineral wool slab, mat or loose fill are ideal for this purpose.



Noise transmitted along an air duct

Noise transmitted via a service penetration

IV

Noise from plant and equipment

A number of practical steps can be taken to reduce noise levels, but it is always advisable to seek expert technical opinion before undertaking any acoustic measures.

- Whenever practicable, choose quieter machines and processes. Ask for an assessment of plant and equipment, bearing in mind the acoustic environment in which they will be operating.
- Surround noisy machines individually, eg heavy motors, high speed fans, with an insulated enclosure, as in figure 1. Such enclosures can typically achieve about 30 dB reduction. Ideally, the design should incorporate an impervious enclosure of high mass and with an absorbent lining to prevent reflected noise build-up within the enclosure. Ventilation should be provided and any openings acoustically lined.
- Machines which produce vibration noise, eg ball mills, diesel engines, centrifuges, should be positioned on resilient isolation mounts, which may include rubber pads and/or Rockwool slabs. The design must take into account the loading of the machine and the vibration frequencies.
- Where there are a number of machines, treat all the interior surfaces of the room or enclosure with noise absorbing panels, as in figure 2. It should be borne in mind, however, that the maximum reduction normally achievable is 10 dB, and in cases where only the ceiling is treated, about 5 dB.
  As an alternative, quiet rooms can be constructed for personnel, using the same techniques.
- When very noisy equipment is mounted directly on an intermediate floor, the floor itself may be subject to very high sound pressure levels, causing noise problems in the room below. Installing a concrete inertia base on resilient mounts can act as an effective barrier.
- Noisy pipes, ducts, and gas or liquid handling plant can be individually sound proofed by enclosing them in dense Rockwool. The bases of plant, such as pumps or fans should be acoustically treated. In special cases, eg to reduce noise levels from turbine systems in power stations, the wrapping is made up of layers of mineral wool and is faced with steel sheet or similar.

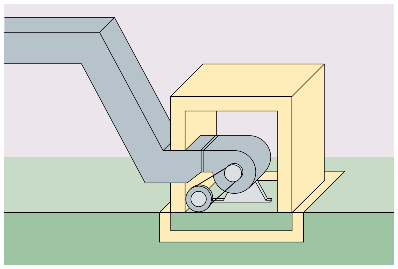


Fig.1 Isolation of plant noise source by individual enclosure and acoustic treatment of base.
Enclosure formed by steel faced partition with perforated inner sheet and incorporating Rockwool Rigid Slab (typically RW3).

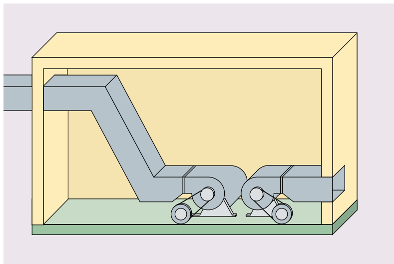


Fig.2 Acoustic lining to plant room where individual enclosure is impracticable.
Lining formed using Rockwool Acoustic Elements (see page 23).

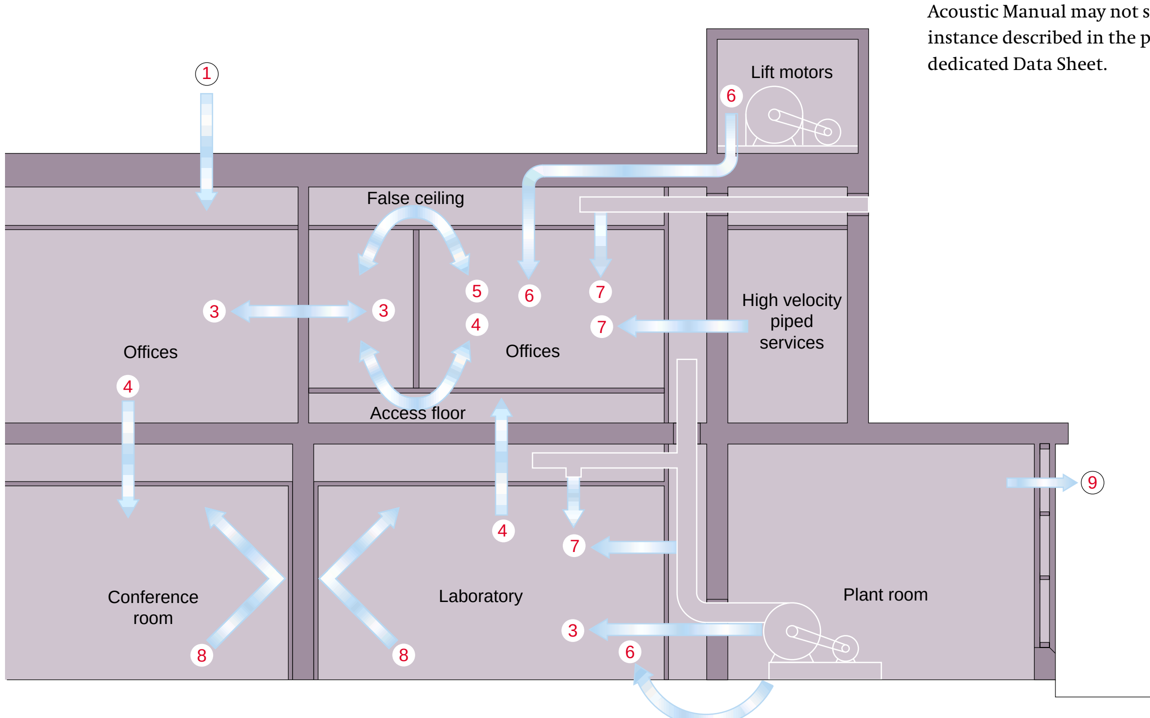
Selecting the right Rockwool product

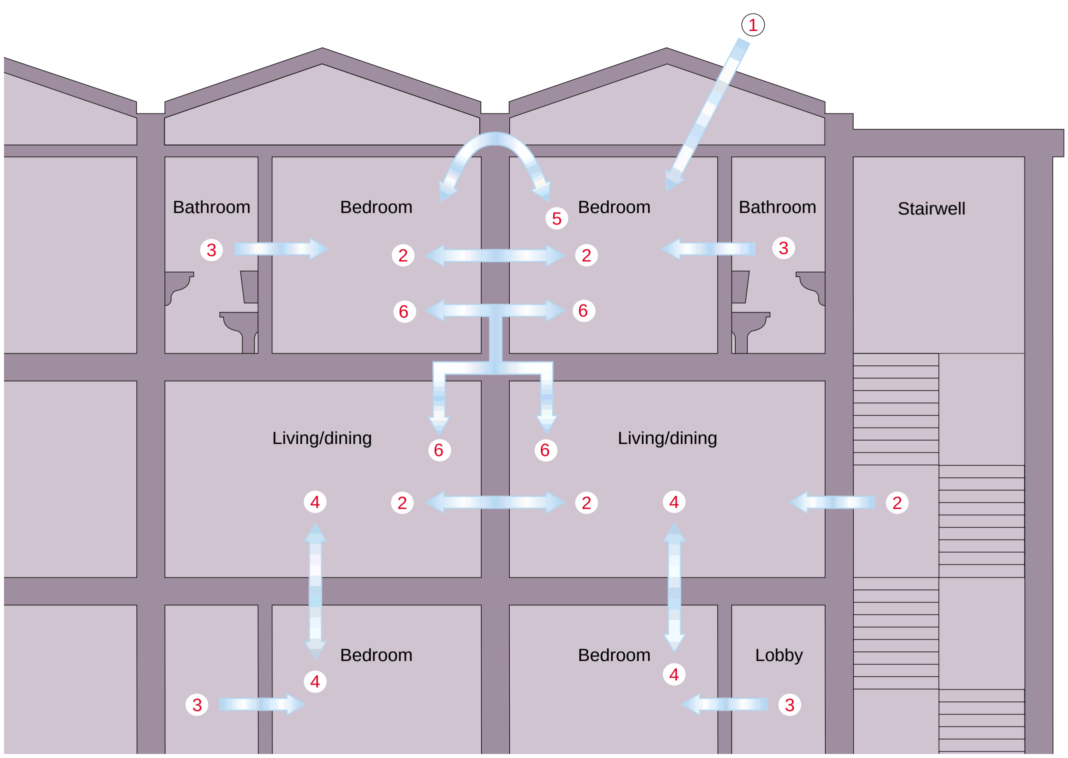
The table on the facing page shows the range of specialised Rockwool acoustic products for use in commercial, residential and other buildings. The table has been split into two parts, the first for commercial and other buildings, the second for dwellings and other residential buildings.

The vertical columns give the location or building element, and the symbol identifies the appropriate product and its acoustic function. Cross reference to the drawings on this page will show specific examples of application.

Reference is made in much of the following text to Rockwool Data Sheets, eg (Data Sheet 081).

These are available from Rockwool's SP&A Department (01656 868420). They describe in full detail the performance and best acoustic application of the product. This Acoustic Manual may not show every instance described in the product's dedicated Data Sheet.





IV

IV

Page No.	Rockwool product	Lightweight partitions	External walls	Internal walls	Intermediate floors	Flat roofs	Ceilings and soffits	Ductwork and pipework
Commercial and other buildings								
10-15	Acoustic Slab	3		2	4			
16	Rockfloor				4			
19	Hardrock					1		
16	Lamella Floor Units				4			
16	Acoustic floor system				4			
19	Cladding Roll		1 9					
19	Composite Panels		1 9					
17	Fire Barrier						5	
18	Soffit Liner						8	
24	Techwrap							7
18	Sound Absorption Board			8			8	
Dwellings and other residential buildings								
11	Timber Batt			2				
10	Liner Board		1	2				
10-15	Acoustic Slab	3		2	4		5	
19	Hardrock					1		
17	Acoustic Party Wall dpc		6					
16	Rockfloor				4			
16	Acoustic floor system				4			
16	Lamella Floor Units				4			

Key to symbols

1. *Noise from outside*
2. *Room-to-room noise (party walls)*
3. *Room-to-room noise (partitions)*
4. *Room-to-room noise (floors)*
5. *Room to room noise (false ceiling)*
6. *Flanking transmission*
7. *Noisy ducts and piped services*
8. *Internal noise levels*
9. *Environmental noise*

Separating walls between dwellings (new build)

The following three construction methods meet Building Regulation requirements if carefully constructed:

Wall Type 1: Solid masonry having a large mass (weight) per unit area.

Wall Type 2: Lightweight masonry of lower mass, but with two leaves separated by an airspace and connected by wall-ties only.

Wall Type 4: Lightweight framed construction (timber or metal stud) of double wall with a wide airspace and the minimum necessary connection between the two leaves; usually with 2 layers of plasterboard on each face, and Rockwool Batts or Acoustic Slab insulation within the cavity (see Figures 5 and 6, page 11).

Upgrading sound insulation of existing masonry party walls

Where a noise nuisance has been identified as being the result of poorly constructed walls between dwellings, and statutory sound transmission values have not been achieved, or are not considered sufficient, Rockwool Liner Board can provide the necessary acoustic improvement. Indicative performances are given in the table below:

Indicative performance - upgraded separating walls

Construction	Uninsulated	Insulated with 59.5 mm Liner Board	
		Single-sided	Both sides
Lightweight block (1) cavity wall	50 dB (2)	55 dB (3)	58 dB (2)
Dense block (4) single leaf	40 dB (5)	50 dB (6)	53 dB (3)

These values are for guidance only. It will be seen that the single leaf block construction lined on one side does not achieve the requirements of the Building Regulations.

(1) Plasmor Stranlite

(2) AIRO Report PT3223

(4) 200 kg/m^2

(5) AIRO Report L/1805/1

IV

Upgrading existing separating walls

Rockwool Acoustic Slab
(Data sheet 081)

Fig. 3 shows a well built solid masonry separating or party wall with plaster on both sides, where improvement of sound attenuation is required, and access is available to only one side of the wall.

The high density of the Rockwool Acoustic Slab and the isolation from the wall itself of the added construction should give a sound reduction in the order of 53 dB.

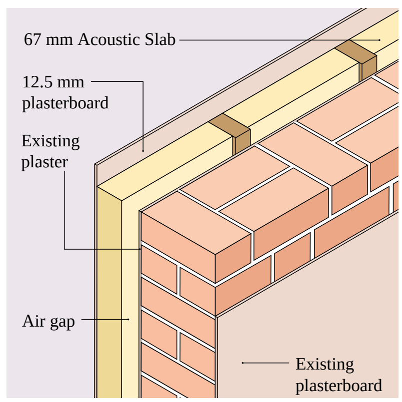


Fig.3 Upgrading an existing separating wall with access limited to one side only

Rockwool Liner Board
(Data sheet 080)

Fig. 4 shows a construction where access is available to both sides of the separating wall. Rockwool Liner Board is adhesive fixed to the blockwork and provides a surface suitable for either skim coat plaster or direct decoration.

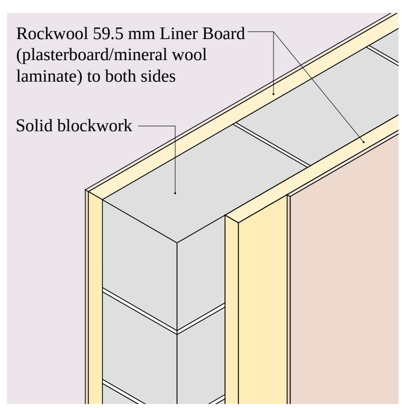


Fig.4 Rockwool Liner Board for upgrading existing cavity block party wall

Timber framed separating walls

Timber Batts (Data Sheet 090, Timber frame products) and Acoustic Slab (Data Sheet 081)
Rockwool Timber Batts or Acoustic Slab, fixed on one side only, in combination with two layers of plasterboard on each side (Fig. 5) provides a sound reduction index which meets the requirements of Approved Document E1/2/3 of the Building Regulations.

Fig. 6 shows a similar construction but with Rockwool Timber Batts placed between the double studs. The constructions shown in both Fig. 5 and Fig. 6 meet the Building Regulations requirement for sound attenuation, as well as affording 1 hour fire resistance.

To achieve these performances, the constructions must be carried up into the roof space.

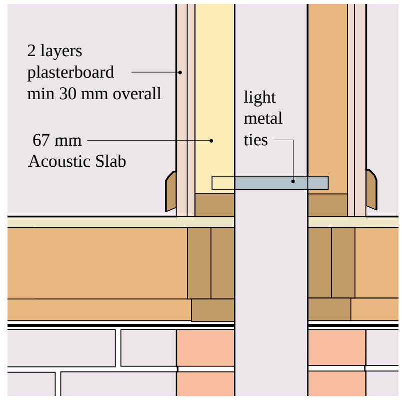


Fig.5 Rockwool Acoustic Slab or Timber Batts as insulation to timber framed separating walls.

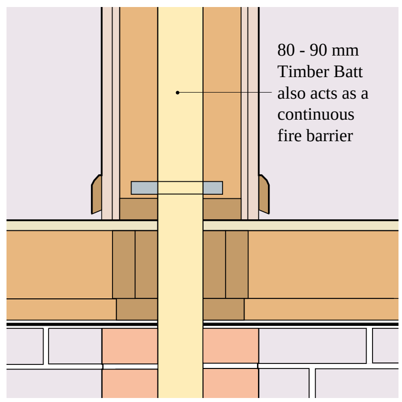


Fig.6 Alternative method of acoustic insulation in timber framed separating wall using Rockwool Timber Batts in the cavity

IV

Acoustic Party Wall dpc
(Data sheet 028)
Acoustic Party Wall dpc provides an acoustic cavity closer where a separating wall joins an external cavity wall. This meets Building Regulations and NHBC requirements.

The Acoustic Party Wall dpc is based on the Rockclose design and is minimum 50 mm† thick, 260 mm wide insulation bonded to 340 mm wide dpc.

†Other thicknesses can be ordered to suit different cavity widths.

Rockwool Acoustic Party Wall dpc has been assessed as giving 1 hour fire resistance (integrity and insulation) by Warrington Fire Research Centre (Ref: C81747).

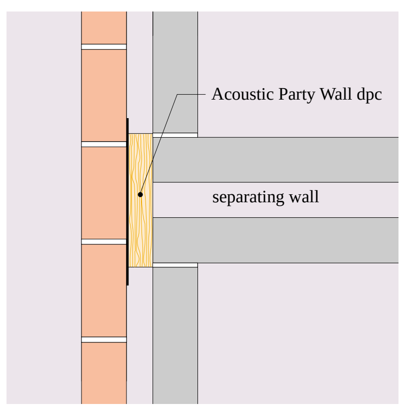


Fig.7 Acoustic Party Wall dpc for sound and fire resistance at separating walls

Partitions with Rockwool infill for improved acoustic performance

Acoustic Slab (data sheet 081)
Rockwool Acoustic Slabs are high quality, resin bonded semi-rigid slabs, designed and manufactured to combine optimum acoustic performance and easy fitting within partition and floor constructions.

