STANDARD METAL BALLS

Standard Metal Balls.—American National Standard ANSI/AFBMA Std 10-1989 provides information for the user of metal balls permitting them to be described readily and accurately. It also covers certain measurable characteristics affecting ball quality.

On the following pages, tables taken from this Standard cover standard balls for bearings and other purposes by type of material, grade, and size range; preferred ball sizes; ball hardness corrections for curvature; various tolerances, marking increments, and maximum surface roughnesses by grades; total hardness ranges for various materials; and minimum case depths for carbon steel balls. The numbers of balls per pound and per kilogram for ferrous and nonferrous metals are also shown.

Definitions and Symbols.—The following definitions and symbols apply to American National Standard metal balls.

Nominal Ball Diameter, D_w: The diameter value that is used for the general identification of a ball size, e.g., $\frac{1}{4}$ inch, 6 mm, etc.

Single Diameter of a Ball, D_{ws}: The distance between two parallel planes tangent to the surface of a ball.

Mean Diameter of a Ball, D_{wm}: The arithmetical mean of the largest and smallest single diameters of a ball.

Ball Diameter Variation, V_{Dws}: The difference between the largest and smallest single diameters of one ball.

Deviation from Spherical Form, ΔR_w: The greatest radial distance in any radial plane between a sphere circumscribed around the ball surface and any point on the ball surface.

Lot: A definite quantity of balls manufactured under conditions that are presumed uniform, considered and identified as an entirety.

Lot Mean Diameter, D_{wmL}: The arithmetical mean of the mean diameter of the largest ball and that of the smallest ball in the lot.

Lot Diameter Variation, V_{DwL}: The difference between the mean diameter of the largest ball and that of the smallest ball in the lot.

Nominal Ball Diameter Tolerance: The maximum allowable deviation of any ball lot mean diameter from the Nominal Ball Diameter.

Container Marking Increment: The Standard unit steps in millionths of an inch or in micrometers used to express the Specific Diameter.

Specific Diameter: The amount by which the lot mean diameter (D_{wmL}) differs from the nominal diameter (D_w), accurate to the container marking increment for that grade; the specific diameter should be marked on the unit container.

Ball Gage Deviation, ΔS: The difference between the lot mean diameter and the sum of the nominal mean diameter and the ball gage.

Surface Roughness, R_a: Surface roughness consists of all those irregularities that form surface relief and are conventionally defined within the area where deviations of form and waviness are eliminated. (See Handbook Surface Texture Section.)

Ordering Specifications.—Unless otherwise agreed between producer and user, orders for metal balls should provide the following information: quantity, material, nominal ball diameter, grade, and ball gage. A *ball grade* embodies a specific combination of dimensional form, and surface roughness tolerances. A *ball gage(s)* is the prescribed small amount, expressed with the proper algebraic sign, by which the lot mean diameter (arithmetic mean of the mean diameters of the largest and smallest balls in the lot) should differ from the nominal diameter, this amount being one of an established series of amounts as shown in the table below. The 0 ball gage is commonly referred to as “OK”.

Preferred Ball Gages for Grades 3 to 200

Grade	Ball Gages (in 0.0001-in. units) Minus	OK	Plus	Ball Gages (in 1µm units) Minus	OK	Plus
3, 5	− 3 − 2 − 1	0	+ 1 + 2 + 3	− 8 − 7 − 6 − 5 − 4 − 3 − 2 − 1	0	+ 1 + 2 + 3 + 4 + 5 + 6 + 7 + 8
10, 16	− 4 − 3 − 2 − 1	0	+ 1 + 2 + 3 + 4	− 10 − 8 − 6 − 4 − 2	0	+ 2 + 4 + 6 + 8 + 10
24	− 5 − 4 − 3 − 2 − 1	0	+ 1 + 2 + 3 + 4 + 5	− 12 − 10 − 8 − 6 − 4 − 2	0	+ 2 + 4 + 6 + 8 + 10 + 12
48	− 6 − 4 − 2	0	+ 2 + 4 + 6	− 16 − 12 − 8 − 4	0	+ 4 + 8 + 12 + 16
100		0			0	
200		0			0	

Table 1. AFBMA Standard Balls — Tolerances for Individual Balls and for Lots of Balls

Grade	Allowable Ball Diameter Variation	Allowable Deviation from Spherical Form	Maximum Surface Roughness R_a	Allowable Lot Diameter Variation	Nominal Ball Diameter Tolerance (±)	Container Marking Increments
	For Individual Balls			For Lots of Balls		
	Millionths of an Inch					
3	3	3	0.5	5	[a]	10
5	5	5	0.8	10	[a]	10
10	10	10	1	20	[a]	10
16	16	16	1	32	[a]	10
24	24	24	2	48	[a]	10
48	48	48	3	96	[a]	50
100	100	100	5	200	500	[a]
200	200	200	8	400	1000	[a]
500	500	500	[a]	1000	2000	[a]
1000	1000	1000	[a]	2000	5000	[a]
	Micrometers					
3	0.08	0.08	0.012	0.13	[a]	0.25
5	0.13	0.13	0.02	0.25	[a]	0.25
10	0.25	0.25	0.025	0.5	[a]	0.25
16	0.4	0.4	0.025	0.8	[a]	0.25
24	0.6	0.6	0.05	1.2	[a]	0.25
48	1.2	1.2	0.08	2.4	[a]	1.25
100	2.5	2.5	0.125	5	12.5	[a]
200	5	5	0.2	10	25	[a]
500	13	13	[a]	25	50	[a]
1000	25	25	[a]	50	125	[a]

[a] Not applicable.

Allowable ball gage (see text) deviation is for Grade 3: + 0.000030, – 0.000030 inch (+0.75, – 0.75 µm); for Grades 5, 10, and 16: + 0.000050, – 0.000040 inch (+ 1.25, – 1 µm); and for Grade 24: + 0.000100, – 0.000100 inch (+ 2.5, – 2.5 µm). Other grades not given.

Examples: A typical order, in inch units, might read as follows: 80,000 pieces, chrome alloy steel, 1⁄4-inch Nominal Diameter, Grade 16, and Ball Gage to be –0.0002 inch.

A typical order, in metric units, might read as follows: 80,000 pieces, chrome alloy steel, 6 mm Nominal Diameter, Grade 16, and Ball Gage to be –4 µm.

Package Marking: The ball manufacturer or supplier will identify packages containing each lot with information provided on the orders, as given above. In addition, the specific diameter of the contents shall be stated. Container marking increments are listed in Table 1.

Examples: Balls supplied to the order of the first of the previous examples would, if perfect size, be D_{wmL} = 0.249800 inch. In Grade 16 these balls would be acceptable with D_{wmL} from 0.249760 to 0.249850 inch. If they actually measured 0.249823 (which would be rounded off to 0.249820), each package would be marked: 5,000 Balls, Chrome Alloy Steel, 1/4″ Nominal Diameter, Grade 16, –0.0002 inch Ball Gage, and –0.000180 inch Specific Diameter.

Balls supplied to the order of the second of the two previous examples would, if perfect size, be D_{wmL} = 5.99600 mm. In Grade 16 these balls would be acceptable with a D_{wmL} from 5.99500 to 5.99725 mm. If they actually measured 5.99627 mm (which would be rounded off to 5.99625 mm), each package would be marked: 5,000 Balls, Chrome Alloy Steel, 6 mm Nominal Diameter, Grade 16, –4 μm Ball Gage, and –3.75 μm Specific Diameter.

Table 2. AFBMA Standard Balls — Typical Nominal Size Ranges by Material and Grade

Steel Balls[a]			
		Size Range[b]	
Material	Grade	Inch	mm
Chrome Alloy	3	1/32–1	0.8–25
	5,10, 16,24	1/64–1 1/2	0.3–38
	48, 100, 200, 500	1/32–2 7/8	0.8–75
	1000	3/8–4 1/2	10–115
AISI M-50	3 5,10,16	1/32–1/2	0.8–12
	24,48	1/32–1 5/8	0.8–40
Corrosion Resisting Hardened	3,5,10,16	1/64–3/4	0.3–19
	24	1/32–1	0.8–25
	48	1/32–2	0.8–50
	100,200	1/32–4 1/2	0.8–115
Corrosion-Resisting Unhardened	100,200, 500	1/16–3/4	1.5–19
Carbon Steel[c]	100,200, 500, 1000	1/16–1 1/2	1.5–38
Silicon Molybdenum	200	1/4–1 1/8	6.5–28

Non-Ferrous Balls[a]			
		Size Range[b]	
Material Grade	Grade	Inch	mm
Aluminum	200	1/16–1	1.5–25
Aluminum Bronze	200	13/16–4	20–100
Brass	100,200, 500, 1000	1/16–3/4	1.5–19
Bronze	200,500, 1000	1/16–3/4	1.5–19
Monel Metal 400	100,200, 500	1/16–3/4	1.5–19
K-Monel Metal 500	100	1/16–3/4	1.5–19
	200	1/16–1 11/16	1.5–45
Tungsten Carbide	5	3/64–1/2	1.2–12
	10	3/64–3/4	1.2–19
	16	3/64–1	1.2–25
	24	3/64–1 1/4	1.2–32

[a] For hardness rages see Table 3.
[b] For tolerances see Table 1.
[c] For minimum case depths refer to the Standard.

Table 3. AFBMA Standard Balls—Typical Hardness Ranges

Material	Common Standard	SAE Unified Number	Rockwell Value[a,b]
Steel—			
Alloy tool	AISI/SAE M50	T-11350	60–65 "C"[c,d]
Carbon[e]	AISI/SAE 1008	G-10080	60 Minimum "C"[b]
	AISI/SAE 1013	G-10130	60 Minimum "C"[b]
	AISI/SAE 1018	G-10180	60 Minimum "C"[b]
AISI/SAE 1022	G-10220	60 Minimum "C"[b]	
Chrome alloy	AISI/SAE E52100	G-5298660	60–67 "C"[c,d]
	AISI/SAE E51100	G-51986	60-67 "C"[c,d]
Corrosion-resisting			
	AISI/SAE 440C	S-44004	58–65 "C"[f,d]
hardened	AISI/SAE 440B	S-44003	55–62 "C"[f,d]
	AISI/SAE 420	S-42000	52 Minimum "C"[f,d]
	AISI/SAE 410	S-41000	97 "B"; 41 "C"[f,d]
	AISI/SAE 329	S-32900	45 Minimum "C"[f,d]
Corrosion-resisting			
unhardened	AISI/SAE 3025	S-30200	25–39 "C"[d,g]
	AISI/SAE 304	S-304000	25–39 "C"[d,g]
	AISI/SAE 305	S-305000	25–39 "C"[d,g]
	AISI/SAE 316	S-31600	25–39 "C"[d,g]
	AISI/SAE 430	S-43000	48–63 "A"[d]
Silicon molybdenum	AISI/SAE S2	T-41902	52–60 "C"[c]
Aluminum	AA-2017	A-92017	54–72 "B"
Aluminium bronze	CDA-624	C-62400	94–98 "B"
	CDA-630	C-63000	94–98 "B"
Brass	CDA-260	C-26000	75–87 "B"
Bronze	CDA-464	C-46400	75–98 "B"
Monel 400	AMS-4730	N-04400	85–95 "B"
Monel K-500	QA-N-286	N-05500	24 Minimum "C"
Tungsten carbide	JIC Carbide Classification	…	84-91.5 "A"

[a] Rockwell Hardness Tests shall be conducted on parallel flats in accordance with ASTM Standard E-18 unless otherwise specified.

[b] Hardness readings taken on spherical surfaces are subject to the corrections shown in Table 5. Hardness readings for carbon steel balls smaller than 5 mm ($\frac{1}{4}$ inch) shall be taken by the microhardness method (detailed in ANSI/AFBMA Std 10-1989) or as agreed between manufacturer and purchaser.

[c] Hardness of balls in any one lot shall be within 3 points on Rockwell C scale.

[d] When microhardness method (see ANSI/AFBMA Std 10-1989 is used, the Rockwell hardness values given are converted to DPH in accordance with ASTM Standard E 140, "Standard Hardness Conversion Tables for Metals."

[e] Choice of carbon steels shown to be at ball manufacturer's option.

[f] Hardness of balls in any one lot shall be within 4 points on Rockwell C scale.

[g] Annealed hardness of 75-90 "B" is available when specified.

For complete details as to material requirements, quality specifications, quality assurance provisions, and methods of hardness testing, reference should be made to the Standard.