Tests have shown that, by fully filling voids in both metal and timber stud partitions, Rockwool Acoustic Slabs provide significantly improved levels of sound reduction compared with glass fibre rolls within the same constructions. Additionally, the product is self-supporting and therefore will not slump or sag inside the partition structure.

Steel stud partitions

These will provide a good level of sound reduction to increase privacy in offices, reasonable sound control in studios, and to isolate noisy machines or processes in industrial applications. Figures 8, 9 and 10 show various applications.

When used to form an enclosure round machinery, the inner steel sheet should be replaced with 30% (min) open area perforated steel, or alternatively Rockwool Acoustic Elements should be used (see page 23).

Fig. 1 Typical lightweight office partition
Weighted sound reduction index: R_W43 dB

Fire resistance: 30 minutes

Studs: 50 mm width @ 600 mm centres

Facings: One layer of 12.5 mm Lafarge Standard Wallboard both sides

Insulation: 47 mm Acoustic Slab

Report No: *BTC 10192A*

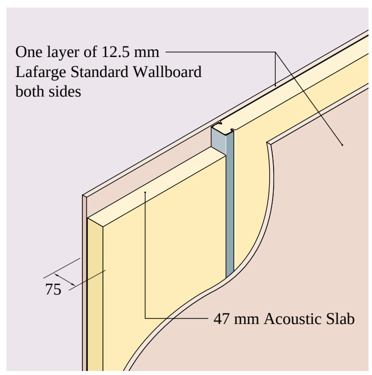


Fig. 8

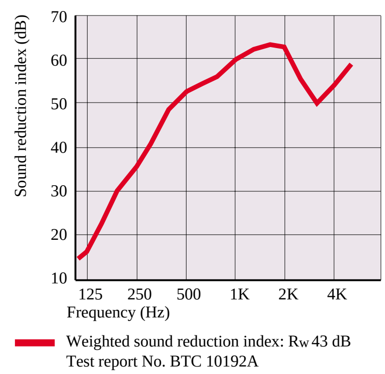


Fig. 2 Typical Hotel bedroom wall partition
Weighted sound reduction index: R_W54 dB

Fire resistance: 30 minutes

Studs: 70 mm width @ 600 mm centres

Facings: One layer of 15 mm Lafarge sound resisting Wallboard one side. Other side one layer of 15 mm sound resisting Wallboard fixed to resilient bar.

Insulation: 67 mm Acoustic Slab

Report No: *BTC 10189A*

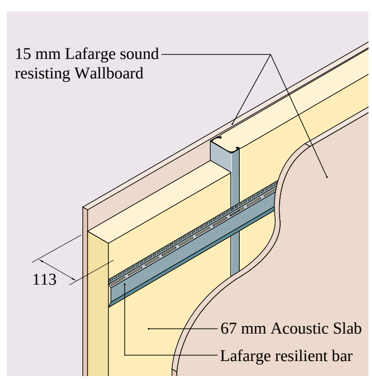


Fig. 9

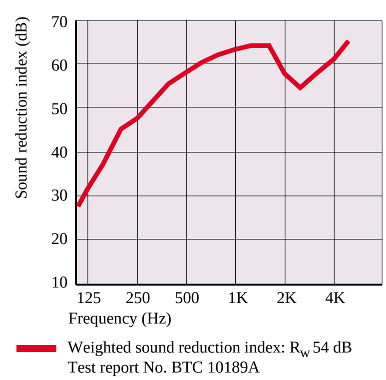


Fig. 3 separating wall/partition
Weighted sound reduction index: R_W55 dB

Fire resistance: 60 minutes

Studs: 70 mm width @ 600 mm centres

Facings: Two layers of 12.5 mm Lafarge Standard Wallboard both sides

Insulation: 67 mm Acoustic Slab

Report No: *BTC 10191A*

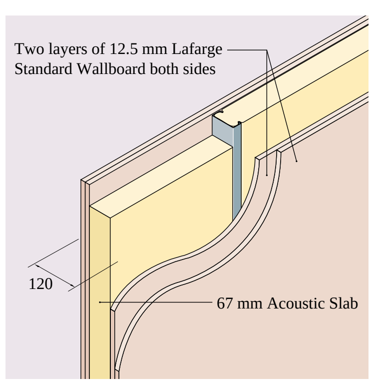
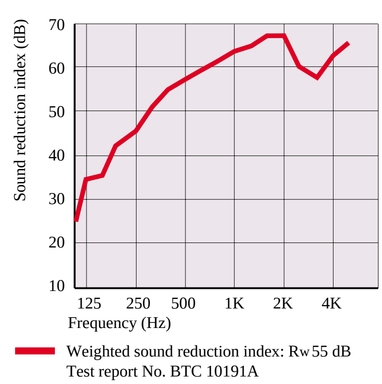


Fig. 10



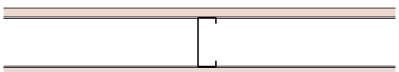
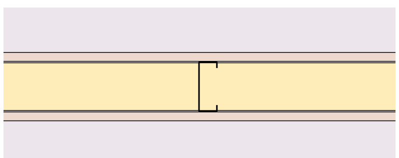
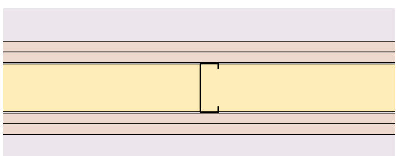
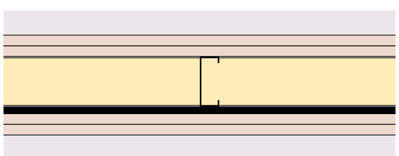
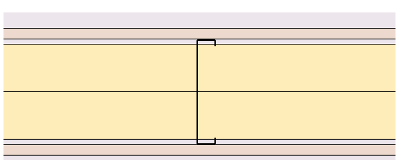
IV

Sound reduction

The composition of Rockwool makes it an ideal product for sound reduction, due to its ability to absorb sound within the cavity of metal and timber framed constructions over a wide frequency range. The table below gives examples of sound reduction figures achievable using Rockwool in a variety of steel frame partitions.

Figures 11 to 14 show four types of metal stud construction using Rockwool Acoustic Slab, with sound reduction figures for each.

The integrity of sound reducing partition constructions can be maintained above suspended ceilings by using Rockwool Fire Barrier. For details see page 17 (Fig. 24).

Partition structure	**Specification**	**Nominal thickness (mm)**	**Weighted sound reduction index (R_WdB)**	**Fire Resistance (minutes)**
	***Studs***: 70 mm width @ 600 mm centres. ***Facings***: One layer 12.5 mm Lafarge Standard Wallboard both sides ***No insulation***	95	36	30
 Fig. 11	As above with 67 mm Acoustic Slab ***Report No***: BTC 10190A	95	44	30
 Fig. 12	***Studs***: 70 mm width @ 600 mm centres. ***Facings***: Two layers of 15 mm Lafarge sound resisting Wallboard both sides ***Insulation***: 67 mm Acoustic Slab ***Report No***: BTC 10183A	132	57	90
 Fig. 13	***Studs***: 70 mm width @ 600 mm centres. ***Facings***: Two layer 15 mm Lafarge sound resisting Wallboard one side. Other side two layers of 15 mm sound resisting Wallboard fixed to resilient bar. ***Insulation***: 67 mm Acoustic Slab ***Report No***: BTC 10187A	143	63	90
 Fig. 14	***Studs***: 146 mm width @ 600 mm centres. ***Facings***: One layer of 15 mm Lafarge Firecheck Wallboard both sides ***Insulation***: 2 layers of 67 mm Acoustic Slab ***Report No***: BTC 10193A	176	53	60

Timber stud partitions

Acoustic Slab performs effectively as an infill between timber studs.

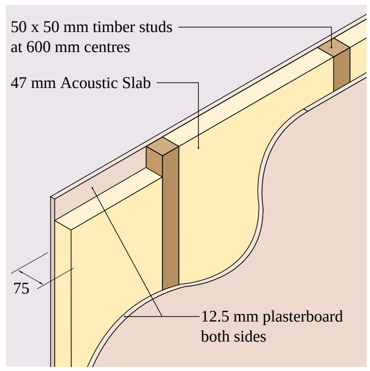
Note: NHBC require a minimum of 38 dB for specific internal partitions.

Fig. 15 Lightweight domestic timber stud partition
Average sound reduction index: $\bar{R}$ = 39 dB
Fire resistance: 30 minutes

Studs: 50 x 50 timber studs @ 600 mm centres

Facings: 12.5 mm plasterboard both sides

Insulation: 47 mm Acoustic Slab



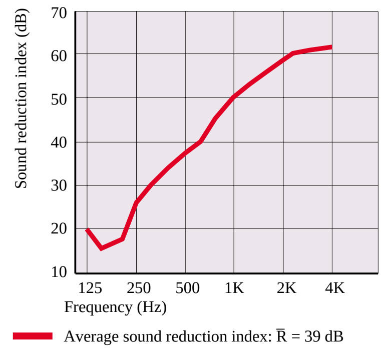


Fig. 15

Fig. 16 Typical lightweight office partition adjacent to factory
Weighted sound reduction index: R_W46 dB

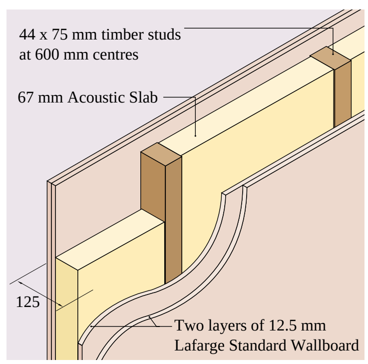
Fire resistance: 60 minutes

Studs: 44 x 75 timber studs @ 600 mm centres

Facings: Two layers of 12.5 mm Lafarge Standard Wallboard both sides

Insulation: 67 mm Acoustic Slab

Report No: L/1944/A/7



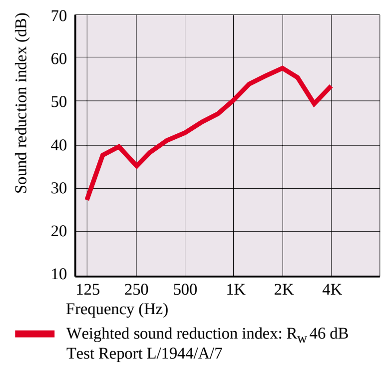


Fig. 16

Fig. 17 Non-loadbearing separating wall construction
Weighted sound reduction index: R_W59 dB

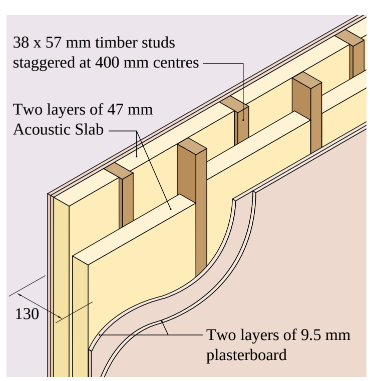
Fire resistance: 60 minutes

Studs: 38 x 57 timber studs @ 400 mm centres

Facings: Two layers of 9.5 mm plasterboard both sides

Insulation: Two layers of 47 mm Acoustic Slab

Report No: RI Test Report P7



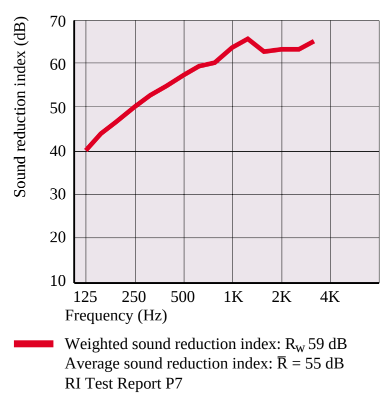


Fig. 17

The National Building Regulations

Floors, walls and partitions within single-occupancy dwellings

Regulations cover only walls and floors separating dwellings from other buildings or separate dwellings within the same building. There are no requirements for walls or floors within individual dwellings.

It is, however, important that architects and designers should provide a degree of acoustic insulation in these situations. Bathrooms and WCs for instance, should not be a source of annoyance to people in adjoining bedrooms. (See notes on page 4).

Floors

The Approved Document E/1/2/3 of the Building Regulations sets minimum sound insulation standards for floors tested in accordance with BS 2750 and BS 5821.

The standards apply to both airborne and impact sound, and separate criteria are set for new-build and conversion applications, as follows:

Airborne sound Weighted standardised level difference ($D_{nT,w}$)	Impact sound Weighted standardised pressure level ($L'_{nT,w}$)
Conversion	
48 dB	65 dB
New-build	
52 dB	61 dB

To achieve the required noise transmission losses in terms of both airborne and structure-borne sound a variety of constructions can be considered, of which the following are the more common:

Floor Type 1: Solid concrete of a specified minimum mass/unit area with a resilient floor covering (to reduce impact noise)

Floor Type 2: Solid concrete of lower mass, with a separating, sound-deadening layer between the structural floor and the floor finish (usually screed). Edge insulation, in the form of Rockwool Isolation Strip is also required to prevent flanking transmissions of structure-borne sound via adjacent walls.

(Floating floor construction)

Floor Type 3A: Timber (or steel) joisted floors, with two layers of plasterboard on the soffit, plus a sound absorbing element between the joists (Rockwool Timber Roll or Rollbatt) and a resilient layer (Rockwool Rockfloor) between subfloor and floor finish.

NOTE: It is strongly recommended that Floor Types 3B and 3C (Ribbed floors) shown in Part E of the Building Regulations are not used.

Upgrading existing floor constructions

In conversions of buildings for multiple occupancy residential use, Rockwool mineral wool rolls, slabs or batts, in conjunction with additional floor and ceiling layers, can be used to upgrade the sound insulation to acceptable levels.

Such a construction is shown in figure 18.

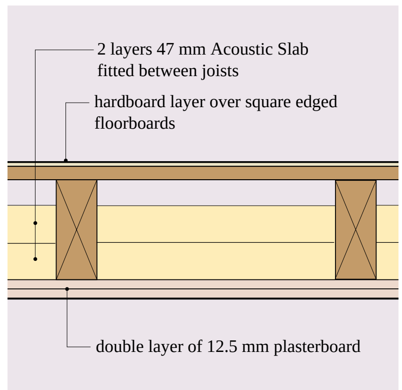


Fig.18 Upgrading the airborne sound insulation of a fixed timber floor using 2 layers of 47 mm Rockwool Acoustic Slab.
Approximate sound reduction: 43 dB.
Note that this treatment will slightly improve the impact noise resistance.

Floating floor constructions - timber joisted and concrete

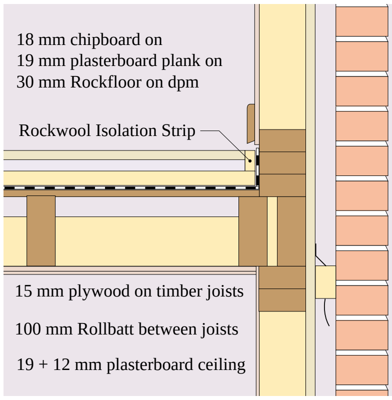


Fig.19 Timber joisted floor using Rockwool Rockfloor and Rockwool Rollbatt

Airborne sound reduction: Rw56 dB

Impact sound reduction: Ln,w62 dB

Laboratory test report L 2206/E* shows that this construction meets the requirements of the Building Regulations for separating floors between dwellings.

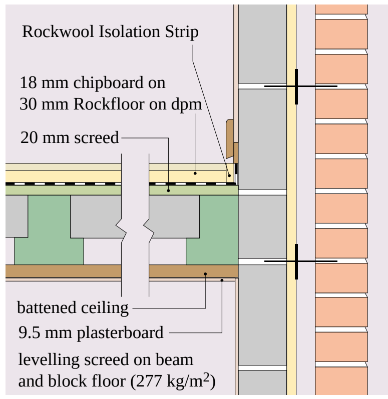


Fig.20 Beam and block floor using Rockwool Rockfloor on levelling screed

Airborne sound reduction: DnT,w52 dB

Impact sound reduction: L,nT,w 54 dB

Site test report AAD 91137* shows that this construction meets the requirements of the Building Regulations for separating floors between dwellings.

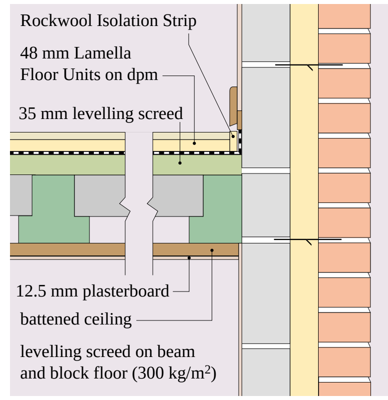


Fig.21 Beam and block floor using Rockwool Lamella Floor Units

Airborne sound reduction: Rw60 dB

Impact sound reduction: Ln,w45 dB

Laboratory test report L 2206/A* shows that this construction meets the requirements of the Building Regulations for separating floors between dwellings. Note: In-situ concrete or concrete plank floors can be substituted for the beam and block floors shown in Fig. 21 provided the overall weight is 300 kg/m2 minimum. If such a construction does not include a battened ceiling, reference should be made to Assessment Letter dated 9th July 1993 in addition to the relevant test report.