Table 4. Preferred Ball Sizes

Nominal Ball Sizes Metric	Diameter mm	Diameter Inches	Nominal Ball Sizes Inch	Nominal Ball Sizes Metric	Diameter mm[b]	Diameter Inches	Nominal Ball Sizes Inch
0.3	0.300 00	0.011 810			0.793 75	0.031 250	$\frac{1}{32}$
	0.396 88	0.015 625	$\frac{1}{64}$	0.8	0.800 00	0.031 496	
0.4	0.400 00	0.015 750		1	1.000 00	0.039 370	
0.5	0.500 00	0.019 680			1.190 63	0.046 875	$\frac{3}{64}$
	0.508 00	0.020 000	0.020	1.2	1.200 00	0.047 240	
0.6	0.600 00	0.023 620		1.5	1.500 00	0.059 060	
	0.635 00	0.025 000	0.025		1.587 50	0.062 500	$\frac{1}{16}$
0.7	0.700 00	0.027 560			1.984 38	0.078 125	$\frac{5}{64}$
2	2.000 00	0.078 740		21	21.000 000	0.826 770	
	2.381 25	0.093 750	$\frac{3}{32}$		21.431 25	0.843 750	$\frac{27}{32}$
2.5	2.500 00	0.098 420		22	22.000 00	0.866 140	
	2.778 00	0.109 375	$\frac{7}{64}$		22.225 00	0.875 000	$\frac{7}{8}$
3	3.000 00	0.118 110		23	23.000 00	0.905 510	
	3.175 00	0.125 000	$\frac{1}{8}$		23.018 75	0.906 250	$\frac{29}{32}$
3.5	3.500 00	0.137 800			23.812 50	0.937 500	$\frac{15}{16}$
	3.571 87	0.140 625	$\frac{9}{64}$	24	24.000 00	0.944 880	
	3.968 75	0.156 250	$\frac{5}{32}$		24.606 25	0.968 750	$\frac{31}{32}$
4	4.000 00	0.157 480		25	25.000 00	0.984 250	
	4.365 63	0.171 875	$\frac{11}{64}$		25.400 00	1.000 000	1
4.5	4.500 00	0.177 160		26	26.000 00	1.023 620	
	4.762 50	0.187 500	$\frac{3}{16}$		26.987 50	1.062 500	$1\frac{1}{16}$
5	5.000 00	0.196 850		28	28.000 00	1.102 360	
5.5	5.500 00	0.216 540			28.575 00	1.125 000	$1\frac{1}{8}$
	5.556 25	0.218 750	$\frac{7}{32}$	30	30.000 00	1.181 100	
	5.953 12	0.234 375	$\frac{15}{64}$		30.162 50	1.187 500	$1\frac{3}{16}$
6	6.000 00	0.236 220			31.750 00	1.250 000	$1\frac{1}{4}$
	6.350 00	0.250 000	$\frac{1}{4}$	32	32.000 00	1.259 840	
6.5	6.500 00	0.255 900			33.337 50	1.312 500	$1\frac{5}{16}$
	6.746 88	0.265 625	$\frac{17}{64}$	34	34.000 00	1.338 580	
7	7.000 00	0.275 590			34.925 00	1.375 000	$1\frac{3}{8}$
	7.143 75	0.281 250	$\frac{9}{32}$	35	35.000 00	1.377 950	
7.5	7.500 00	0.295 280		36	36.000 00	1.417 320	
	7.540 63	0.296 875	$\frac{19}{64}$		36.512 50	1.437 500	$1\frac{7}{16}$
	7.937 50	0.312 500	$\frac{5}{16}$	38	38.000 00	1.496 060	
8	8.000 00	0.314 960			38.100 00	1.500 000	$1\frac{1}{2}$
8.5	8.500 00	0.334 640			39.687 50	1.562 500	$1\frac{9}{16}$
	8.731 25	0.343 750	$\frac{11}{32}$	40	40.000 00	1.574 800	
9	9.000 00	0.354 330			41.275 00	1.625 000	$1\frac{5}{8}$
	9.128 12	0.359 375	$\frac{23}{64}$		42.862 50	1.687 500	$1\frac{11}{16}$
	9.525 00	0.375 000	$\frac{3}{8}$		44.450 00	1.750 000	$1\frac{3}{4}$
	9.921 87	0.390 625	$\frac{25}{64}$	45	45.000 00	1.771 650	
10	10.000 00	0.393 700			46.037 50	1.812 500	$1\frac{13}{16}$
	10.318 75	0.406 250	$\frac{13}{32}$		47.625 00	1.875 000	$1\frac{7}{8}$
11	11.000 00	0.433 070			49.212 50	1.937 500	$1\frac{15}{16}$
	11.112 50	0.437 500	$\frac{7}{16}$	50	50.000 00	1.968 500	
11.5	11.500 00	0.452 756			50.800 00	2.000 000	2
	11.509 38	0.453 125	$\frac{29}{64}$		53.975 00	2.125 000	$2\frac{1}{8}$

Table 4. *(Continued)* **Preferred Ball Sizes**

Nominal Ball Sizes Metric	Diameter mm	Diameter Inches	Nominal Ball Sizes Inch	Nominal Ball Sizes Metric	Diameter mmb	Diameter Inches	Nominal Ball Sizes Inch
	11.906 25	0.468 750	15⁄32	55	55.000 00	2.165 354	
12	12.000 00	0.472 440			57.150 00	2.250 000	2 1⁄4
	12.303 12	0.484 375	31⁄64	60	60.000 00	2.362 205	
	12.700 00	0.500 000	1⁄2		60.325 00	2.375 00	2 3⁄8
13	13.000 00	0.511 810			63.500 00	2.500 000	2 1⁄2
	13.493 75	0.531 250	17⁄32	65	65.000 00	2.559 055	
14	14.000 00	0.551 180			66.675 00	2.625 000	2 5⁄8
	14.287 50	0.562 500	9⁄16		69.850 00	2.750 000	2 3⁄4
15	15.000 00	0.590 550			73.025 00	2.875 000	2 7⁄8
	15.081 25	0.593 750	19⁄32		76.200 00	3.000 000	3
	15.875 00	0.625 000	5⁄8		79.375 00	3.125 000	3 1⁄8
16	16.000 00	0.629 920			82.550 00	3.250 000	3 1⁄4
	16.668 75	0.656 250	21⁄32		85.725 00	3.375 00	3 3⁄8
17	17.000 00	0.669 290			88.900 00	3.500 000	3 1⁄2
	17.462 50	0.687 500	11⁄16		92.075 00	3.625 000	3 5⁄8
18	18.000 00	0.708 660			95.250 00	3.750 000	3 3⁄4
	18.256 25	0.718 750	23⁄32		98.425 00	3.875 000	3 7⁄8
19	19.000 00	0.748 030			101.600 00	4.000 000	4
	19.050 00	0.750 000	3⁄4		104.775 00	4.125 000	4 1⁄8
	19.843 75	0.781 250	25⁄32		107.950 00	4.250 000	4 1⁄4
20	20.000 00	0.787 400			111.125 00	4.375 000	4 3⁄8
	20.637 50	0.812 500	13⁄16		114.300 00	4.500 000	4 1⁄2

Table 5. Ball Hardness Corrections for Curvatures

Hardness Reading, Rockwell C	Ball Diameters, Inch						
	1⁄4	5⁄16	3⁄8	1⁄2	5⁄8	3⁄4	1
	Correction—Rockwell C						
20	12.1	9.3	7.7	6.1	4.9	4.1	3.1
25	11.0	8.4	7.0	5.5	4.4	3.7	2.7
30	9.8	7.5	6.2	4.9	3.9	3.2	2.4
35	8.6	6.6	5.5	4.3	3.4	2.8	2.1
40	7.5	5.7	4.7	3.6	2.9	2.4	1.7
45	6.3	4.9	4.0	3.0	2.4	1.9	1.4
50	5.2	4.0	3.2	2.4	1.9	1.5	1.1
55	4.1	3.1	2.5	1.8	1.4	1.1	0.8
60	2.9	2.2	1.8	1.2	0.9	0.7	0.4
65	1.8	1.3	1.0	0.5	0.3	0.2	0.1
20	12.8	9.3	7.6	6.6	5.2	4.0	3.2
25	11.7	8.4	6.9	5.9	4.6	3.5	2.8
30	10.5	7.5	6.1	5.2	4.1	3.1	2.4
35	9.4	6.6	5.4	4.6	3.6	2.7	2.1
40	8.0	5.7	4.5	3.8	3.0	2.2	1.8
45	6.7	4.9	3.8	3.2	2.5	1.8	1.4
50	5.5	4.0	3.0	2.6	2.0	1.4	1.1
55	4.3	3.1	2.3	1.9	1.5	1.0	0.8
60	3.0	2.2	1.7	1.2	1.0	0.6	0.4
65	1.9	1.3	0.9	0.6	0.4	0.2	0.1

Corrections to be added to Rockwell C readings obtained on spherical surfaces of chrome alloy steel, corrosion resisting hardened and unhardened steel, and carbon steel balls. For other ball sizes and hardness readings, interpolate between correction values shown.

Table 6. Number of Metal Balls per Pound

Nom. Dia.,[a] Inches	Material Density, Pounds per Cubic Inch												
	.101	.274	.277	.279	.283	.284	.286	.288	.301	.304	.306	.319	.540
1/32	620 000	228 000	226 000	224 000	221 000	220 000	219 000	217 000	208 000	206 000	205 000	196 000	116 000
1/16	77 500	28 600	28 200	28 000	27 600	27 500	27 400	27 200	26 000	25 700	25 600	24 500	14 500
3/32	22 900	8 460	8 370	8 310	8 190	8 160	8 100	8 050	7 700	7 620	7 570	7 270	4 290
1/8	9 680	3 570	3 530	3 500	3 460	3 440	3 420	3 400	3 250	3 220	3 200	3 070	1 810
5/32	4 960	1 830	1 810	1 790	1 770	1 760	1 750	1 740	1 660	1 650	1 640	1 570	927
3/16	2 870	1 060	1 050	1 040	1 020	1 020	1 010	1 010	963	953	947	908	537
7/32	1 810	666	659	654	645	642	638	634	606	600	596	572	338
1/4	1 210	446	441	438	432	430	427	424	406	402	399	383	226
9/32	850	313	310	308	303	302	300	298	285	282	281	269	159
5/16	620	228	226	224	221	220	219	217	208	206	205	196	116
11/32	466	172.	170.	169.	166.	166.	164.	163.	156.	155.	154.	147.	87.1
3/8	359	132.	131.	130.	128.	128.	127.	126.	120.	119.	118.	114.	67.1
13/32	282	104.	103.	102.	101.	100.	99.6	98.9	94.6	93.7	93.1	89.3	52.8
7/16	226	83.2	82.3	81.7	80.6	80.3	79.7	79.2	75.8	75.0	74.5	71.5	42.2
15/32	184	67.7	66.9	66.5	65.5	65.3	64.8	64.4	61.6	61.0	60.6	58.1	34.3
1/2	151.	55.8	55.2	54.8	54.0	53.8	53.4	53.1	50.8	50.3	49.9	47.9	28.3
17/32	126.	46.5	46.0	45.7	45.0	44.9	44.5	44.2	42.3	41.9	41.6	39.9	23.6
9/16	106.	39.2	38.7	38.5	37.9	37.8	37.5	37.3	35.7	35.3	35.1	33.6	19.9
19/32	90.3	33.3	32.9	32.7	32.2	32.1	31.9	31.7	30.3	30.0	29.8	28.6	16.9
5/8	77.5	28.6	28.2	28.0	27.6	27.5	27.4	27.2	26.0	25.7	25.6	24.5	14.5
21/32	66.9	24.7	24.4	24.2	23.9	23.8	23.6	23.5	22.5	22.2	22.1	21.2	12.5
11/16	58.2	21.5	21.2	21.1	20.8	20.7	20.6	20.4	19.5	19.3	19.2	18.4	10.9
23/32	50.9	18.8	18.6	18.4	18.2	18.1	18.0	17.9	17.1	16.9	16.8	16.1	9.53
3/4	44.8	16.5	16.3	16.2	16.0	15.9	15.8	15.7	15.0	14.9	14.8	14.2	8.38
25/32	39.7	14.6	14.5	14.4	14.2	14.1	14.0	13.9	13.3	13.2	13.1	12.6	7.42
13/16	35.3	13.0	12.9	12.8	12.6	12.5	12.5	12.4	11.8	11.7	11.6	11.2	6.59
27/32	31.5	11.6	11.5	11.4	11.2	11.2	11.1	11.0	10.6	10.5	10.4	9.97	5.89
7/8	28.2	10.4	10.3	10.2	10.1	10.0	9.97	9.90	9.47	9.38	9.32	8.94	5.28
29/32	25.4	9.37	9.26	9.20	9.07	9.04	8.97	8.91	8.53	8.44	8.39	8.04	4.75
15/16	22.9	8.46	8.37	8.31	8.19	8.16	8.10	8.05	7.70	7.62	7.57	7.27	4.29
31/32	20.8	7.67	7.58	7.53	7.42	7.40	7.35	7.29	6.98	6.91	6.87	6.59	3.89
1	18.9	6.97	6.89	6.85	6.75	6.72	6.68	6.63	6.35	6.28	6.24	5.99	3.54

[a] For sizes above 1 in. diameter, use the following formula: No. balls per pound = 1.91 ÷ [(nom. dia., in.)^3 × (material density, lbs. per cubic in.)].

Ball material densities in pounds per cubic inch: aluminum .101; aluminum bronze .274; corrosion resisting hardened steel .277; AISI M-50 and silicon molybdenum steels .279; chrome alloy steel .283; carbon steel .284; AISI 302 corrosion resisting unhardened steel .286; AISI 316 corrosion resisting unhardened steel .288; bronze .304; brass and K-Monel metal .306; Monel metal .319; and tungsten carbide .540.

Table 7. Number of Metal Balls per Kilogram

Nom.Dia.,[a] mm	Material Density, Grams per Cubic Centimeter											
	2.796	7.584	7.667	7.723	7.833	7.861	7.916	7.972	8.332	8.415	8.470	8.830	14.947
0.3	25 300 000	9 330 000	9 230 000	9 160 000	9 030 000	9 000 000	8 940 000	8 870 000	8 490 000	8 410 000	8 350 000	8 010 000	4 730 000
0.4	10 670 000	3 930 000	3 890 000	3 860 000	3 810 000	3 800 000	3 770 000	3 740 000	3 580 000	3 550 000	3 520 000	3 380 000	2 000 000
0.5	5 470 000	2 010 000	1 990 000	1 980 000	1 950 000	1 940 000	1 930 000	1 920 000	1 830 000	1 820 000	1 800 000	1 730 000	1 020 000
0.7	1 990 000	734 000	726 000	721 000	711 000	708 000	703 000	698 000	668 000	662 000	657 000	631 000	373 000
0.8	1 330 000	492 000	487 000	483 000	476 000	475 000	471 000	468 000	448 000	443 000	440 000	422 000	250 000
1.0	683 000	252 000	249 000	247 000	244 000	243 000	241 000	240 000	229 000	227 000	225 000	216 000	128 000
1.2	395 000	146 000	144 000	143 000	141 000	141 000	140 000	139 000	133 000	131 000	130 000	125 000	73 900
1.5	202 000	74 600	73 800	73 300	72 200	72 000	71 500	71 000	67 900	67 200	66 800	64 100	37 900
2.0	85 400	31 500	31 100	30 900	30 500	30 400	30 200	29 900	28 700	28 400	28 200	27 000	16 000
2.5	43 700	16 100	15 900	15 800	15 600	15 500	15 400	15 300	14 700	14 500	14 400	13 800	8 180
3.0	25 300	9 330	9 230	9 160	9 030	9 000	8 940	8 870	8 490	8 410	8 350	8 010	4 730
3.5	15 900	5 870	5 810	5 770	5 690	5 670	5 630	5 590	5 350	5 290	5 260	5 040	2 980
4.0	10 700	3 930	3 890	3 860	3 810	3 800	3 770	3 740	3 580	3 550	3 520	3 380	2 000
4.5	7 500	2 760	2 730	2 710	2 680	2 670	2 650	2 630	2 520	2 490	2 470	2 370	1 400
5.0	5 470	5 010	1 990	1 980	1 950	1 940	1 930	1 920	1 830	1 820	1 800	1 730	1 020
5.5	4 110	1 510	1 500	1 490	1 470	1 460	1 450	1 440	1 380	1 360	1 360	1 300	768
6.0	3 160	1 170	1 150	1 140	1 130	1 120	1 120	1 110	1 060	1 050	1 040	1 000	592
6.5	2 490	917	907	901	888	885	878	872	835	826	821	788	465
7.0	1 990	734	726	721	711	708	703	698	668	662	657	631	373
7.5	1 620	597	590	586	578	576	572	568	543	538	534	513	303
8.0	1 330	492	487	483	476	475	471	468	448	443	440	422	250
8.5	1 110	410	406	403	397	396	393	390	373	370	367	352	208
9.0	937	345	342	339	334	333	331	329	314	311	309	297	175
10.0	683	252	249	247	244	243	241	240	229	227	225	216	128
11.0	513.0	189.0	187.0	186.0	183.0	183.0	181.0	180.0	172.0	171.0	169.0	163.0	96.0
11.5	449.0	166.0	164.0	163.0	160.0	160.0	159.0	158.0	151.0	149.0	148.0	142.0	84.0
12.0	395.0	146.0	144.0	143.0	141.0	141.0	140.0	139.0	133.0	131.0	130.0	125.0	73.9
13.0	311.0	115.0	113.0	113.0	111.0	111.0	110.0	109.0	104.0	103.0	103.0	98.5	58.2
14.0	249.0	91.8	90.8	90.1	88.9	88.5	87.9	87.3	83.5	82.7	82.2	78.8	46.6
15.0	202.0	74.6	73.8	73.3	72.2	72.0	71.5	71.0	67.9	67.2	66.8	64.1	37.9
16.0	167.0	61.5	60.8	60.4	59.5	59.3	58.9	58.5	56.0	55.4	55.1	52.8	31.2
17.0	139.0	51.3	50.7	50.3	49.6	49.5	49.1	48.8	46.7	46.2	45.9	44.0	26.0