IV

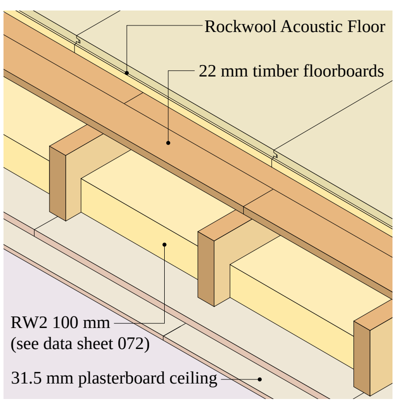


Fig.22 Timber joisted floor using Rockwool Acoustic Floor System

Airborne sound reduction: Rw53 dB

Impact sound reduction: Ln,w62 dB

Laboratory test report AIRO No. L/2388/3. *The construction consists of panels manufactured from 25 mm Rockwool bonded to 20 mm tongued and grooved, special high density cement particle board, together with 100 mm RW2 between joists. Tests show that this construction meets the requirements of the National Building Regulations and Technical Standards for conversion applications.

Note
*In order to protect the chipboard from moisture migrating from a damp structural floor, it is recommended that a dpm (e.g. polyethylene) is placed below and lapped up and over the perimeter of the insulation.

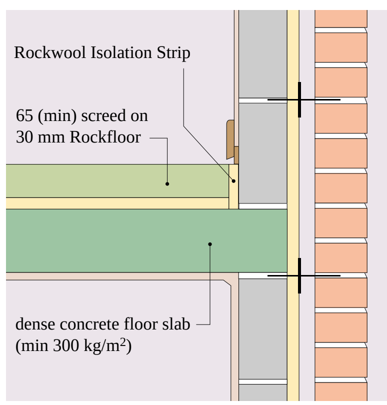


Fig.23 Dense concrete floor using Rockwool Rockfloor under screed

Airborne sound reduction: DnT,w 52 dB

Impact sound reduction: L,nT,w 53 dB

Laboratory test report L 2206/B modified by Assessment letter 9th July 1993 shows that this construction meets the requirements of the Building Regulations for separating floors between dwellings.

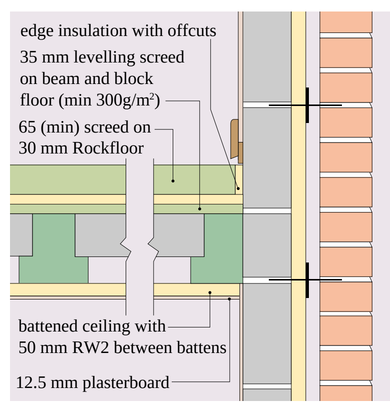


Fig.23a Screeded beam and block floor

Airborne sound reduction: Rw 59 dB

Impact sound reduction: LnT,w 44 dB

Laboratory test report L 2206/B shows that this construction meets the requirements of the Building Regulations for separating floors between dwellings.

Improving room-to-room sound attenuation above internal partitions

The integrity of sound reducing partition constructions can be maintained above suspended ceilings by the use of Rockwool insulation products.

There are two methods. The first involves filling the void between the soffit and the head of the partition with a dense insulation. The other method is to totally overlay both ceilings either side of the partition with a lightweight insulation.

The figures quoted below are based on test results obtained using a typical suspended ceiling lay-in grid tile system providing a basic 30 dB room-to-room sound attenuation:

1. Rockwool Fire Barrier hung from soffit or beam and draped over the partition head (See Fig. 22)

	Improvement	Total reduction
50 mm Fire Barrier	+12 dB	42 dB
100 mm Fire Barrier (2 x 50 mm)	+16 dB	46 dB
50 mm Fire Barrier foil faced	+14 dB	44 dB
100 mm (2 x 50 mm) Fire Barrier foil faced	+20 dB	50 dB

2. Both ceilings overlaid with lightweight insulation

47 mm Rockwool Acoustic Slab	+7 dB	37 dB
67 mm Rockwool Acoustic Slab or 100 mm Rollbatts	+10 dB	40 dB
75 mm Rockwool RWA45	+12 dB	42 dB
100 mm Rockwool RWA45 or 150 mm Rollbatts	+16 dB	46 dB
120 mm Rockwool RWA45 or 100+80 mm Rollbatts	+18 dB	48 dB

N.B. Overlaying ceiling tiles with an insulating product might impair the fire resistance of the tiles. The ceiling manufacturer should be consulted if this method is used.

IV

Rockwool Acoustic Party Wall dpc
(data sheet 028)
The use of this product as illustrated in Fig. 25 will avoid the flanking transmission commonly associated with lightweight inner leaves of blockwork. It will also afford the requisite fire resistance as defined in the Building Regulations.

Rockwool Access Floor FireStop Slab
(data sheet 061)
Access Floor FireStop Slab, used primarily for firestopping purposes, has the added advantage of reducing unwanted flanking transmission via the access floor void. The product is illustrated in Fig. 26 and is cut on site to suit access floor void depth.

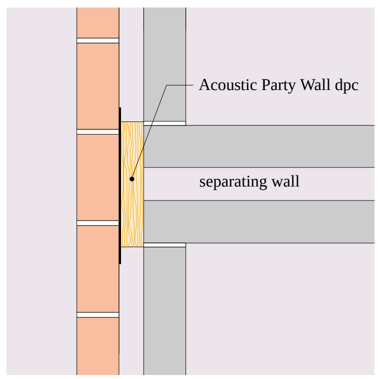


Fig.25 Acoustic Party Wall dpc

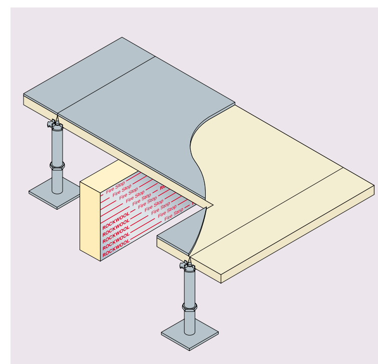


Fig.26 Access Floor FireStop Slab (typically directly under partition line)

Rockwool Fire Barrier
(data sheet 060)
A single layer of Fire Barrier will improve the room-to-room performance of the ceiling by approx. 12 dB, and by up to 20 dB when a double layer is used with foil facing. (see Fig. 24 below).

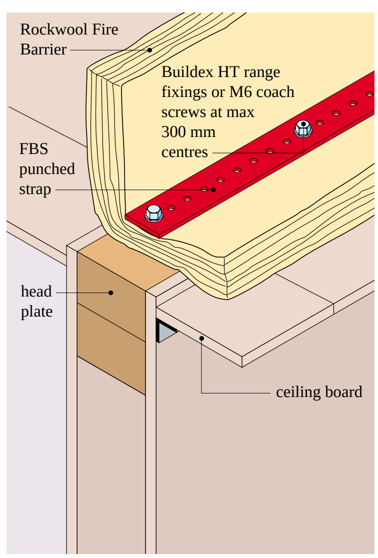


Fig.24 Fire Barrier above partition

Acoustic control

Rockwool Sound Absorption Board

The product consists of rigid, resin bonded insulation, faced on one side with black glass tissue. The structure of Sound Absorption Board makes it an ideal product for use as a sound absorber with characteristically high coefficients over a wide range of frequencies.

The product is designed for use behind perforated/slatted linings in areas such as cinemas, theatres, leisure centres, auditoria and other locations where high levels of acoustic control are required (See Fig. 27). The product can also be used for noise reduction requirements in plant room areas.

Rockwool Sound Absorption Boards are manufactured to standard dimensions of 1200 x 600 x 50 mm and 1200 x 600 x 70 mm.

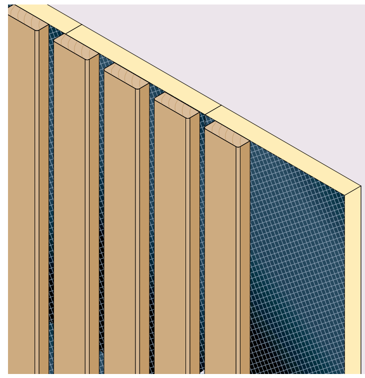


Fig.27 Rockwool Sound Absorption Board behind a slatted hardwood facing for acoustic control in an auditorium.

Rockwool Acoustic Mat

Rockwool Acoustic Mat is a flexible liner which offers a simple technique for sound absorption in temporary or permanent situations. It is ideal for both commercial and industrial use, and is typically manufactured to a standard width of 1200 mm and nominal thickness of 50 mm.

Acoustic Mat consists of a Rockwool core totally enclosed in a fire retardant fabric, which is stitched at intervals to form a quilt. It can be used in either vertical or horizontal applications.

Typically, Acoustic Mat is fixed as indicated in Fig. 28. The product can be used in recording or broadcasting studios and auditoria for effective sound control. It can also be used for sound absorption requirements in industrial environments.

Absorption coefficients are shown below the diagram.

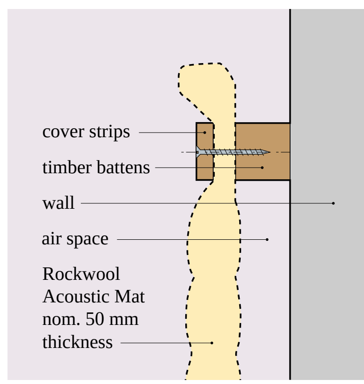
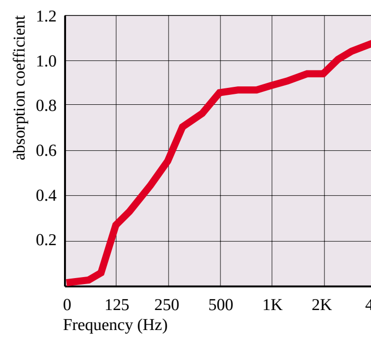


Fig.28 Rockwool Acoustic Mat showing fixing method



Rating according to EN ISO 11654: 1997
Weighted Sound Absorption Coefficient $\alpha_w = 0.85$
Test report BTC 10268A

Soffit Lining Solutions
(Data sheet 201)

Two products, High Impact Liner Board and Soffit Liner provide both effective acoustic and thermal insulation for concrete soffits.

High Impact Liner Board is a laminate of high density Rockwool Lamella Slabs bonded with an adhesive to cement bonded particle board. High Impact Liner Board provides a cost effective means of achieving excellent levels of thermal insulation coupled with fire safety and acoustic insulation. The boards offer excellent impact resistance and will significantly reduce the risk of condensation (Fig. 29).

Soffit Liners are robust Rockwool Slabs with a high quality, Class 'O' decorative finish particularly suited for underground vehicle parks and other situations requiring significant thermal insulation and acoustic absorption (Fig. 30).

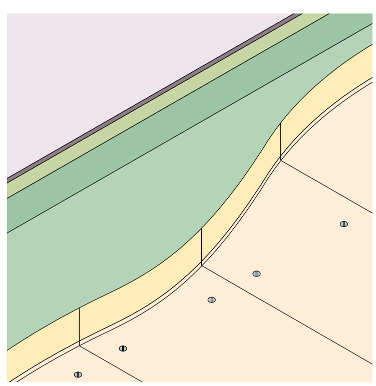


Fig.29 High Impact Liner Board

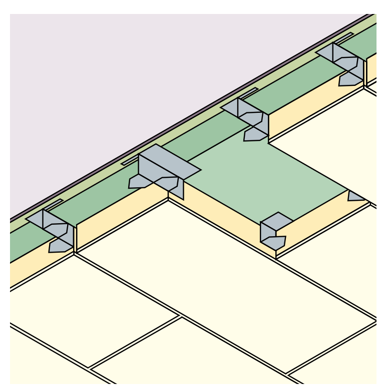


Fig.30 Soffit Liner

IV

Rockwool Hardrock Roofing Boards (data sheet 050)

A commercial flat roof incorporating 150 mm Rockwool Hardrock with a single ply waterproof membrane has been shown to achieve a sound reduction index of 44 dB (SRL Ltd test report No. C/6487/405/1). This construction is particularly suitable for buildings close to airports (see Fig. 31).

Alternatively, 80 mm Hardrock in a similar roof construction will achieve a weighted sound reduction index of 38 dB (University of Salford test reference No. 90/4/18).

Rockwool Cladding Roll (data sheet 011)

Rockwool Cladding Roll U/F or Cladding Roll Alu-faced, used as the thermal insulant between profiled steel sidewall cladding and steel sheet liner panels, provides a good sound insulation barrier. An weighted sound reduction index of Rw37 dB can be expected (see Fig. 32).

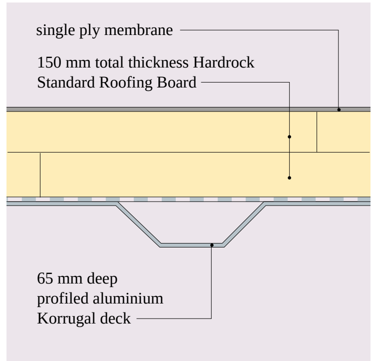


Fig.31 Flat roof detail, Stansted Airport, with Rockwool Hardrock Roofing Board - Sound reduction index = 44 dB.

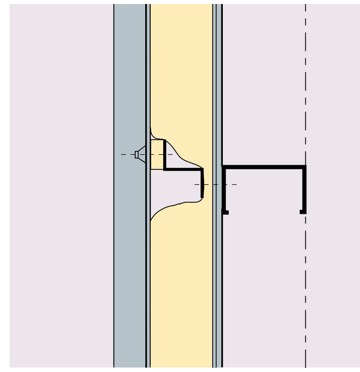


Fig.32 Sidewall cladding with Rockwool Cladding Roll and Rock-Lam Thermal Break strips - Weighted sound reduction index = Rw37 dB

IV

Rockwool-CORED composite acoustic panels

Composite Acoustic Panels consist of sheet steel faces, bonded to a Rockwool core. The constructions shown below (Figs. 33 and 34) achieve an average

32 dB sound attenuation within the range 125 Hz to 4000 Hz. The panels can be profiled to interlock at the edges. For further information on panel systems, please contact Rockwool Ltd., Systems Division.

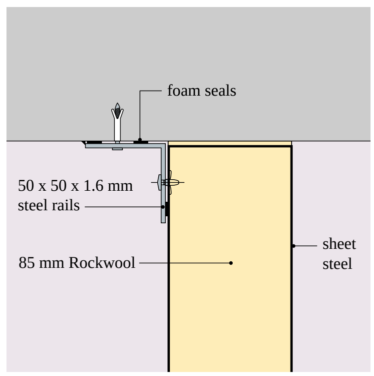


Fig.33 Support detail

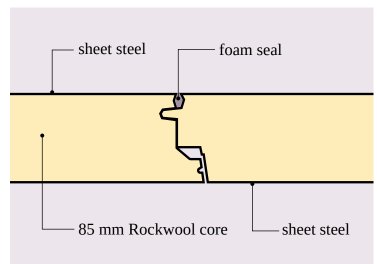
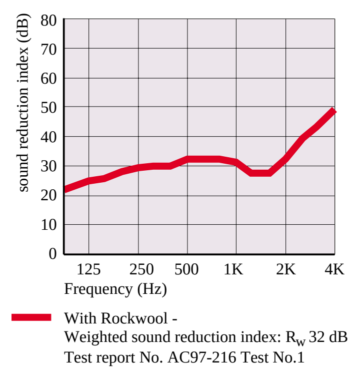


Fig.34 Edge profile



The prestigious telecommunications centre for Orange at Darlington required a high performance acoustic roofing system. Aesthetically it also demanded a system which did not involve mechanical fixings penetrating the undersurface of the 'on view' roofing deck.

The solution was an acoustic flat roofing system comprising a perforated steel deck overlaid with a non-woven glass fleece barrier, above which the insulation build-up includes 60 mm Hardrock covered with a bituminous vapour control layer and a further 70 mm of Hardrock Single-Ply Adhered boards.

The roofing system is considered environmentally friendly, has excellent thermal and fire protection capabilities as well as outstanding dimensional stability.

Industrial buildings

Selecting the right Rockwool product

The table on the facing page shows the range of specialised Rockwool acoustic products for use in industrial buildings, and in particular, for the acoustic insulation of services, plant and machinery.

The vertical columns give the location or building element, and the symbol identifies the appropriate product and its acoustic function. Cross reference to the drawing will show specific areas of application.