[a] For sizes above 17 mm diameter, use the following formula: No. balls per kilogram = 1,910,000 ÷ [(nom. dia., mm)^3 × (material density, grams per cu. cm)].

Ball material densities in grams per cubic centimeter: aluminum, 2.796; aluminum bronze, 7.584; corrosion-resisting hardened steel, 7.677; AISI M-50 and silicon molybdenum steel, 7.723; chrome alloy steel, 7.833; carbon steel, 7.861; AISI 302 corrosion-resisting unhardened steel, 7.916; AISI 316 corrosion-resisting unhardened steel, 7.972; bronze, 8.415; brass and K-Monel metal, 8.470; Monel metal, 8.830; tungsten carbide, 14.947.

LUBRICANTS AND LUBRICATION

A lubricant is used for one or more of the following purposes: to reduce friction; to prevent wear; to prevent adhesion; to aid in distributing the load; to cool the moving elements; and to prevent corrosion.

The range of materials used as lubricants has been greatly broadened over the years, so that in addition to oils and greases, many plastics and solids and even gases are now being applied in this role. The only limitations on many of these materials are their ability to replenish themselves, to dissipate frictional heat, their reaction to high environmental temperatures, and their stability in combined environments. Because of the wide selection of lubricating materials available, great care is advisable in choosing the material and the method of application. The following types of lubricants are available: petroleum fluids, synthetic fluids, greases, solid films, working fluids, gases, plastics, animal fat, metallic and mineral films, and vegetable oils.

Lubricating Oils.—The most versatile and best-known lubricant is mineral oil. When applied in well-designed applications that provide for the limitations of both mechanical and hydraulic elements, oil is recognized as the most reliable lubricant. Concurrently, it is offered in a wide selection of stocks, carefully developed to meet the requirements of the specific application.

Lubricating oils are seldom marketed without additives blended for a narrow range of applications. These “additive packages” are developed for particular applications, so it is advisable to consult the sales-engineering representatives of a reputable petroleum company on the proper selection for the conditions under consideration. The following are the most common types of additives: wear preventive, oxidation inhibitor, rust inhibitor, detergent-dispersant, viscosity index improver, defoaming agent, and pour-point depressant.

A more recent development in the field of additives is a series of organic compounds that leave no ash when heated to a temperature high enough to evaporate or burn off the base oil. Initially produced for internal-combustion-engine applications these additives have found ready acceptance in those other applications where metallic or mineral trace elements would promote catalytic, corrosive, deposition, or degradation effects on mechanism materials.

Additives usually are not stable over the entire temperature and shear-rate ranges considered acceptable for the base stock oil application. Because of this problem, additive type oils must be carefully monitored to ensure that they are not continued in service after their principal capabilities have been diminished or depleted. Of primary importance in this regard is the action of the detergent-dispersant additives that function so well to reduce and control degradation products that would otherwise deposit on the operating parts and oil cavity walls. Because the materials cause the oil to carry a higher than normal amount of the breakdown products in a fine suspension, they may cause an accelerated deposition rate or foaming when they have been depleted or degenerated by thermal or contamination action. Ingestion of water by condensation or leaking can cause markedly harmful effects.

Viscosity index improvers serve to modify oils so that their change in viscosity is reduced over the operating temperature range. These materials may be used to improve both a heavy or a light oil; however, the original stock will tend to revert to its natural state when the additive has been depleted or degraded due to exposure to high temperatures or to the high shear rates normally encountered in the load-carrying zones of bearings and gears. In heavy-duty installations, it is generally advisable to select a heavier or a more highly refined oil (and one that is generally more costly) rather than to rely on a less stable viscosity-index-improvement product. Viscosity-index-improved oils are generally used in applications where the shear rate is well below 1,000,000 reciprocal seconds, as determined by the following formula:

$$\text{Shear rate}(s^{-1}) = \frac{DN}{60t}$$

where D is the journal diameter in inches, N is the journal speed in rpm, and t is the film thickness in inches.

Types of Oils.—Aside from being aware of the many additives available to satisfy particular application requirements and improve the performance of fluids, the designer must also be acquainted with the wide variety of oils, natural and synthetic, which are also available. Each oil has its own special features that make it suitable for specific applications and limit its utility in others. Though a complete description of each oil and its application feasibility cannot be given here, reference to major petroleum and chemical company sales engineers will provide full descriptions and sound recommendations. In some applications, however, it must be accepted that the interrelation of many variables, including shear rate, load, and temperature variations, prohibit precise recommendations or predictions of fluid durability and performance. Thus, prototype and rig testing are often required to ensure the final selection of the most satisfactory fluid.

The following table lists the major classifications and properties of available commercial petroleum oils.

Properties of Commercial Petroleum Oils and Their Applications

Group A				Group B			
Type	Viscosity, Centistokes		Density, g/cc at 60°F	Type	Viscosity, Centistokes		Density, g/cc at 60°F
	100°F	210°F			100°F	210°F	
SAE 10 W	41	6.0	0.870	General Purpose	22	3.9	0.880
SAE 20 W	71	8.5	0.885		44	6.0	0.898
SAE 30	114	11.2	0.890		66	7.0	0.915
SAE 40	173	14.5	0.890		110	9.9	0.915
SAE 50	270	19.5	0.900		200	15.5	0.890
Group C				Group D			
SAE 75	47	7.0	0.930, approx.	Turbine			
SAE 80	69	8.0		Light	32	5.5	0.871
SAE 90	285	20.5		Medium	65	8.1	0.876
SAE 140	725	34.0		Heavy	99	10.7	0.885
SAE 250	1,220	47.0					
Group E				Group F			
Aviation	5	1.5	0.858	Aviation	76	9.3	0.875
	10	2.5	0.864		268	20.0	0.891
					369	25.0	0.892

Group A. Automotive. With increased additives, diesel and marine reciprocating engines.
Group B. Gear trains and transmissions. With E. P. additives, hypoid gears.
Group C. Machine tools and other industrial applications.
Group D. Marine propulsion and stationary power turbines.
Group E. Turbojet engines.
Group F. Reciprocating engines.