Key to symbols

1 *Environmental noise*

2 *Noise from outside*

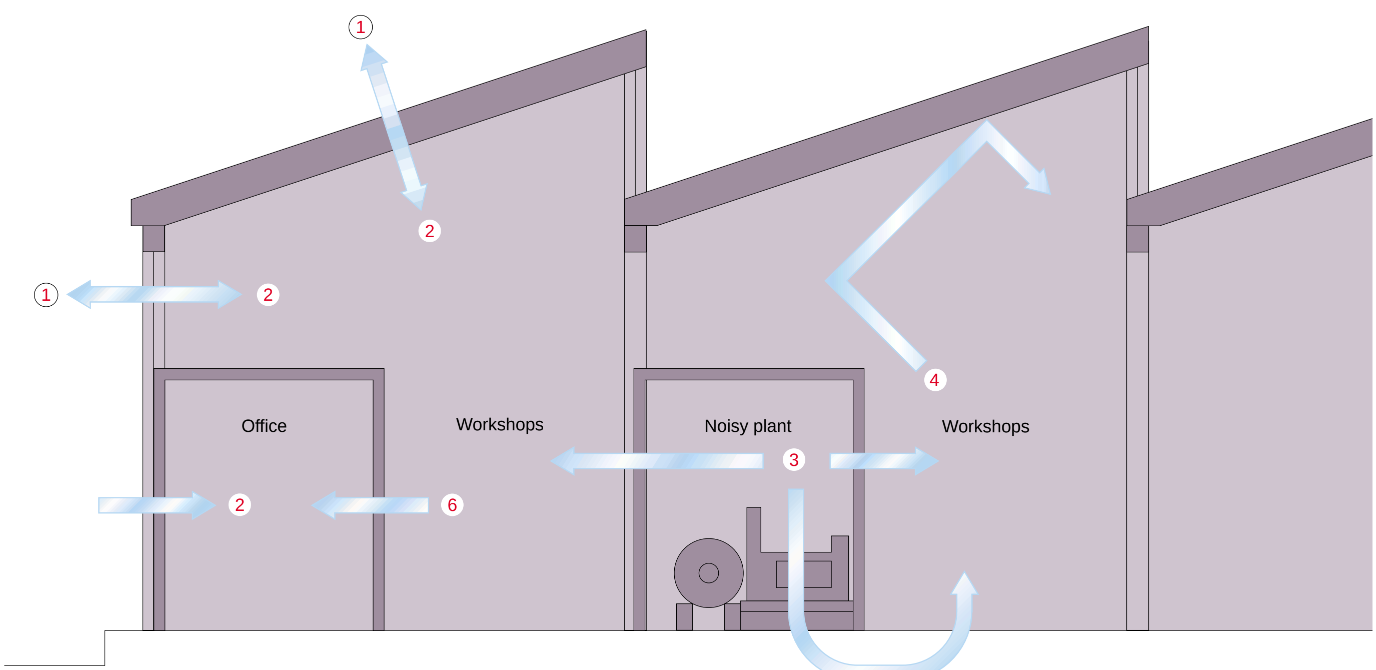
3 *Noisy plant and machinery*

4 *Internal noise levels*

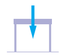
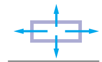
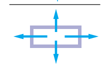
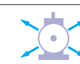
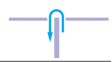
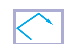
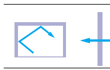
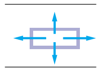
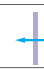
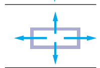
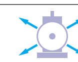
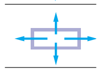
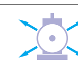
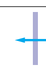
5 *Noisy ducts and piped services*

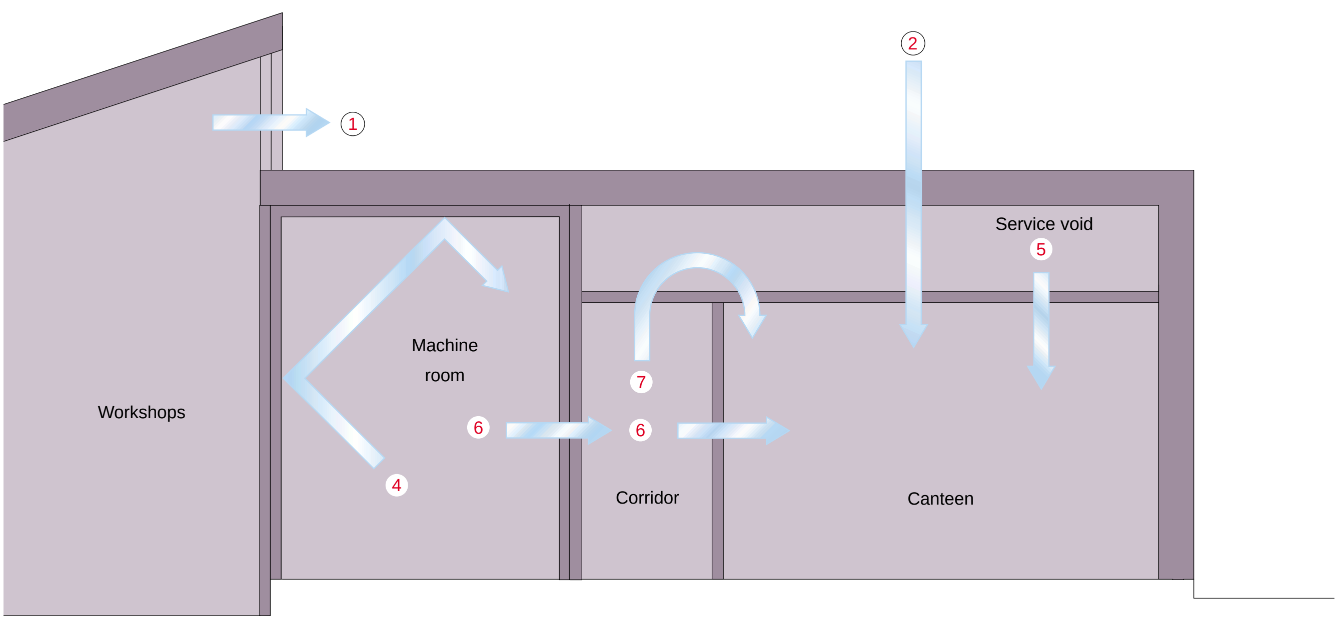
6 *Room-to-room noise (partitions)*

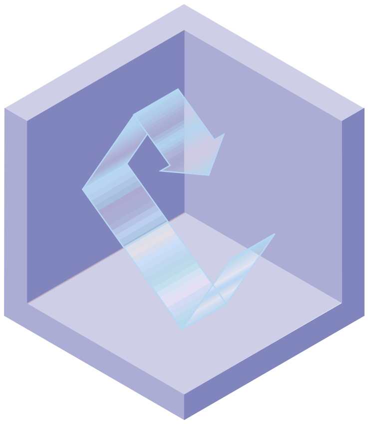
7 *Room-to-room noise (ceilings)*



Product selector

Page No	Rockwool product	External walls	Flat roofs	Suspended ceilings	Internal walls and partitions	Compartment linings	Acoustic enclosures	Ducts and pipes	Plant and machinery
19	Hardrock								
25	Process Pipe Section								
22	Wired Mat								
19	Cladding Roll								
17	Fire Barrier								
24	RW3 with tissue facing								
22	Rigid Slabs								
24	Techwrap								
18	Acoustic Mat								
23	Acoustic Elements								
23	Acoustic Panel Wall Lining System								
10-15	Acoustic Slab								
19	Composite Panels								





Industrial processes

Many industrial processes are inherently noisy and require acoustic treatment to reduce noise to acceptable levels. To protect personnel from prolonged exposure to high levels of noise, which can induce hearing loss, the The Health and Safety at Work Act (1974) and Noise at Work Regulations (1989) stipulate a “first action level” of 85 dB(A) daily personal noise exposure normalised to an eight hour working day. HSE guidance on the Noise at Work Regulations states that there is a quantifiable risk of hearing damage from exposure between 85 dB(A) (first action level) and 90 dB(A) (second action level), and a residual though small risk below 85 dB(A).

To achieve reduced noise levels, a number of steps can be taken:

1. Use quieter machines and processes

2. Enclose noisy machines individually

3. Where there are a large number of machines, treat the whole interior surface with noise absorbing panels.

(The maximum practicable reduction in environmental noise within plant rooms is 10 dB. A typical reduction in a room where only the ceiling is treated is 5 dB).

4. Apply acoustic control methods to noisy pipes, ducts and air-handling plant.

Noisy equipment

Industrial and HD Wired Mats (data sheet 120)

Rockwool Industrial and HD Wired Mats are ideally suited for use where acoustic or thermal insulation of large or irregularly shaped equipment is required.

Alternatively, where a higher degree of acoustic performance is required, or where access to machinery cannot be restricted, an enclosure may be installed around the equipment or the whole compartment may be treated.

The figures below show the two alternatives diagrammatically.

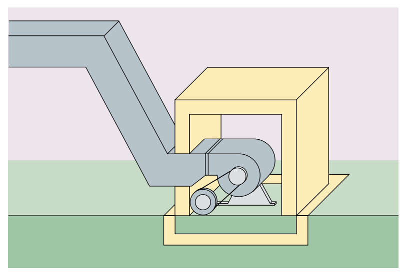


Fig.35 Isolation of plant noise source by acoustic treatment of base and individual enclosure.
Enclosure formed by steel faced partition with perforated inner sheet and incorporating Rockwool Rigid Slab (typically RW3).

Rockwool Rigid Slabs (data sheet 030)

Rigid Slabs are part of a range of rigid, semi-rigid and flexible slabs which, depending on type, can be used at temperatures of up to 800oC without any functional deterioration. They are manufactured in a variety of thicknesses and densities to suit most requirements.

The range of densities allows the designer to select the product with the correct compression strength and elastic limit properties for its structural use in the sound insulation of plant and machinery.

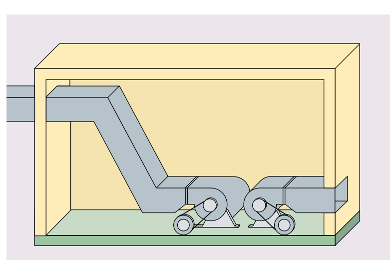


Fig.36 Acoustic lining to plant room where individual enclosure is impracticable.
Lining formed using Rockwool Acoustic Elements (see page 23)
Ventilation should be provided and any openings acoustically lined.

IV

Rockwool Acoustic Elements
Rockwool Acoustic Elements provide exceptional levels of acoustic absorption - at certain octave bands the absorption coefficient can be 1.0 (measured in accordance with ISO 354).

Particularly suitable for use in work areas subject to intense noise levels from plant and machinery, Rockwool Acoustic Elements are available in a range of sizes and are simply bolted to bulkhead and soffit areas. The perforated steel and 'impervious film' facing allows easy washdown and maintenance and provides high resistance to impact damage - especially important features in plant rooms.

Standard dimensions
Thickness: 50 mm

Outside dimensions: 600 x 1200 mm (nom.). The Elements can be easily cut to size on site and positioned to avoid services and penetrations.

Other sizes may be available to special order.

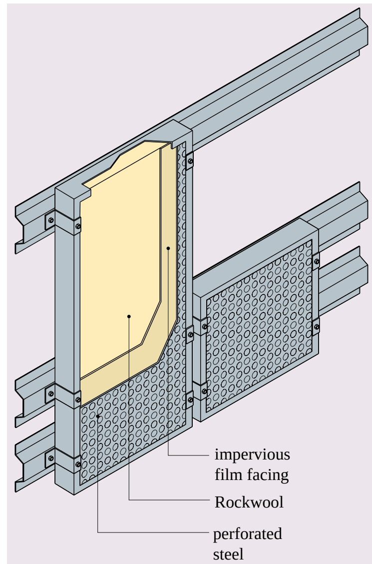
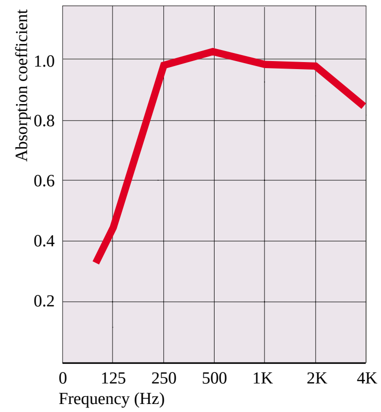


Fig.37 Rockwool Acoustic Element
Casing: Single piece, aluminium, galvanised or stainless steel sheet formed into open-backed 'cassette'. Front face perforated with 3 mm dia holes, 33% open area.
Inlay: Rockwool with plastics wrapping.
Note: Elements also available in 50 and 100 mm thicknesses and in outside dimensions to suit.



Rockwool Acoustic Element, standard size, fixed over a 50 mm cavity.
Test method: ISO 354 (1985)

IV

Rockwool Acoustic Panel Wall Lining System
The system provides a simple technique for the wall application of a sound absorbent lining protected by a perforated metal facing. The unique construction results in a panel with optimum acoustic absorption with the advantage of high impact resistance.

The system consists of lengths of retaining channel which are fixed at 1200 mm centres into which the absorber panels are held by cover strips, secured by self tapping screws.

The metal facing of the panel forms a tray into which a specially constructed Rockwool core is laminated.

No vertical supports are required other than the use of the same retaining channels at external corners or vertical terminations of the system.

The product has been tested, with a 50 mm air gap, for random incidence sound absorption to BS 3638: 1987 and the test results, shown in the graph, give the measured absorption coefficients over a range of frequencies taken at 1/3 octave intervals.

Rockwool Acoustic Panel Wall Lining System can also be fixed direct to a wall.

For further details please contact the Industrial Technical Services Department.

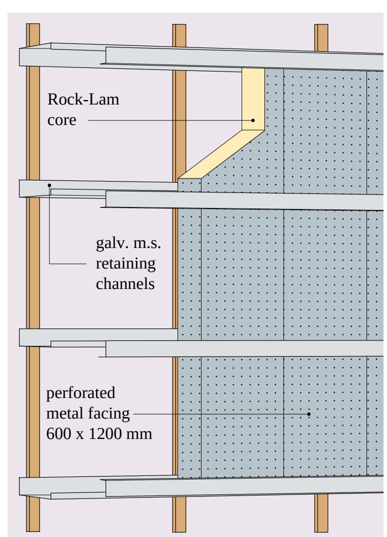
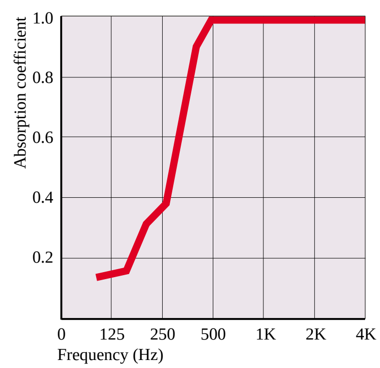


Fig.38 Rockwool Acoustic Panel Wall Lining System on 50 mm battens.



Rockwool Acoustic Acoustic Panel Wall Lining System on 50 mm battens - Average noise absorption coefficient: 0.90.

Air conditioning and ventilation ductwork

Rockwool RW3 with tissue lining
RW3 with a tissue lining applied to the exposed surface can be used as an internal lining to ducts (see Fig. 39). The perimeter of the tissue facing is returned behind the edges of the slab to help prevent delamination of the facing in turbulent or high velocity airstreams. The acoustic lining attenuates noise levels both within the duct and outside.

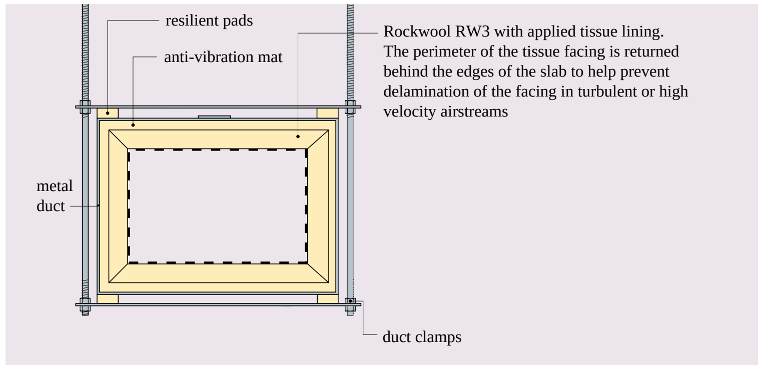


Fig.39 Internal acoustic lining to attenuate duct-borne noise - Rockwool RW3 and tissue lining on the exposed side

Rockwool Techwrap
(data sheet 131)
Techwrap is a 'high mass' faced Rockwool mineral wool comprising:

- Reinforced aluminium foil (inner)
- Rockwool, 25 mm standard (other thicknesses available to special order)
- High mass polymeric, 5 kg/m^2 standard (10 and 15 kg/m^2 also available to special order)
- Woven glass cloth
- Reinforced aluminium foil (outer)

Sound reduction test
Noise level improvements (in dB) through a tested 0.8 mm duct are given in the graph below for both single and double layers of standard 25 mm Rockwool Techwrap.