Viscosity.—As noted before, fluids used as lubricants are generally categorized by their viscosity at 100 and 210 deg. F. Absolute viscosity is defined as a fluid's resistance to shear or motion—its internal friction in other words. This property is described in several ways, but basically it is the force required to move a plane surface of unit area with unit speed parallel to a second plane and at unit distance from it. In the metric system, the unit of viscosity is called the "poise" and in the English system is called the "reyn." One reyn is equal to 68,950 poises. One poise is the viscosity of a fluid, such that one dyne force is required to move a surface of one square centimeter with a speed of one centimeter per second, the distance between surfaces being one centimeter. The range of kinematic viscosity for a series of typical fluids is shown in the table on page 2312. Kinematic viscosity is related directly to the flow time of a fluid through the viscosimeter capillary. By multiplying the kinematic viscosity by the density of the fluid at the test temperature, one can determine the absolute viscosity. Because, in the metric system, the mass density is equal to the specific gravity, the conversion from kinematic to absolute viscosity is generally made in this system and then converted to English units where required. The densities of typical lubricating fluids with comparable viscosities at 100 deg. F and 210 deg. F are shown in this same table.

The following conversion table may be found helpful.

Multiply	By	To Get
Centipoises, Z, $\frac{\text{dyne-s}}{100\ \text{cm}^2}$	1.45×10^{-7}	Reyns, μ, $\frac{\text{lb force-s}}{\text{in.}^2}$
Centistokes, v, $\frac{\text{cm}^2}{100\ \text{s}}$	Density in g/cc	Centipoises, Z, $\frac{\text{dyne-s}}{100\ \text{cm}^2}$
Saybolt Universal Seconds, t_s	$0.22t_s - \frac{180}{t_s}$	Centistokes, v, $\frac{\text{cm}^2}{100\ \text{s}}$

Finding Specific Gravity of Oils at Different Temperatures.—The standard practice in the oil industry is to obtain a measure of specific gravity at 60 deg. F on an arbitrary scale, in degrees API, as specified by the American Petroleum Institute. As an example, API gravity, ρ_{API}, may be expressed as 27.5 degrees at 60 deg. F.

The relation between gravity in API degrees and specific gravity (grams of mass per cubic centimeter) at 60 deg. F, ρ_{60}, is

$$\rho_{60} = \frac{141.5}{131.5 + \rho_{API}}$$

The specific gravity, ρ_T, at some other temperature, T, is found from the equation

$$\rho_T = \rho_{60} - 0.00035(T - 60)$$

Normal values of specific gravity for sleeve-bearing lubricants range from 0.75 to 0.95 at 60 deg. F. If the API rating is not known, an assumed value of 0.85 may be used.

Application of Lubricating Oils.—In the selection and application of lubricating oils, careful attention must be given to the temperature in the critical operating area and its effect on oil properties. Analysis of each application should be made with detailed attention given to cooling, friction losses, shear rates, and contaminants.

Many oil selections are found to result in excessive operating temperatures because of a viscosity that is initially too high, which raises the friction losses. As a general rule, the lightest-weight oil that can carry the maximum load should be used. Where it is felt that the load carrying capacity is borderline, lubricity improvers may be employed rather than an arbitrarily higher viscosity fluid. It is well to remember that in many mechanisms the thicker fluid may increase friction losses sufficiently to lower the operating viscosity into the range provided by an initially lighter fluid. In such situations also, improved cooling, such as may be accomplished by increasing the oil flow, can improve the fluid properties in the load zone.

Similar improvements can be accomplished in many gear trains and other mechanisms by reducing churning and aeration through improved scavenging, direction of oil jets, and elimination of obstacles to the flow of the fluid. Many devices, such as journal bearings, are extremely sensitive to the effects of cooling flow and can be improved by greater flow rates with a lighter fluid. In other cases it is well to remember that the load carrying capacity of a petroleum oil is affected by pressure, shear rate, and bearing surface finish as well as initial viscosity and therefore these must be considered in the selection of the fluid. Detailed explanation of these factors is not within the scope of this text; however the technical representatives of the petroleum companies can supply practical guides for most applications.

Other factors to consider in the selection of an oil include the following:

1) Compatibility with system materials
2) Water absorption properties
3) Break-in requirements
4) Detergent requirements

5) Corrosion protection
6) Low temperature properties
7) Foaming tendencies
8) Boundary lubrication properties
9) Oxidation resistance (high temperature properties)
10) Viscosity/temperature stability (Viscosity Temperature Index).

Generally, the factors listed above are those which are usually modified by additives as described earlier. Since additives are used in limited amounts in most petroleum products, blended oils are not as durable as the base stock and must therefore be used in carefully worked-out systems. Maintenance procedures must be established to monitor the oil so that it may be replaced when the effect of the additive is noted or expected to degrade. In large systems supervised by a lubricating engineer, sampling and associated laboratory analysis can be relied on, while in customer-maintained systems as in automobiles and reciprocating engines, the design engineer must specify a safe replacement period which takes into account any variation in type of service or utilization.

Some large systems, such as turbine-power units, have complete oil systems which are designed to filter, cool, monitor, meter, and replenish the oil automatically. In such facilities, much larger oil quantities are used and they are maintained by regularly assigned lubricating personnel. Here reliance is placed on conservatively chosen fluids with the expectation that they will endure many months or even years of service.

Centralized Lubrication Systems.—Various forms of centralized lubrication systems are used to simplify and render more efficient the task of lubricating machines. In general, a central reservoir provides the supply of oil, which is conveyed to each bearing either through individual lines of tubing or through a single line of tubing that has branches extending to each of the different bearings. Oil is pumped into the lines either manually by a single movement of a lever or handle, or automatically by mechanical drive from some revolving shaft or other part of the machine. In either case, all bearings in the central system are lubricated simultaneously. Centralized force-feed lubrication is adaptable to various classes of machine tools such as lathes, planers, and milling machines and to many other types of machines. It permits the use of a lighter grade of oil, especially where complete coverage of the moving parts is assured.

Gravity Lubrication Systems.—Gravity systems of lubrication usually consist of a small number of distributing centers or manifolds from which oil is taken by piping as directly as possible to the various surfaces to be lubricated, each bearing point having its own independent pipe and set of connections. The aim of the gravity system, as of all lubrication systems, is to provide a reliable means of supplying the bearing surfaces with the proper amount of lubricating oil. The means employed to maintain this steady supply of oil include drip feeds, wick feeds, and the wiping type of oiler. Most manifolds are adapted to use either or both drip and wick feeds.

Drip-feed Lubricators: A drip feed consists of a simple cup or manifold mounted in a convenient position for filling and connected by a pipe or duct to each bearing to be oiled. The rate of feed in each pipe is regulated by a needle or conical valve. A loose-fitting cover is usually fitted to the manifold in order to prevent cinders or other foreign matter from becoming mixed with the oil. When a cylinder or other chamber operating under pressure is to be lubricated, the oil-cup takes the form of a lubricator having a tight-fitting screw cover and a valve in the oil line. To fill a lubricator of this kind, it is only necessary to close the valve and unscrew the cover.

Operation of Wick Feeds: For a wick feed, the siphoning effect of strands of worsted yarn is employed. The worsted wicks give a regular and reliable supply of oil and at the same time act as filters and strainers. A wick composed of the proper number of strands is fitted into each oil-tube. In order to insure using the proper sizes of wicks, a study should be made of the oil requirements of each installation, and the number of strands necessary to meet the demands of bearings at different rates of speed should be determined. When the necessary data have been obtained, a table should be prepared showing the size of wick or the number of strands to be used for each bearing of the machine.

Oil-conducting Capacity of Wicks: With the oil level maintained at a point $\frac{3}{8}$ to $\frac{3}{4}$ inch below the top of an oil-tube, each strand of a clean worsted yarn will carry slightly more than one drop of oil a minute. A twenty-four-strand wick will feed approximately thirty drops a minute, which is ordinarily sufficient for operating a large bearing at high speed. The wicks should be removed from the oil-tubes when the machinery is idle. If left in place, they will continue to deliver oil to the bearings until the supply in the cup is exhausted, thus wasting a considerable quantity of oil, as well as flooding the bearing. When bearings require an extra supply of oil temporarily, it may be supplied by dipping the wicks or by pouring oil down the tubes from an oil-can or, in the case of drip feeds, by opening the needle valves. When equipment that has remained idle for some time is to be started up, the wicks should be dipped and the moving parts oiled by hand to insure an ample initial supply of oil. The oil should be kept at about the same level in the cup, as otherwise the rate of flow will be affected. Wicks should be lifted periodically to prevent dirt accumulations at the ends from obstructing the flow of oil.

How Lubricating Wicks are Made: Wicks for lubricating purposes are made by cutting worsted yarn into lengths about twice the height of the top of the oil-tube above the bottom of the oil-cup, plus 4 inches. Half the required number of strands are then assembled and doubled over a piece of soft copper wire, laid across the middle of the strands. The free ends are then caught together by a small piece of folded sheet lead, and the copper wire twisted together throughout its length. The lead serves to hold the lower end of the wick in place, and the wire assists in forcing the other end of the wick several inches into the tube. When the wicks are removed, the free end of the copper wire may be hooked over the tube end to indicate which tube the wick belongs to. Dirt from the oil causes the wick to become gummy and to lose its filtering effect. Wicks that have thus become clogged with dirt should be cleaned or replaced by new ones. The cleaning is done by boiling the wicks in soda water and then rinsing them thoroughly to remove all traces of the soda. Oil-pipes are sometimes fitted with openings through which the flow of oil can be observed. In some installations, a short glass tube is substituted for such an opening.

Wiper-type Lubricating Systems: Wiper-type lubricators are used for out-of-the-way oscillating parts. A wiper consists of an oil-cup with a central blade or plate extending above the cup, and is attached to a moving part. A strip of fibrous material fed with oil from a source of supply is placed on a stationary part in such a position that the cup in its motion scrapes along the fibrous material and wipes off the oil, which then passes to the bearing surfaces.

Oil manifolds, cups, and pipes should be cleaned occasionally with steam conducted through a hose or with boiling soda water. When soda water is used, the pipes should be disconnected, so that no soda water can reach the bearings.

Oil Mist Systems.—A very effective system for both lubricating and cooling many elements which require a limited quantity of fluid is found in a device which generates a mist of oil, separates out the denser and larger (wet) oil particles, and then distributes the mist through a piping or conduit system. The mist is delivered into the bearing, gear, or lubricated element cavity through a condensing or spray nozzle, which also serves to meter the flow. In applications which do not encounter low temperatures or which permit the use of visual devices to monitor the accumulation of solid oil, oil mist devices offer advantages in providing cooling, clean lubricant, pressurized cavities which prevent entrance of contaminants, efficient application of limited lubricant quantities, and near-automatic performance. These devices are supplied with fluid reservoirs holding from a few ounces up to several gallons of oil and with accommodations for either accepting shop air or working from a self-contained compressor powered by electricity. With proper control of the fluid temperature, these units can atomize and dispense most motor and many gear oils.

Lubricating Greases.—In many applications, fluid lubricants cannot be used because of the difficulty of retention, relubrication, or the danger of churning. To satisfy these and other requirements such as simplification, greases are applied. These formulations are usually petroleum oils thickened by dispersions of soap, but may consist of synthetic oils with soap or inorganic thickeners, or oil with silaceous dispersions. In all cases, the thickener, which must be carefully prepared and mixed with the fluid, is used to immobilize the oil, serving as a storehouse from which the oil bleeds at a slow rate. Though the thickener very often has lubricating properties itself, the oil bleeding from the bulk of the grease is the determining lubricating function. Thus, it has been shown that when the oil has been depleted to the level of 50 per cent of the total weight of the grease, the lubricating ability of the material is no longer reliable. In some applications requiring an initially softer and wetter material, however, this level may be as high as 60 per cent.

Grease Consistency Classifications.—To classify greases as to mobility and oil content, they are divided into Grades by the NLGI (National Lubricating Grease Institute). These grades, ranging from 0, the softest, up through 6, the stiffest, are determined by testing in a penetrometer, with the depth of penetration of a specific cone and weight being the controlling criterion. To insure proper averaging of specimen resistance to the cone, most specifications include a requirement that the specimen be worked in a sieve-like device before being packed into the penetrometer cup for the penetration test. Since many greases exhibit thixotropic properties (they soften with working, as they often do in an application with agitation of the bulk of the grease by the working elements or accelerations), this penetration of the worked specimen should be used as a guide to compare the material to the original manufactured condition of it and other greases, rather than to the exact condition in which it will be found in the application. Conversely, many greases are found to stiffen when exposed to high shear rates at moderate loads as in automatic grease dispensing equipment. The application of a grease, therefore must be determined by a carefully planned cut-and-try procedure. Most often this is done by the original equipment manufacturer with the aid of the petroleum company representatives, but in many cases it is advisable to include the bearing engineer as well. In this general area it is well to remember that shock loads, axial or thrust movement within or on the grease cavity can cause the grease to contact the moving parts and initiate softening due to the shearing or working thus induced. To limit this action, grease-lubricated bearing assemblies often utilize dams or dividers to keep the bulk of the grease contained and unchanged by this working. Successful application of a grease depends however, on a relatively small amount of mobile lubricant (the oil bled out of the bulk) to replenish that small amount of lubricant in the element to be lubricated. If the space between the bulk of the mobile grease and the bearing is too large, then a critical delay period (which will be regulated by the grease bleed rate and the temperature at which it is held) will ensue before lubricant in the element can be resupplied. Since most lubricants undergo some attrition due to thermal degradation, evaporation, shearing, or decomposition in the bearing area to which applied, this delay can be fatal.

To prevent this from leading to failure, grease is normally applied so that the material in the cavity contacts the bearing in the lower quadrants, insuring that the excess originally packed into it impinges on the material in the reservoir. With the proper selection of a grease which does not slump excessively, and a reservoir construction to prevent churning, the initial action of the bearing when started into operation will be to purge itself of excess grease, and to establish a flow path for bleed oil to enter the bearing. For this purpose, most greases selected will be of a grade 2 or 3 consistency, falling into the “channelling” variety or designation.

Types of Grease.—Greases are made with a variety of soaps and are chosen for many particular characteristics. Most popular today, however, are the lithium, or soda-soap grease and the modified-clay thickened materials. For high temperature applications (250 deg. F. and above) certain finely divided dyes and other synthetic thickeners are applied. For all-around use the lithium soap greases are best for moderate temperature applications (up to 225 deg. F.) while a number of soda-soap greases have been found to work well up to 285 deg. F. Since the major suppliers offer a number of different formulations for these temperature ranges it is recommended that the user contact the engineering representatives of a reputable petroleum company before choosing a grease. Greases also vary in volatility and viscosity according to the oil used. Since the former will affect the useful life of the bulk applied to the bearing and the latter will affect the load carrying capacity of the grease, they must both be considered in selecting a grease.