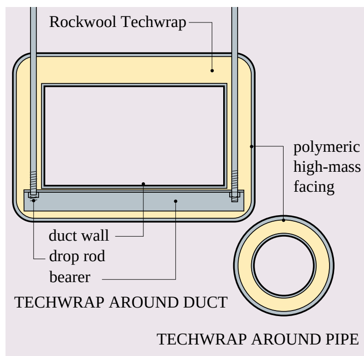
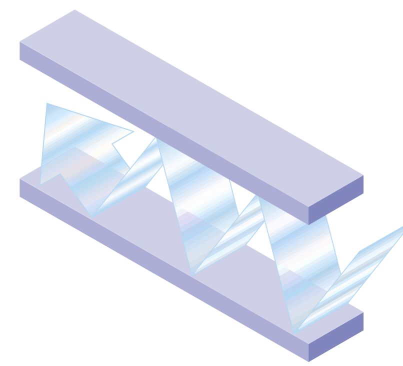


Fig.40 High level acoustic attenuation provided to ducts, pipes and enclosures - Rockwool Techwrap



Noise transmitted along an air duct

IV

Services penetrations

Figures 41 to 43 below show various techniques, using Rockwool materials, for dealing with noise transmission through a wall (or floor) caused by the penetration of ducts, pipes or other services.

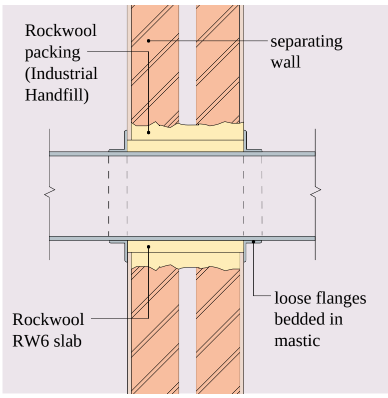


Fig.41 Reduction of structure borne noise at duct penetration - Rockwool Industrial Handfill / RW6 Slab

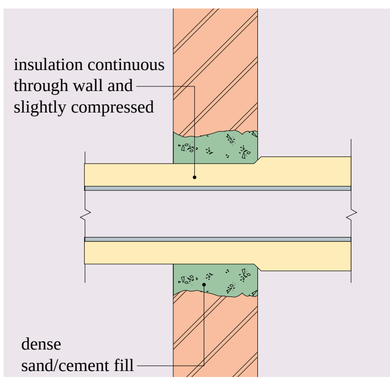


Fig.42 Penetration of structure by insulated pipework - RockLap 800 H & V Pipe Sections

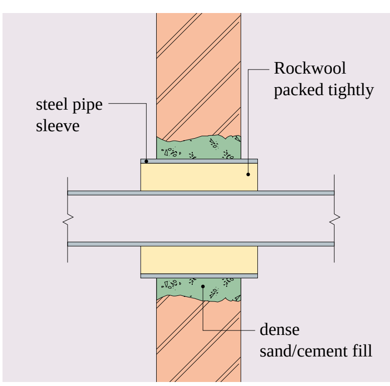


Fig.43 Alternative method with pipe sleeved through wall - using low density Rockwool packed tightly

Pipework

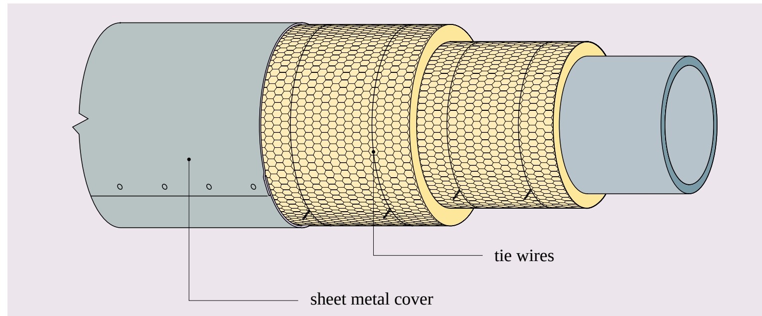


Fig.44 100 mm Wired Mat + 0.75 mm steel cladding without support rings. Pipe diameter 308 mm. The sound reduction achieved is shown in the graph below.

Rockwool Process Pipe Sections and Rockwool Wired Mats (data sheets 101 and 120)

These products can be used to significantly reduce noise emissions from pipes.

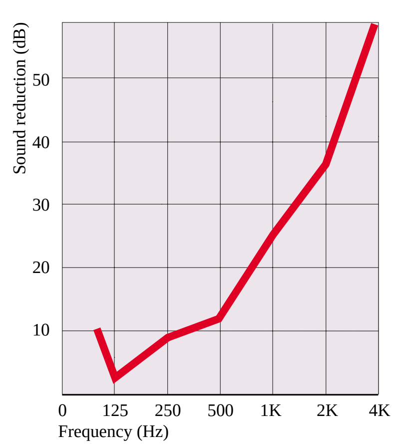
For optimum acoustic performance, the following principles should be followed.

- Process Pipe Section or Wired Mat to be installed tightly around the pipe
- Where suitable, Wired Mat to be used in preference to Pipe Section
- An outer cladding to be applied to the Rockwool insulation
- Cladding weight to be as great as practicable
- Rigid mechanical contact between pipe and outer cladding to be diligently avoided

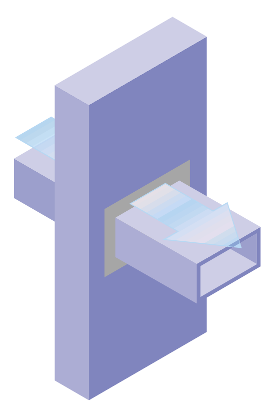
Although the sound reduction resulting from use of Rockwool pipe insulation is significantly affected by the size of the pipe treated, the example shows a typical level of performance.

It should be noted that the sound attenuation obtained at low frequencies will be less for smaller diameter pipes. In some cases, a negative insertion loss may occur at frequencies below 250 Hz.

Note: An extensive range of Rockwool Offshore products is also available to provide thermal and acoustic insulation to plant. Please contact the Rockwool Industrial Division for further details.



Sound reduction graph for construction shown in Figure 44



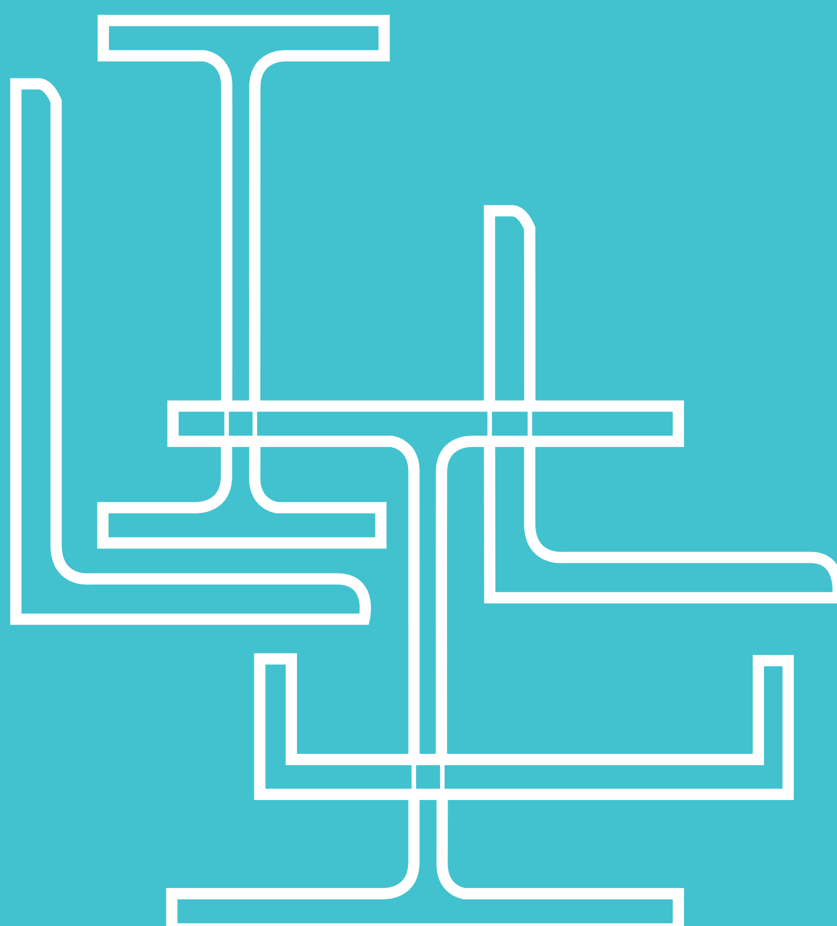
Noise transmitted via a service penetration corus

Corus Construction & Industrial

Structural sections

to BS4: Part 1: 1993 and BS EN10056: 1999

IV



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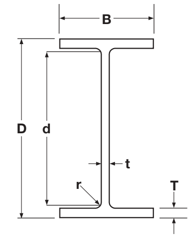


Universal beams

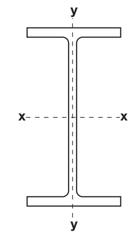
Dimensions and properties to BS 4: Part 1: 1993

Designation Serial Size	Mass per metre kg/m	Depth of Section D mm	Width of Section B mm	Thickness of Web t mm	Thickness of Flange T mm	Root Radius r mm	Depth between fillets d mm	Ratios for Local Buckling Flange B/2T	Ratios for Local Buckling Web d/t	Second Moment of Area Axis x-x cm^4	Second Moment of Area Axis y-y cm^4
† **1016 x 305 x 487**	486.6	1036.1	308.5	30.0	54.1	30.0	867.9	2.85	28.9	1021400	26720
† **1016 x 305 x 437**	436.9	1025.9	305.4	26.9	49.0	30.0	867.9	3.12	32.3	909900	23450
† **1016 x 305 x 393**	392.7	1016.0	303.0	24.4	43.9	30.0	868.2	3.45	35.6	807700	20500
† **1016 x 305 x 349**	349.4	1008.1	302.0	21.1	40.0	30.0	868.1	3.78	41.1	723100	18460
† **1016 x 305 x 314**	314.3	1000.0	300.0	19.1	35.9	30.0	868.2	4.18	45.5	644200	16230
† **1016 x 305 x 272**	272.3	990.1	300.0	16.5	31.0	30.0	868.1	4.84	52.6	554000	14000
† **1016 x 305 x 249**	248.7	980.2	300.0	16.5	26.0	30.0	868.2	5.77	52.6	481300	11750
† **1016 x 305 x 222**	222.0	970.3	300.0	16.0	21.1	30.0	868.1	7.11	54.3	408000	9546
914 x 419 x 388	388.0	921.0	420.5	21.4	36.6	24.1	799.6	5.74	37.4	719600	45440
914 x 419 x 343	343.3	911.8	418.5	19.4	32.0	24.1	799.6	6.54	41.2	625800	39160
914 x 305 x 289	289.1	926.6	307.7	19.5	32.0	19.1	824.4	4.81	42.3	504200	15600
914 x 305 x 253	253.4	918.4	305.5	17.3	27.9	19.1	824.4	5.47	47.7	436300	13300
914 x 305 x 224	224.2	910.4	304.1	15.9	23.9	19.1	824.4	6.36	51.8	376400	11240
914 x 305 x 201	200.9	903.0	303.3	15.1	20.2	19.1	824.4	7.51	54.6	325300	9423
838 x 292 x 226	226.5	850.9	293.8	16.1	26.8	17.8	761.7	5.48	47.3	339700	11360
838 x 292 x 194	193.8	840.7	292.4	14.7	21.7	17.8	761.7	6.74	51.8	279200	9066
838 x 292 x 176	175.9	834.9	291.7	14.0	18.8	17.8	761.7	7.76	54.4	246000	7799
762 x 267 x 197	196.8	769.8	268.0	15.6	25.4	16.5	686.0	5.28	44.0	240000	8175
762 x 267 x 173	173.0	762.2	266.7	14.3	21.6	16.5	686.0	6.17	48.0	205300	6850
762 x 267 x 147	146.9	754.0	265.2	12.8	17.5	16.5	686.0	7.58	53.6	168500	5455
762 x 267 x 134	133.9	750.0	264.4	12.0	15.5	16.5	686.0	8.53	57.2	150700	4788
686 x 254 x 170	170.2	692.9	255.8	14.5	23.7	15.2	615.1	5.40	42.4	170300	6630
686 x 254 x 152	152.4	687.5	254.5	13.2	21.0	15.2	615.1	6.06	46.6	150400	5784
686 x 254 x 140	140.1	683.5	253.7	12.4	19.0	15.2	615.1	6.68	49.6	136300	5183
686 x 254 x 125	125.2	677.9	253.0	11.7	16.2	15.2	615.1	7.81	52.6	118000	4383
610 x 305 x 238	238.1	635.8	311.4	18.4	31.4	16.5	540.0	4.96	29.3	209500	15840
610 x 305 x 179	179.0	620.2	307.1	14.1	23.6	16.5	540.0	6.51	38.3	153000	11410
610 x 305 x 149	149.2	612.4	304.8	11.8	19.7	16.5	540.0	7.74	45.8	125900	9308
610 x 229 x 140	139.9	617.2	230.2	13.1	22.1	12.7	547.6	5.21	41.8	111800	4505
610 x 229 x 125	125.1	612.2	229.0	11.9	19.6	12.7	547.6	5.84	46.0	98610	3932
610 x 229 x 113	113.0	607.6	228.2	11.1	17.3	12.7	547.6	6.60	49.3	87320	3434
610 x 229 x 101	101.2	602.6	227.6	10.5	14.8	12.7	547.6	7.69	52.2	75780	2915
533 x 210 x 122	122.0	544.5	211.9	12.7	21.3	12.7	476.5	4.97	37.5	76040	3388
533 x 210 x 109	109.0	539.5	210.8	11.6	18.8	12.7	476.5	5.61	41.1	66820	2943
533 x 210 x 101	101.0	536.7	210.0	10.8	17.4	12.7	476.5	6.03	44.1	61520	2692
533 x 210 x 92	92.1	533.1	209.3	10.1	15.6	12.7	476.5	6.71	47.2	55230	2389
533 x 210 x 82	82.2	528.3	208.8	9.6	13.2	12.7	476.5	7.91	49.6	47540	2007
457 x 191 x 98	98.3	467.2	192.8	11.4	19.6	10.2	407.6	4.92	35.8	45730	2347
457 x 191 x 89	89.3	463.4	191.9	10.5	17.7	10.2	407.6	5.42	38.8	41020	2089
457 x 191 x 82	82.0	460.0	191.3	9.9	16.0	10.2	407.6	5.98	41.2	37050	1871
457 x 191 x 74	74.3	457.0	190.4	9.0	14.5	10.2	407.6	6.57	45.3	33320	1671
457 x 191 x 67	67.1	453.4	189.9	8.5	12.7	10.2	407.6	7.48	48.0	29380	1452

† These dimensions are in addition to our standard range to BS4 specifications

Please consult with Corus for availability. See page 42/43.

IV

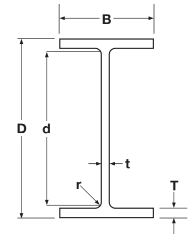


Universal beams

Dimensions and properties to BS 4: Part 1: 1993

Radius of Gyration Axis x-x cm	Radius of Gyration Axis y-y cm	Elastic Modulus Axis x-x cm^3	Elastic Modulus Axis y-y cm^3	Plastic Modulus Axis x-x cm^3	Plastic Modulus Axis y-y cm^3	Buckling Parameter u	Torsional Index x	Warping Constant H dm^6	Torsional Constant J cm^4	Area of Section cm^2	Mass per metre kg/m	Designation Serial Size
40.6	6.57	19720	1732	23200	2800	0.867	21.1	64.4	4299	620	486.6	**1016 x 305 x 487**
40.4	6.49	17740	1535	20760	2469	0.868	23.1	55.9	3185	557	436.9	**1016 x 305 x 437**
40.2	6.40	15900	1353	18540	2168	0.868	25.5	48.4	2330	500	392.7	**1016 x 305 x 393**
40.3	6.44	14350	1223	16590	1941	0.872	27.9	43.3	1718	445	349.4	**1016 x 305 x 349**
40.1	6.37	12880	1082	14850	1713	0.872	30.7	37.7	1264	400	314.3	**1016 x 305 x 314**
40.0	6.35	11190	934	12830	1470	0.873	35.0	32.2	835	347	272.3	**1016 x 305 x 272**
39.0	6.09	9821	784	11350	1245	0.861	39.9	26.8	582	317	248.7	**1016 x 305 x 249**
38.0	5.81	8409	636	9807	1020	0.850	45.7	21.5	390	283	222.0	**1016 x 305 x 222**
38.2	9.59	15630	2161	17670	3341	0.885	26.7	88.9	1734	494	388.0	**914 x 419 x 388**
37.8	9.46	13730	1871	15480	2890	0.883	30.1	75.8	1193	437	343.3	**914 x 419 x 343**
37.0	6.51	10880	1014	12570	1601	0.867	31.9	31.2	926	368	289.1	**914 x 305 x 289**
36.8	6.42	9501	871	10940	1371	0.866	36.2	26.4	626	323	253.4	**914 x 305 x 253**
36.3	6.27	8269	739	9535	1163	0.861	41.3	22.1	422	286	224.2	**914 x 305 x 224**
35.7	6.07	7204	621	8351	982	0.854	46.8	18.4	291	256	200.9	**914 x 305 x 201**
34.3	6.27	7985	773	9155	1212	0.870	35.0	19.3	514	289	226.5	**838 x 292 x 226**
33.6	6.06	6641	620	7640	974	0.862	41.6	15.2	306	247	193.8	**838 x 292 x 194**
33.1	5.90	5893	535	6808	842	0.856	46.5	13.0	221	224	175.9	**838 x 292 x 176**
30.9	5.71	6234	610	7167	959	0.869	33.2	11.3	404	251	196.8	**762 x 267 x 197**
30.5	5.58	5387	514	6198	807	0.864	38.1	9.39	267	220	173.0	**762 x 267 x 173**
30.0	5.40	4470	411	5156	647	0.858	45.2	7.40	159	187	146.9	**762 x 267 x 147**
29.7	5.30	4018	362	4644	570	0.854	49.8	6.46	119	171	133.9	**762 x 267 x 134**
28.0	5.53	4916	518	5631	811	0.872	31.8	7.42	308	217	170.2	**686 x 254 x 170**
27.8	5.46	4374	455	5000	710	0.871	35.5	6.42	220	194	152.4	**686 x 254 x 152**
27.6	5.39	3987	409	4558	638	0.868	38.7	5.72	169	178	140.1	**686 x 254 x 140**
27.2	5.24	3481	346	3994	542	0.862	43.9	4.80	116	159	125.2	**686 x 254 x 125**
26.3	7.23	6589	1017	7486	1574	0.886	21.3	14.5	785	303	238.1	**610 x 305 x 238**
25.9	7.07	4935	743	5547	1144	0.886	27.7	10.2	340	228	179.0	**610 x 305 x 179**
25.7	7.00	4111	611	4594	937	0.886	32.7	8.17	200	190	149.2	**610 x 305 x 149**
25.0	5.03	3622	391	4142	611	0.875	30.6	3.99	216	178	139.9	**610 x 229 x 140**
24.9	4.97	3221	343	3676	535	0.873	34.1	3.45	154	159	125.1	**610 x 229 x 125**
24.6	4.88	2874	301	3281	469	0.870	38.0	2.99	111	144	113.0	**610 x 229 x 113**
24.2	4.75	2515	256	2881	400	0.864	43.1	2.52	77.0	129	101.2	**610 x 229 x 101**
22.1	4.67	2793	320	3196	500	0.877	27.6	2.32	178	155	122.0	**533 x 210 x 122**
21.9	4.60	2477	279	2828	436	0.875	30.9	1.99	126	139	109.0	**533 x 210 x 109**
21.9	4.57	2292	256	2612	399	0.874	33.2	1.81	101	129	101.0	**533 x 210 x 101**
21.7	4.51	2072	228	2360	356	0.872	36.5	1.60	75.7	117	92.1	**533 x 210 x 92**
21.3	4.38	1800	192	2059	300	0.864	41.6	1.33	51.5	105	82.2	**533 x 210 x 82**
19.1	4.33	1957	243	2232	379	0.881	25.7	1.18	121	125	98.3	**457 x 191 x 98**
19.0	4.29	1770	218	2014	338	0.880	28.3	1.04	90.7	114	89.3	**457 x 191 x 89**
18.8	4.23	1611	196	1831	304	0.877	30.9	0.922	69.2	104	82.0	**457 x 191 x 82**
18.8	4.20	1458	176	1653	272	0.877	33.9	0.818	51.8	94.6	74.3	**457 x 191 x 74**
18.5	4.12	1296	153	1471	237	0.872	37.9	0.705	37.1	85.5	67.1	**457 x 191 x 67**

IV

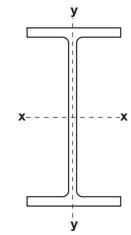


Universal beams

Dimensions and properties to BS 4: Part 1: 1993

IV

Designation Serial Size	Mass per metre kg/m	Depth of Section D mm	Width of Section B mm	Thickness of Web t mm	Thickness of Flange T mm	Root Radius r mm	Depth between fillets d mm	Ratios for Local Buckling: Flange B/2T	Ratios for Local Buckling: Web d/t	Second Moment of Area: Axis x-x cm^4	Second Moment of Area: Axis y-y cm^4
457 x 152 x 82	82.1	465.8	155.3	10.5	18.9	10.2	407.6	4.11	38.8	36590	1185
457 x 152 x 74	74.2	462.0	154.4	9.6	17.0	10.2	407.6	4.54	42.5	32670	1047
457 x 152 x 67	67.2	458.0	153.8	9.0	15.0	10.2	407.6	5.13	45.3	28930	913
457 x 152 x 60	59.8	454.6	152.9	8.1	13.3	10.2	407.6	5.75	50.3	25500	795
457 x 152 x 52	52.3	449.8	152.4	7.6	10.9	10.2	407.6	6.99	53.6	21370	645
406 x 178 x 74	74.2	412.8	179.5	9.5	16.0	10.2	360.4	5.61	37.9	27310	1545
406 x 178 x 67	67.1	409.4	178.8	8.8	14.3	10.2	360.4	6.25	41.0	24330	1365
406 x 178 x 60	60.1	406.4	177.9	7.9	12.8	10.2	360.4	6.95	45.6	21600	1203
406 x 178 x 54	54.1	402.6	177.7	7.7	10.9	10.2	360.4	8.15	46.8	18720	1021
406 x 140 x 46	46.0	403.2	142.2	6.8	11.2	10.2	360.4	6.35	53.0	15690	538
406 x 140 x 39	39.0	398.0	141.8	6.4	8.6	10.2	360.4	8.24	56.3	12510	410
356 x 171 x 67	67.1	363.4	173.2	9.1	15.7	10.2	311.6	5.52	34.2	19460	1362
356 x 171 x 57	57.0	358.0	172.2	8.1	13.0	10.2	311.6	6.62	38.5	16040	1108
356 x 171 x 51	51.0	355.0	171.5	7.4	11.5	10.2	311.6	7.46	42.1	14140	968
356 x 171 x 45	45.0	351.4	171.1	7.0	9.7	10.2	311.6	8.82	44.5	12070	811
356 x 127 x 39	39.1	353.4	126.0	6.6	10.7	10.2	311.6	5.89	47.2	10170	358
356 x 127 x 33	33.1	349.0	125.4	6.0	8.5	10.2	311.6	7.38	51.9	8249	280
305 x 165 x 54	54.0	310.4	166.9	7.9	13.7	8.9	265.2	6.09	33.6	11700	1063
305 x 165 x 46	46.1	306.6	165.7	6.7	11.8	8.9	265.2	7.02	39.6	9899	896
305 x 165 x 40	40.3	303.4	165.0	6.0	10.2	8.9	265.2	8.09	44.2	8503	764
305 x 127 x 48	48.1	311.0	125.3	9.0	14.0	8.9	265.2	4.48	29.5	9575	461
305 x 127 x 42	41.9	307.2	124.3	8.0	12.1	8.9	265.2	5.14	33.2	8196	389
305 x 127 x 37	37.0	304.4	123.4	7.1	10.7	8.9	265.2	5.77	37.4	7171	336
305 x 102 x 33	32.8	312.7	102.4	6.6	10.8	7.6	275.9	4.74	41.8	6501	194
305 x 102 x 28	28.2	308.7	101.8	6.0	8.8	7.6	275.9	5.78	46.0	5366	155
305 x 102 x 25	24.8	305.1	101.6	5.8	7.0	7.6	275.9	7.26	47.6	4455	123
254 x 146 x 43	43.0	259.6	147.3	7.2	12.7	7.6	219.0	5.80	30.4	6544	677
254 x 146 x 37	37.0	256.0	146.4	6.3	10.9	7.6	219.0	6.72	34.8	5537	571
254 x 146 x 31	31.1	251.4	146.1	6.0	8.6	7.6	219.0	8.49	36.5	4413	448
254 x 102 x 28	28.3	260.4	102.2	6.3	10.0	7.6	225.2	5.11	35.7	4005	179
254 x 102 x 25	25.2	257.2	101.9	6.0	8.4	7.6	225.2	6.07	37.5	3415	149
254 x 102 x 22	22.0	254.0	101.6	5.7	6.8	7.6	225.2	7.47	39.5	2841	119
203 x 133 x 30	30.0	206.8	133.9	6.4	9.6	7.6	172.4	6.97	26.9	2896	385
203 x 133 x 25	25.1	203.2	133.2	5.7	7.8	7.6	172.4	8.54	30.2	2340	308
203 x 102 x 23	23.1	203.2	101.8	5.4	9.3	7.6	169.4	5.47	31.4	2105	164
178 x 102 x 19	19.0	177.8	101.2	4.8	7.9	7.6	146.8	6.41	30.6	1356	137
152 x 89 x 16	16.0	152.4	88.7	4.5	7.7	7.6	121.8	5.76	27.1	834	89.8
127 x 76 x 13	13.0	127.0	76.0	4.0	7.6	7.6	96.6	5.00	24.2	473	55.7

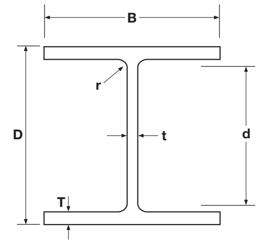


Universal beams

Dimensions and properties to BS 4: Part 1: 1993

Radius of Gyration Axis x-x cm	Radius of Gyration Axis y-y cm	Elastic Modulus Axis x-x cm^3	Elastic Modulus Axis y-y cm^3	Plastic Modulus Axis x-x cm^3	Plastic Modulus Axis y-y cm^3	Buckling Parameter u	Torsional Index x	Warping Constant H dm^6	Torsional Constant J cm^4	Area of Section cm^2	Mass per metre kg/m	Designation Serial Size
18.7	3.37	1571	153	1811	240	0.873	27.4	0.591	89.2	105	82.1	**457 x 152 x 82**
18.6	3.33	1414	136	1627	213	0.873	30.1	0.518	65.9	94.5	74.2	**457 x 152 x 74**
18.4	3.27	1263	119	1453	187	0.869	33.6	0.448	47.7	85.6	67.2	**457 x 152 x 67**
18.3	3.23	1122	104	1287	163	0.868	37.5	0.387	33.8	76.2	59.8	**457 x 152 x 60**
17.9	3.11	950	84.6	1096	133	0.859	43.9	0.311	21.4	66.6	52.3	**457 x 152 x 52**
17.0	4.04	1323	172	1501	267	0.882	27.6	0.608	62.8	94.5	74.2	**406 x 178 x 74**
16.9	3.99	1189	153	1346	237	0.880	30.5	0.533	46.1	85.5	67.1	**406 x 178 x 67**
16.8	3.97	1063	135	1199	209	0.880	33.8	0.466	33.3	76.5	60.1	**406 x 178 x 60**
16.5	3.85	930	115	1055	178	0.871	38.3	0.392	23.1	69.0	54.1	**406 x 178 x 54**
16.4	3.03	778	75.7	888	118	0.871	38.9	0.207	19.0	58.6	46.0	**406 x 140 x 46**
15.9	2.87	629	57.8	724	90.8	0.858	47.5	0.155	10.7	49.7	39.0	**406 x 140 x 39**
15.1	3.99	1071	157	1211	243	0.886	24.4	0.412	55.7	85.5	67.1	**356 x 171 x 67**
14.9	3.91	896	129	1010	199	0.882	28.8	0.330	33.4	72.6	57.0	**356 x 171 x 57**
14.8	3.86	796	113	896	174	0.881	32.1	0.286	23.8	64.9	51.0	**356 x 171 x 51**
14.5	3.76	687	94.8	775	147	0.874	36.8	0.237	15.8	57.3	45.0	**356 x 171 x 45**
14.3	2.68	576	56.8	659	89.1	0.871	35.2	0.105	15.1	49.8	39.1	**356 x 127 x 39**
14.0	2.58	473	44.7	543	70.3	0.863	42.2	0.0812	8.79	42.1	33.1	**356 x 127 x 33**
13.0	3.93	754	127	846	196	0.889	23.6	0.234	34.8	68.8	54.0	**305 x 165 x 54**
13.0	3.90	646	108	720	166	0.891	27.1	0.195	22.2	58.7	46.1	**305 x 165 x 46**
12.9	3.86	560	92.6	623	142	0.889	31.0	0.164	14.7	51.3	40.3	**305 x 165 x 40**
12.5	2.74	616	73.6	711	116	0.873	23.3	0.102	31.8	61.2	48.1	**305 x 127 x 48**
12.4	2.70	534	62.6	614	98.4	0.872	26.5	0.0846	21.1	53.4	41.9	**305 x 127 x 42**
12.3	2.67	471	54.5	539	85.4	0.872	29.7	0.0725	14.8	47.2	37.0	**305 x 127 x 37**
12.5	2.15	416	37.9	481	60.0	0.866	31.6	0.0442	12.2	41.8	32.8	**305 x 102 x 33**
12.2	2.08	348	30.5	403	48.5	0.859	37.4	0.0349	7.40	35.9	28.2	**305 x 102 x 28**
11.9	1.97	292	24.2	342	38.8	0.846	43.4	0.0273	4.77	31.6	24.8	**305 x 102 x 25**
10.9	3.52	504	92.0	566	141	0.891	21.2	0.103	23.9	54.8	43.0	**254 x 146 x 43**
10.8	3.48	433	78.0	483	119	0.890	24.3	0.0857	15.3	47.2	37.0	**254 x 146 x 37**
10.5	3.36	351	61.3	393	94.1	0.880	29.6	0.0660	8.55	39.7	31.1	**254 x 146 x 31**
10.5	2.22	308	34.9	353	54.8	0.874	27.5	0.0280	9.57	36.1	28.3	**254 x 102 x 28**
10.3	2.15	266	29.2	306	46.0	0.866	31.5	0.0230	6.42	32.0	25.2	**254 x 102 x 25**
10.1	2.06	224	23.5	259	37.3	0.856	36.4	0.0182	4.15	28.0	22.0	**254 x 102 x 22**
8.71	3.17	280	57.5	314	88.2	0.881	21.5	0.0374	10.3	38.2	30.0	**203 x 133 x 30**
8.56	3.10	230	46.2	258	70.9	0.877	25.6	0.0294	5.96	32.0	25.1	**203 x 133 x 25**
8.46	2.36	207	32.2	234	49.8	0.888	22.5	0.0154	7.02	29.4	23.1	**203 x 102 x 23**
7.48	2.37	153	27.0	171	41.6	0.888	22.6	0.0099	4.41	24.3	19.0	**178 x 102 x 19**
6.41	2.10	109	20.2	123	31.2	0.890	19.6	0.0047	3.56	20.3	16.0	**152 x 89 x 16**
5.35	1.84	74.6	14.7	84.2	22.6	0.895	16.3	0.0020	2.85	16.5	13.0	**127 x 76 x 13**

IV



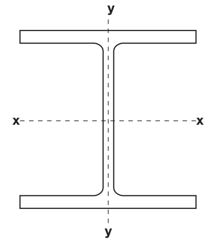
Universal columns

Dimensions and properties to BS 4: Part 1: 1993

Designation Serial Size	Mass per metre kg/m	Depth of Section D mm	Width of Section B mm	Thickness of Web t mm	Thickness of Flange T mm	Root Radius r mm	Depth between fillets d mm	Ratios for Local Buckling Flange B/2T	Ratios for Local Buckling Web d/t	Second Moment of Area Axis x-x cm^4	Second Moment of Area Axis y-y cm^4
356 x 406 x 634	633.9	474.6	424.0	47.6	77.0	15.2	290.2	2.75	6.10	274800	98130
356 x 406 x 551	551.0	455.6	418.5	42.1	67.5	15.2	290.2	3.10	6.89	226900	82670
356 x 406 x 467	467.0	436.6	412.2	35.8	58.0	15.2	290.2	3.55	8.11	183000	67830
356 x 406 x 393	393.0	419.0	407.0	30.6	49.2	15.2	290.2	4.14	9.48	146600	55370
356 x 406 x 340	339.9	406.4	403.0	26.6	42.9	15.2	290.2	4.70	10.9	122500	46850
356 x 406 x 287	287.1	393.6	399.0	22.6	36.5	15.2	290.2	5.47	12.8	99880	38680
356 x 406 x 235	235.1	381.0	394.8	18.4	30.2	15.2	290.2	6.54	15.8	79080	30990
356 x 368 x 202	201.9	374.6	374.7	16.5	27.0	15.2	290.2	6.94	17.6	66260	23690
356 x 368 x 177	177.0	368.2	372.6	14.4	23.8	15.2	290.2	7.83	20.2	57120	20530
356 x 368 x 153	152.9	362.0	370.5	12.3	20.7	15.2	290.2	8.95	23.6	48590	17550
356 x 368 x 129	129.0	355.6	368.6	10.4	17.5	15.2	290.2	10.5	27.9	40250	14610
305 x 305 x 283	282.9	365.3	322.2	26.8	44.1	15.2	246.7	3.65	9.21	78870	24630
305 x 305 x 240	240.0	352.5	318.4	23.0	37.7	15.2	246.7	4.22	10.7	64200	20310
305 x 305 x 198	198.1	339.9	314.5	19.1	31.4	15.2	246.7	5.01	12.9	50900	16300
305 x 305 x 158	158.1	327.1	311.2	15.8	25.0	15.2	246.7	6.22	15.6	38750	12570
305 x 305 x 137	136.9	320.5	309.2	13.8	21.7	15.2	246.7	7.12	17.9	32810	10700
305 x 305 x 118	117.9	314.5	307.4	12.0	18.7	15.2	246.7	8.22	20.6	27670	9059
305 x 305 x 97	96.9	307.9	305.3	9.9	15.4	15.2	246.7	9.91	24.9	22250	7308
254 x 254 x 167	167.1	289.1	265.2	19.2	31.7	12.7	200.3	4.18	10.4	30000	9870
254 x 254 x 132	132.0	276.3	261.3	15.3	25.3	12.7	200.3	5.16	13.1	22530	7531
254 x 254 x 107	107.1	266.7	258.8	12.8	20.5	12.7	200.3	6.31	15.6	17510	5928
254 x 254 x 89	88.9	260.3	256.3	10.3	17.3	12.7	200.3	7.41	19.4	14270	4857
254 x 254 x 73	73.1	254.1	254.6	8.6	14.2	12.7	200.3	8.96	23.3	11410	3908
203 x 203 x 86	86.1	222.2	209.1	12.7	20.5	10.2	160.8	5.10	12.7	9449	3127
203 x 203 x 71	71.0	215.8	206.4	10.0	17.3	10.2	160.8	5.97	16.1	7618	2537
203 x 203 x 60	60.0	209.6	205.8	9.4	14.2	10.2	160.8	7.25	17.1	6125	2065
203 x 203 x 52	52.0	206.2	204.3	7.9	12.5	10.2	160.8	8.17	20.4	5259	1778
203 x 203 x 46	46.1	203.2	203.6	7.2	11.0	10.2	160.8	9.25	22.3	4568	1548
152 x 152 x 37	37.0	161.8	154.4	8.0	11.5	7.6	123.6	6.71	15.5	2210	706
152 x 152 x 30	30.0	157.6	152.9	6.5	9.4	7.6	123.6	8.13	19.0	1748	560
152 x 152 x 23	23.0	152.4	152.2	5.8	6.8	7.6	123.6	11.2	21.3	1250	400

IV

Please consult with Corus for availability. See page 42/43.



Universal columns

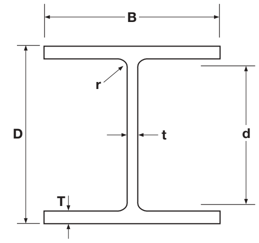
Dimensions and properties to BS 4: Part 1: 1993

Radius of Gyration		Elastic Modulus		Plastic Modulus		Buckling Parameter	Torsional Index	Warping Constant	Torsional Constant	Area of Section	Mass per metre	Designation
Axis x-x cm	Axis y-y cm	Axis x-x cm^3	Axis y-y cm^3	Axis x-x cm^3	Axis y-y cm^3	u	x	H dm^6	J cm^4	cm^2	kg/m	Serial Size
18.4	11.0	11580	4629	14240	7108	0.843	5.46	38.8	13720	808	633.9	**356 x 406 x 634**
18.0	10.9	9962	3951	12080	6058	0.841	6.05	31.1	9240	702	551.0	**356 x 406 x 551**
17.5	10.7	8383	3291	10000	5034	0.839	6.86	24.3	5809	595	467.0	**356 x 406 x 467**
17.1	10.5	6998	2721	8222	4154	0.837	7.86	18.9	3545	501	393.0	**356 x 406 x 393**
16.8	10.4	6031	2325	6999	3544	0.836	8.85	15.5	2343	433	339.9	**356 x 406 x 340**
16.5	10.3	5075	1939	5812	2949	0.835	10.2	12.3	1441	366	287.1	**356 x 406 x 287**
16.3	10.2	4151	1570	4687	2383	0.834	12.1	9.54	812	299	235.1	**356 x 406 x 235**
16.1	9.60	3538	1264	3972	1920	0.844	13.4	7.16	558	257	201.9	**356 x 368 x 202**
15.9	9.54	3103	1102	3455	1671	0.844	15.0	6.09	381	226	177.0	**356 x 368 x 177**
15.8	9.49	2684	948	2965	1435	0.844	17.0	5.11	251	195	152.9	**356 x 368 x 153**
15.6	9.43	2264	793	2479	1199	0.844	19.9	4.18	153	164	129.0	**356 x 368 x 129**
14.8	8.27	4318	1529	5105	2342	0.855	7.65	6.35	2034	360	282.9	**305 x 305 x 283**
14.5	8.15	3643	1276	4247	1951	0.854	8.74	5.03	1271	306	240.0	**305 x 305 x 240**
14.2	8.04	2995	1037	3440	1581	0.854	10.2	3.88	734	252	198.1	**305 x 305 x 198**
13.9	7.90	2369	808	2680	1230	0.851	12.5	2.87	378	201	158.1	**305 x 305 x 158**
13.7	7.83	2048	692	2297	1053	0.851	14.2	2.39	249	174	136.9	**305 x 305 x 137**
13.6	7.77	1760	589	1958	895	0.850	16.2	1.98	161	150	117.9	**305 x 305 x 118**
13.4	7.69	1445	479	1592	726	0.850	19.3	1.56	91.2	123	96.9	**305 x 305 x 97**
11.9	6.81	2075	744	2424	1137	0.851	8.49	1.63	626	213	167.1	**254 x 254 x 167**
11.6	6.69	1631	576	1869	878	0.850	10.3	1.19	319	168	132.0	**254 x 254 x 132**
11.3	6.59	1313	458	1484	697	0.848	12.4	0.898	172	136	107.1	**254 x 254 x 107**
11.2	6.55	1096	379	1224	575	0.850	14.5	0.717	102	113	88.9	**254 x 254 x 89**
11.1	6.48	898	307	992	465	0.849	17.3	0.562	57.6	93.1	73.1	**254 x 254 x 73**
9.28	5.34	850	299	977	456	0.850	10.2	0.318	137	110	86.1	**203 x 203 x 86**
9.18	5.30	706	246	799	374	0.853	11.9	0.250	80.2	90.4	71.0	**203 x 203 x 71**
8.96	5.20	584	201	656	305	0.846	14.1	0.197	47.2	76.4	60.0	**203 x 203 x 60**
8.91	5.18	510	174	567	264	0.848	15.8	0.167	31.8	66.3	52.0	**203 x 203 x 52**
8.82	5.13	450	152	497	231	0.847	17.7	0.143	22.2	58.7	46.1	**203 x 203 x 46**
6.85	3.87	273	91.5	309	140	0.848	13.3	0.0399	19.2	47.1	37.0	**152 x 152 x 37**
6.76	3.83	222	73.3	248	112	0.849	16.0	0.0308	10.5	38.3	30.0	**152 x 152 x 30**
6.54	3.70	164	52.6	182	80.2	0.840	20.7	0.0212	4.63	29.2	23.0	**152 x 152 x 23**

IV

Universal bearing piles

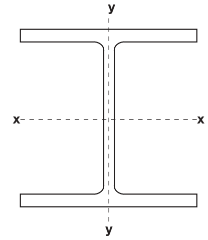
Dimensions and properties to BS 4: Part 1: 1993



Designation Serial Size	Mass per metre kg/m	Depth of Section D mm	Width of Section B mm	Thickness of Web t mm	Thickness of Flange T mm	Root Radius r mm	Depth between fillets d mm	Ratios for Local Buckling Flange B/2T	Ratios for Local Buckling Web d/t	Second Moment of Area Axis x-x cm^4	Second Moment of Area Axis y-y cm^4
356 x 368 x 174	173.9	361.4	378.5	20.3	20.4	15.2	290.2	9.28	14.3	51010	18460
356 x 368 x 152	152.0	356.4	376.0	17.8	17.9	15.2	290.2	10.5	16.3	43970	15880
356 x 368 x 133	133.0	352.0	373.8	15.6	15.7	15.2	290.2	11.9	18.6	37980	13680
356 x 368 x 109	108.9	346.4	371.0	12.8	12.9	15.2	290.2	14.4	22.7	30630	10990
305 x 305 x 223	222.9	337.9	325.7	30.3	30.4	15.2	246.7	5.36	8.14	52700	17580
305 x 305 x 186	186.0	328.3	320.9	25.5	25.6	15.2	246.7	6.27	9.67	42610	14140
305 x 305 x 149	149.1	318.5	316.0	20.6	20.7	15.2	246.7	7.63	12.0	33070	10910
305 x 305 x 126	126.1	312.3	312.9	17.5	17.6	15.2	246.7	8.89	14.1	27410	9002
305 x 305 x 110	110.0	307.9	310.7	15.3	15.4	15.2	246.7	10.1	16.1	23560	7709
305 x 305 x 95	94.9	303.7	308.7	13.3	13.3	15.2	246.7	11.6	18.5	20040	6529
305 x 305 x 88	88.0	301.7	307.8	12.4	12.3	15.2	246.7	12.5	19.9	18420	5984
305 x 305 x 79	78.9	299.3	306.4	11.0	11.1	15.2	246.7	13.8	22.4	16440	5326
254 x 254 x 85	85.1	254.3	260.4	14.4	14.3	12.7	200.3	9.10	13.9	12280	4215
254 x 254 x 71	71.0	249.7	258.0	12.0	12.0	12.7	200.3	10.8	16.7	10070	3439
254 x 254 x 63	63.0	247.1	256.6	10.6	10.7	12.7	200.3	12.0	18.9	8860	3016
203 x 203 x 54	53.9	204.0	207.7	11.3	11.4	10.2	160.8	9.11	14.2	5027	1705
203 x 203 x 45	44.9	200.2	205.9	9.5	9.5	10.2	160.8	10.8	16.9	4100	1384

IV

Please consult with Corus for availability. See page 42/43.

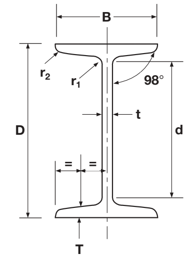


Universal bearing piles

Dimensions and properties to BS 4: Part 1: 1993

Radius of Gyration		Elastic Modulus		Plastic Modulus		Buckling Parameter	Torsional Index	Warping Constant	Torsional Constant	Area of Section	Mass per metre	Designation
Axis x-x cm	Axis y-y cm	Axis x-x cm^3	Axis y-y cm^3	Axis x-x cm^3	Axis y-y cm^3	u	x	H dm^6	J cm^4	cm^2	kg/m	Serial Size
15.2	9.13	2823	976	3186	1497	0.821	15.8	5.37	330	221	173.9	**356 x 368 x 174**
15.1	9.05	2468	845	2767	1293	0.821	17.8	4.55	223	194	152.0	**356 x 368 x 152**
15.0	8.99	2158	732	2406	1118	0.822	20.1	3.87	151	169	133.0	**356 x 368 x 133**
14.9	8.90	1769	592	1956	903	0.823	24.2	3.05	84.6	139	108.9	**356 x 368 x 109**
13.6	7.87	3119	1079	3653	1680	0.826	9.55	4.15	943	284	222.9	**305 x 305 x 223**
13.4	7.73	2596	881	3003	1366	0.827	11.1	3.24	560	237	186.0	**305 x 305 x 186**
13.2	7.58	2076	691	2370	1065	0.828	13.5	2.42	295	190	149.1	**305 x 305 x 149**
13.1	7.49	1755	575	1986	885	0.829	15.7	1.95	182	161	126.1	**305 x 305 x 126**
13.0	7.42	1531	496	1720	762	0.830	17.7	1.65	122	140	110.0	**305 x 305 x 110**
12.9	7.35	1320	423	1475	648	0.830	20.2	1.38	80.0	121	94.9	**305 x 305 x 95**
12.8	7.31	1221	389	1361	595	0.830	21.6	1.25	64.2	112	88.0	**305 x 305 x 88**
12.8	7.28	1099	348	1218	531	0.832	23.9	1.11	46.9	100	78.9	**305 x 305 x 79**
10.6	6.24	966	324	1092	498	0.825	15.6	0.607	81.8	108	85.1	**254 x 254 x 85**
10.6	6.17	807	267	904	409	0.826	18.4	0.486	48.4	90.4	71.0	**254 x 254 x 71**
10.5	6.13	717	235	799	360	0.827	20.5	0.421	34.3	80.2	63.0	**254 x 254 x 63**
8.55	4.98	493	164	557	252	0.827	15.8	0.158	32.7	68.7	53.9	**203 x 203 x 54**
8.46	4.92	410	134	459	206	0.827	18.6	0.126	19.2	57.2	44.9	**203 x 203 x 45**

IV



Joists

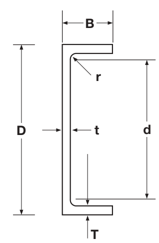
Dimensions and properties to BS 4: Part 1: 1993

Inside slope 8°

Designation Serial Size	Mass per metre kg/m	Depth of Section D mm	Width of Section B mm	Thickness of Web t mm	Thickness of Flange T mm	Radius Root r_1 mm	Radius Toe r_2 mm	Depth between fillets d mm	Ratios for Local Buckling Flange B/2T	Ratios for Local Buckling Web d/t	Second Moment of Area Axis x-x cm^4	Second Moment of Area Axis y-y cm^4
254 x 203 x 82	82	254	203.2	10.2	19.9	19.6	9.7	166.6	5.11	16.3	12000	2280
203 x 152 x 52	52.3	203.2	152.4	8.9	16.5	15.5	7.6	133.2	4.62	15.0	4800	816
152 x 127 x 37	37.3	152.4	127	10.4	13.2	13.5	6.6	94.3	4.81	9.07	1820	378
127 x 114 x 29	29.3	127	114.3	10.2	11.5	9.9	4.8	79.5	4.97	7.79	979	242
127 x 114 x 27	26.9	127	114.3	7.4	11.4	9.9	5.0	79.5	5.01	10.7	946	236
102 x 102 x 23	23	101.6	101.6	9.5	10.3	11.1	3.2	55.2	4.93	5.81	486	154
102 x 44 x 7	7.5	101.6	44.5	4.3	6.1	6.9	3.3	74.6	3.65	17.3	153	7.82
89 x 89 x 19	19.5	88.9	88.9	9.5	9.9	11.1	3.2	44.2	4.49	4.65	307	101
76 x 76 x 13	12.8	76.2	76.2	5.1	8.4	9.4	4.6	38.1	4.54	7.47	158	51.8

IV

All joists subject to viable mount size. Please consult with Corus for availability. See page 42/43.

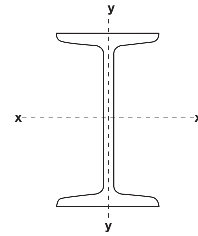


Parallel flange channels

Dimensions and properties to BS 4: Part 1: 1993

Designation Serial Size	Mass per metre kg/m	Depth of Section D mm	Width of Section B mm	Thickness of Web t mm	Thickness of Flange T mm	Distance of Cy cm	Root Radius r mm	Depth between fillets d mm	Ratios for Local Buckling Flange B/T	Ratios for Local Buckling Web d/t	Second Moment of Area Axis x-x cm^4	Second Moment of Area Axis y-y cm^4
430 x 100 x 64	64.40	430	100	11.00	19.00	2.62	15	362.0	5.26	32.9	21940	722
380 x 100 x 54	54.00	380	100	9.50	17.50	2.79	15	315.0	5.71	33.2	15030	643
300 x 100 x 46	45.50	300	100	9.00	16.50	3.05	15	237.0	6.06	26.3	8229	568
300 x 90 x 41	41.40	300	90	9.00	15.50	2.60	12	245.0	5.81	27.2	7218	404
260 x 90 x 35	34.80	260	90	8.00	14.00	2.74	12	208.0	6.43	26.0	4728	353
260 x 75 x 28	27.60	260	75	7.00	12.00	2.10	12	212.0	6.25	30.3	3619	185
230 x 90 x 32	32.20	230	90	7.50	14.00	2.92	12	178.0	6.43	23.7	3518	334
230 x 75 x 26	25.70	230	75	6.50	12.50	2.30	12	181.0	6.00	27.8	2748	181
200 x 90 x 30	29.70	200	90	7.00	14.00	3.12	12	148.0	6.43	21.1	2523	314
200 x 75 x 23	23.40	200	75	6.00	12.50	2.48	12	151.0	6.00	25.2	1963	170
180 x 90 x 26	26.10	180	90	6.50	12.50	3.17	12	131.0	7.20	20.2	1817	277
180 x 75 x 20	20.30	180	75	6.00	10.50	2.41	12	135.0	7.14	22.5	1370	146
150 x 90 x 24	23.90	150	90	6.50	12.00	3.30	12	102.0	7.50	15.7	1162	253
150 x 75 x 18	17.90	150	75	5.50	10.00	2.58	12	106.0	7.50	19.3	861	131
125 x 65 x 15	14.80	125	65	5.50	9.50	2.25	12	82.0	6.84	14.9	483	80.0
100 x 50 x 10	10.20	100	50	5.00	8.50	1.73	9	65.0	5.88	13.0	208	32.3

Please consult with Corus for availability. See page 42/43.

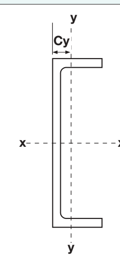


Joists

Dimensions and properties to BS 4: Part 1: 1993

Radius of Gyration Axis x-x cm	Radius of Gyration Axis y-y cm	Elastic Modulus Axis x-x cm³	Elastic Modulus Axis y-y cm³	Plastic Modulus Axis x-x cm³	Plastic Modulus Axis y-y cm³	Buckling Parameter u	Torsional Index x	Warping Constant H dm⁶	Torsional Constant J cm⁴	Area of Section cm²	Mass per metre kg/m	Designation Serial Size
10.7	4.67	947	224	1080	371	0.888	11	0.312	152	105	82	**254 x 203 x 82**
8.49	3.5	472	107	541	176	0.890	10.7	0.0711	64.8	66.6	52.3	**203 x 152 x 52**
6.19	2.82	239	59.6	279	99.8	0.867	9.33	0.0183	33.9	47.5	37.3	**152 x 127 x 37**
5.12	2.54	154	42.3	181	70.8	0.853	8.77	0.00807	20.8	37.4	29.3	**127 x 114 x 29**
5.26	2.63	149	41.3	172	68.2	0.868	9.31	0.00788	16.9	34.2	26.9	**127 x 114 x 27**
4.07	2.29	95.6	30.3	113	50.6	0.836	7.42	0.00321	14.2	29.3	23	**102 x 102 x 23**
4.01	0.907	30.1	3.51	35.4	6.03	0.872	14.9	0.000178	1.25	9.5	7.5	**102 x 44 x 7**
3.51	2.02	69	22.8	82.7	38	0.830	6.58	0.00158	11.5	24.9	19.5	**89 x 89 x 19**
3.12	1.79	41.5	13.6	48.7	22.4	0.853	7.21	0.000595	4.59	16.2	12.8	**76 x 76 x 13**

IV



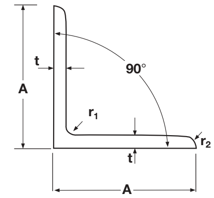
Parallel flange channels

Dimensions and properties to BS 4: Part 1: 1993

Radius of Gyration Axis x-x cm	Radius of Gyration Axis y-y cm	Elastic Modulus Axis x-x cm³	Elastic Modulus Axis y-y cm³	Plastic Modulus Axis x-x cm³	Plastic Modulus Axis y-y cm³	Buckling Parameter u	Torsional Index x	Warping Constant H dm⁶	Torsional Constant J cm⁴	Area of Section cm²	Mass per metre kg/m	Designation Serial Size
16.3	2.97	1020	97.9	1222	176	0.917	22.5	0.219	63.0	82.1	64.4	**430 x 100 x 64**
14.8	3.06	791	89.2	933	161	0.932	21.2	0.150	45.7	68.7	54	**380 x 100 x 54**
11.9	3.13	549	81.7	641	148	0.944	17.0	0.0813	36.8	58.0	45.5	**300 x 100 x 46**
11.7	2.77	481	63.1	568	114	0.934	18.4	0.0581	28.8	52.7	41.4	**300 x 90 x 41**
10.3	2.82	364	56.3	425	102	0.942	17.2	0.0379	20.6	44.4	34.8	**260 x 90 x 35**
10.1	2.30	278	34.4	328	62.0	0.932	20.5	0.0203	11.7	35.1	27.6	**260 x 75 x 28**
9.27	2.86	306	55.0	355	98.9	0.950	15.1	0.0279	19.3	41.0	32.2	**230 x 90 x 32**
9.17	2.35	239	34.8	278	63.2	0.947	17.3	0.0153	11.8	32.7	25.7	**230 x 75 x 26**
8.16	2.88	252	53.4	291	94.5	0.954	12.9	0.0197	18.3	37.9	29.7	**200 x 90 x 30**
8.11	2.39	196	33.8	227	60.6	0.956	14.8	0.0107	11.1	29.9	23.4	**200 x 75 x 23**
7.40	2.89	202	47.4	232	83.5	0.949	12.8	0.0141	13.3	33.2	26.1	**180 x 90 x 26**
7.27	2.38	152	28.8	176	51.8	0.946	15.3	0.0075	7.34	25.9	20.3	**180 x 75 x 20**
6.18	2.89	155	44.4	179	76.9	0.936	10.8	0.0089	11.8	30.4	23.9	**150 x 90 x 24**
6.15	2.40	115	26.6	132	47.2	0.946	13.1	0.0047	6.10	22.8	17.9	**150 x 75 x 18**
5.07	2.06	77.3	18.8	89.9	33.2	0.942	11.1	0.0019	4.72	18.8	14.8	**125 x 65 x 15**
4.00	1.58	41.5	9.89	48.9	17.5	0.942	10.0	0.0005	2.53	13.0	10.2	**100 x 50 x 10**

Equal angles

Dimensions and properties to BS EN 10056-1: 1999



Designation Serial Size A x A x t mm x mm x mm	Mass per metre kg/m	Root Radius r_1 mm	Toe Radius r_2 mm	Area of Section cm^2	Distance of centre of gravity C cm	Second Moment of Area Axis x-x, y-y cm^4	Axis u-u cm^4	Axis v-v cm^4
200 x 200 x 24	71.1	18	9.0	90.6	5.84	3331	5280	1380
200 x 200 x 20	59.9	18	9.0	76.3	5.68	2851	4530	1170
200 x 200 x 18	54.2	18	9.0	69.1	5.60	2600	4150	1050
200 x 200 x 16	48.5	18	9.0	61.8	5.52	2342	3720	960
† **150 x 150 x 18**	40.1	16	8.0	51.0	4.37	1050	1680	440
150 x 150 x 15	33.8	16	8.0	43.0	4.25	898	1430	370
150 x 150 x 12	27.3	16	8.0	34.8	4.12	737	1170	303
150 x 150 x 10	23.0	16	8.0	29.3	4.03	624	990	258
† **120 x 120 x 15**	26.6	13	6.5	33.9	3.51	445	710	186
120 x 120 x 12	21.6	13	6.5	27.5	3.40	368	584	152
120 x 120 x 10	18.2	13	6.5	23.2	3.31	313	497	129
† **120 x 120 x 8**	14.7	13	6.5	18.7	3.23	256	411	107
† **100 x 100 x 15**	21.9	12	6.0	27.9	3.02	249	395	105
100 x 100 x 12	17.8	12	6.0	22.7	2.90	207	328	85.7
100 x 100 x 10	15.0	12	6.0	19.2	2.82	177	280	73
100 x 100 x 8	12.2	12	6.0	15.5	2.74	145	230	59.9
† **90 x 90 x 12**	15.9	11	5.5	20.3	2.66	148	235	62
90 x 90 x 10	13.4	11	5.5	17.1	2.58	127	201	52.6
90 x 90 x 8	10.9	11	5.5	13.9	2.50	104	166	43.1
90 x 90 x 7	9.6	11	5.5	12.2	2.45	92.6	147	38.3

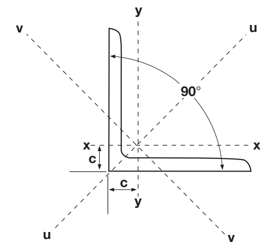
† These sizes are in addition to our standard range to BS EN 10056-1: 1999 specification

Please consult with Corus for availability. See page 42/43.

IV

Equal angles

Dimensions and properties to BS EN 10056-1: 1999

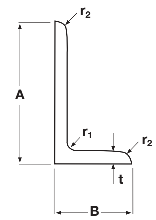


Radius of Gyration Axis x-x, y-y cm	Radius of Gyration Axis u-u cm	Radius of Gyration Axis v-v cm	Elastic Modulus Axis x-x, y-y cm^3	Torsional Constant J cm^4	Equivalent Slenderness Coefficient ϕ_a	Mass per metre kg/m	Designation Serial Size A x A x t mm x mm x mm	
6.06	7.64	3.90	235	182	2.50	71.1	**200 x 200 x 24**	
6.11	7.70	3.92	199	107	3.05	59.9	**200 x 200 x 20**	
6.13	7.75	3.90	181	78.9	3.43	54.2	**200 x 200 x 18**	
6.16	7.76	3.94	162	56.1	3.85	48.5	**200 x 200 x 16**	
4.54	5.73	2.92	98.8	58.6	2.48	40.1	**150 x 150 x 18**	†
4.57	5.76	2.93	83.5	34.6	3.01	33.8	**150 x 150 x 15**	
4.60	5.80	2.95	67.8	18.2	3.77	27.3	**150 x 150 x 12**	
4.62	5.82	2.97	56.9	10.8	4.51	23.0	**150 x 150 x 10**	
3.62	4.57	2.34	52.4	27.0	2.37	26.6	**120 x 120 x 15**	†
3.65	4.60	2.35	42.7	14.2	2.99	21.6	**120 x 120 x 12**	
3.67	4.63	2.36	36.0	8.41	3.61	18.2	**120 x 120 x 10**	
3.69	4.67	2.38	29.1	4.44	4.56	14.7	**120 x 120 x 8**	†
2.98	3.76	1.94	35.6	22.3	1.92	21.9	**100 x 100 x 15**	†
3.02	3.80	1.94	29.1	11.8	2.44	17.8	**100 x 100 x 12**	
3.04	3.83	1.95	24.6	6.97	2.94	15.0	**100 x 100 x 10**	
3.06	3.85	1.96	20.0	3.68	3.70	12.2	**100 x 100 x 8**	
2.70	3.40	1.75	23.4	10.46	2.17	15.9	**90 x 90 x 12**	†
2.72	3.42	1.75	19.8	6.20	2.64	13.4	**90 x 90 x 10**	
2.74	3.45	1.76	16.1	3.28	3.33	10.9	**90 x 90 x 8**	
2.75	3.46	1.77	14.1	2.24	3.80	9.6	**90 x 90 x 7**	

IV

Unequal angles

Dimensions and properties to BS EN 10056-1: 1999



Designation Serial Size A x B x t mm x mm x mm	Mass per metre kg/m	Root Radius r_1 mm	Toe Radius r_2 mm	Area of Section cm^2	Distance of centre of gravity C_x cm	C_y cm	Second Moment of Area Axis x-x cm^4	Axis y-y cm^4	Axis u-u cm^4	Axis v-v cm^4
† **200 x 150 x 18**	47.1	15	7.5	60.0	6.33	3.85	2376	1146	2920	623
200 x 150 x 15	39.6	15	7.5	50.5	6.21	3.73	2023	979	2480	526
200 x 150 x 12	32.0	15	7.5	40.8	6.08	3.61	1653	803	2030	430
200 x 100 x 15	33.7	15	7.5	43.0	7.16	2.22	1759	299	1860	193
200 x 100 x 12	27.3	15	7.5	34.8	7.03	2.10	1441	247	1530	159
200 x 100 x 10	23.0	15	7.5	29.2	6.93	2.01	1219	210	1290	135
150 x 90 x 15	26.6	12	6.0	33.9	5.21	2.23	761	205	841	126
150 x 90 x 12	21.6	12	6.0	27.5	5.08	2.12	627	171	694	104
150 x 90 x 10	18.2	12	6.0	23.2	5.00	2.04	533	146	591	88.3
150 x 75 x 15	24.8	12	6.0	31.7	5.52	1.81	713	119	753	78.6
150 x 75 x 12	20.2	12	6.0	25.7	5.40	1.69	589	99.6	623	64.7
150 x 75 x 10	17.0	12	6.0	21.7	5.31	1.61	501	85.4	531	55.1
125 x 75 x 12	17.8	11	5.5	22.7	4.31	1.84	354	95.5	391	58.5
125 x 75 x 10	15.0	11	5.5	19.1	4.23	1.76	302	82.1	334	49.9
125 x 75 x 8	12.2	11	5.5	15.5	4.14	1.68	247	67.6	274	40.9
100 x 75 x 12	15.4	10	5.0	19.7	3.27	2.03	189	90.2	230	49.5
100 x 75 x 10	13.0	10	5.0	16.6	3.19	1.95	162	77.6	197	42.2
100 x 75 x 8	10.6	10	5.0	13.5	3.10	1.87	133	64.1	162	34.6
† **100 x 65 x 10**	12.3	10	5.0	15.6	3.36	1.63	154	51.0	175	30.1
† **100 x 65 x 8**	9.9	10	5.0	12.7	3.27	1.55	127	42.2	144	24.8
† **100 x 65 x 7**	8.8	10	5.0	11.2	3.23	1.51	113	37.6	128	22

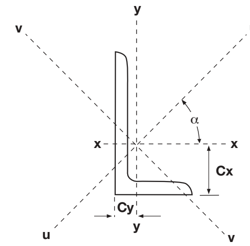
† These sizes are in addition to our standard range to BS EN 10056-1: 1999 specification

Please consult with Corus for availability. See page 42/43.

IV

Unequal angles

Dimensions and properties to BS EN 10056-1: 1999



Radius of Gyration				Elastic Modulus		Angle Axis x-x to Axis y-y	Torsinal Constant	Equivalent Slenderness Coefficient		Mono-Symmetry Index	Mass per metre	Designation Serial Size	
Axis x-x cm	Axis y-y cm	Axis u-u cm	Axis v-v cm	Axis x-x cm^3	Axis y-y cm^3	Tan α	J cm^4	Min. Φ_a	Max. Φ_a	Ψ_a	kg/m	A x B x t mm x mm x mm	
6.29	4.37	6.97	3.22	174	103	0.549	67.9	2.93	3.72	4.60	47.1	**200 x 150 x 18**	†
6.33	4.40	7.00	3.23	147	86.9	0.551	39.9	3.53	4.50	5.55	39.6	**200 x 150 x 15**	
6.36	4.44	7.04	3.25	119	70.5	0.552	20.9	4.43	5.70	6.97	32.0	**200 x 150 x 12**	
6.40	2.64	6.59	2.12	137	38.4	0.260	34.3	3.54	5.17	9.19	33.7	**200 x 100 x 15**	
6.43	2.67	6.63	2.14	111	31.3	0.262	18.0	4.42	6.57	11.5	27.3	**200 x 100 x 12**	
6.46	2.68	6.65	2.15	93.3	26.3	0.263	10.66	5.26	7.92	13.9	23.0	**200 x 100 x 10**	
4.74	2.46	4.98	1.93	77.7	30.4	0.354	26.8	2.58	3.59	5.96	26.6	**150 x 90 x 15**	
4.78	2.49	5.02	1.94	63.3	24.8	0.358	14.1	3.24	4.58	7.50	21.6	**150 x 90 x 12**	
4.80	2.51	5.05	1.95	53.3	21.0	0.360	8.30	3.89	5.56	9.03	18.2	**150 x 90 x 10**	
4.75	1.94	4.88	1.58	75.2	21.0	0.253	25.1	2.62	3.74	6.84	24.8	**150 x 75 x 15**	
4.78	1.97	4.92	1.59	61.3	17.1	0.258	13.2	3.30	4.79	8.60	20.2	**150 x 75 x 12**	
4.81	1.99	4.95	1.60	51.7	14.5	0.261	7.8	3.95	5.83	10.4	17.0	**150 x 75 x 10**	
3.95	2.05	4.15	1.61	43.2	16.9	0.354	11.6	2.66	3.73	6.23	17.8	**125 x 75 x 12**	
3.97	2.07	4.18	1.61	36.5	14.3	0.357	6.87	3.21	4.55	7.50	15.0	**125 x 75 x 10**	
4.00	2.09	4.21	1.63	29.6	11.6	0.360	3.62	4.00	5.75	9.43	12.2	**125 x 75 x 8**	
3.10	2.14	3.42	1.59	28.1	16.5	0.540	10.05	2.10	2.64	3.46	15.4	**100 x 75 x 12**	
3.12	2.16	3.45	1.59	23.8	14.0	0.544	5.95	2.54	3.22	4.17	13.0	**100 x 75 x 10**	
3.14	2.18	3.47	1.60	19.3	11.4	0.547	3.13	3.18	4.08	5.24	10.6	**100 x 75 x 8**	
3.14	1.81	3.35	1.39	23.2	10.5	0.410	5.61	2.52	3.43	5.45	12.3	**100 x 65 x 10**	†
3.16	1.83	3.37	1.40	18.9	8.54	0.413	2.96	3.14	4.35	6.86	9.9	**100 x 65 x 8**	†
3.17	1.83	3.39	1.40	16.6	7.53	0.415	2.02	3.58	5.00	7.85	8.8	**100 x 65 x 7**	†

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