For application to certain gears and slow-speed journal bearings, a variety of greases are thickened with carbon, graphite, molybdenum disulfide, lead, or zinc oxide. Some of these materials are likewise used to inhibit fretting corrosion or wear in sliding or oscillating mechanisms and in screw or thread applications. One material used as a “gear grease” is a residual asphaltic compound which is known as a “Crater Compound.” Being extremely stiff and having an extreme temperature-viscosity relationship, its application must also be made with careful consideration of its limitations and only after careful evaluation in the actual application. Its oxidation resistance is limited and its low mobility in winter temperature ranges make it a material to be used with care. However, it is used extensively in the railroad industry and in other applications where containment and application of lubricants is difficult. In such conditions its ability to adhere to gear and chain contact surfaces far outweighs its limitations and in some extremes it is “painted” onto the elements at regular intervals.

Temperature Effects on Grease Life.—Since most grease applications are made where long life is important and relubrication is not too practical, operating temperatures must be carefully considered and controlled. Being a hydro-carbon, and normally susceptible to oxidation, grease is subject to the general rule that: Above a critical threshold temperature, each 15- to 18-deg. F. rise in temperature reduces the oxidation life of the lubricant by half. For this reason, it is vital that all elements affecting the operating temperature of the application be considered, correlated, and controlled. With sealed-for-life bearings, in particular, grease life must be determined for representative bearings and limits must be established for all subsequent applications.

Most satisfactory control can be established by measuring bearing temperature rise during a controlled test, at a consistent measuring point or location. Once a base line and limiting range are determined, all deviating bearings should be dismantled, inspected, and reassembled with fresh lubricant for retest. In this manner mavericks or faulty assemblies will be ferreted out and the reliability of the application established. Generally, a well lubricated grease packed bearing will have a temperature rise above ambient, as measured at the outer race, of from 10 to 50 deg. F. In applications where heat is introduced into the bearing through the shaft or housing, a temperature rise must be added to that of the frame or shaft temperature.

In bearing applications care must be taken not to fill the cavity too full. The bearing should have a practical quantity of grease worked into it with the rolling elements thoroughly coated and the cage covered, but the housing (cap and cover) should be no more than 75 per cent filled; with softer greases, this should be no more than 50 per cent. Excessive packing is evidenced by overheating, churning, aerating, and eventual purging with final failure due to insufficient lubrication. In grease lubrication, *never* add a bit more for good luck — hold to the prescribed amount and determine this with care on a number of representative assemblies.

Relubricating with Grease.—In some applications, sealed-grease methods are not applicable and addition of grease at regular intervals is required. Where this is recommended by the manufacturer of the equipment, or where the method has been worked out as part of a development program, the procedure must be carefully followed. *First,* use the proper lubricant — the same as recommended by the manufacturer or as originally applied (grease performance can be drastically impaired if contaminated with another lubricant). *Second,* clean the lubrication fitting thoroughly with materials which will not affect the mechanism or penetrate into the grease cavity. *Third,* remove the cap (and if applicable, the drain or purge plug). *Fourth,* clean and inspect the drain or scavenge cavity. *Fifth,* weigh the grease gun or calibrate it to determine delivery rate. *Sixth,* apply the directed quantity or fill until grease is detected coming out the drain or purge hole. *Seventh,* operate the mechanism with the drain open so that excess grease is purged. *Last,* continue to operate the mechanism while determining the temperature rise and insure that it is within limits. Where there is access to a laboratory, samples of the purged material may be analyzed to determine the deterioration of the lubricant and to search for foreign material which may be evidence of contamination or of bearing failure.

Normally, with modern types of grease and bearings, lubrication need only be considered at overhaul periods or over intervals of three to ten years.

Solid Film Lubricants.—Solids such as graphite, molybdenum disulfide, polytetrafluoroethylene, lead, babbit, silver, or metallic oxides are used to provide dry film lubrication in high-load, slow-speed or oscillating load conditions. Though most are employed in conjunction with fluid or grease lubricants, they are often applied as the primary or sole lubricant where their inherent limitations are acceptable. Of foremost importance is their inability to carry away heat. Second, they cannot replenish themselves, though they generally do lay down an oriented film on the contacting interface. Third, they are relatively immobile and must be bonded to the substrate by a carrier, by plating, fusing, or by chemical or thermal deposition.

Though these materials do not provide the low coefficient of friction associated with fluid lubrication, they do provide coefficients in the range of 0.4 down to 0.02, depending on the method of application and the material against which they rub. Polytetrafluoroethylene, in normal atmospheres and after establishing a film on both surfaces has been found to exhibit a coefficient of friction down to 0.02. However, this material is subject to cold flow and must be supported by a filler or on a matrix to continue its function. Since it can now be cemented in thin sheets and is often supplied with a fine glass fiber filler, it is practical in a number of installations where the speed and load do not combine to melt the bond or cause the material to sublime.

Bonded films of molybdenum disulfide, using various resins and ceramic combinations as binders, are deposited over phosphate treated steel, aluminum, or other metals with good success. Since its action produces a gradual wear of the lubricant, its life is limited by the thickness which can be applied (not over a thousandth or two in the conventional application). In most applications this is adequate if the material is used to promote break-in, prevent galling or pick-up, and to reduce fretting or abrasion in contacts otherwise impossible to separate.

In all applications of solid film lubricants, the performance of the film is limited by the care and preparation of the surface to which they are applied. If they can't adhere properly, they cannot perform, coming off in flakes and often jamming under flexible components. The best advice is to seek the assistance of the supplier's field engineer and set up a close control of the surface preparation and solid film application procedure. It should be noted that the functions of a good solid film lubricant cannot overcome the need for better surface finishing. Contacting surfaces should be smooth and flat to insure long life and minimum friction forces. Generally, surfaces should be finished to no more than 24 micro-inches AA with wariness no greater than 0.00002 inch.

Anti-friction Bearing Lubrication.—The limiting factors in bearing lubrication are the load and the linear velocity of the centers of the balls or rollers. Since these are difficult to evaluate, a speed factor which consists of the inner race bore diameter × RPM is used as a criterion. This factor will be referred to as S_i where the bore diameter is in inches and S_m where it is in millimeters.

In order to be suitable for use in anti-friction bearings, grease must have the following properties:

1) Freedom from chemically or mechanically active ingredients such as uncombined metals or oxides, and similar mineral or solid contaminants.

2) The slightest possible tendency of change in consistency, such as thickening, separation of oil, evaporation or hardening.

3) A melting point considerably higher than the operating temperatures.

The choice of lubricating oils is easier. They are more uniform in their characteristics and if resistant to oxidation, gumming and evaporation, can be selected primarily with regard to a suitable viscosity.

Grease Lubrication: Anti-friction bearings are normally grease lubricated, both because grease is much easier than oil to retain in the housing over a long period and because it acts to some extent as a seal against the entry of dirt and other contaminants into the bearings. For almost all applications, a No. 2 soda-base grease or a mixed-base grease with up to 5 per cent calcium soap to give a smoother consistency, blended with an oil of around 250 to 300 SSU (Saybolt Universal Seconds) at 100 degrees F. is suitable. In cases where speeds are high, say S_i is 5000 or over, a grease made with an oil of about 150 SSU at 100 degrees F. may be more suitable especially if temperatures are also high. In many cases where bearings are exposed to large quantities of water, it has been found that a standard soda-base ball-bearing grease, although classed as water soluble gives better results than water-insoluble types. Greases are available that will give satisfactory lubrication over a temperature range of −40 degrees to +250 degrees F.

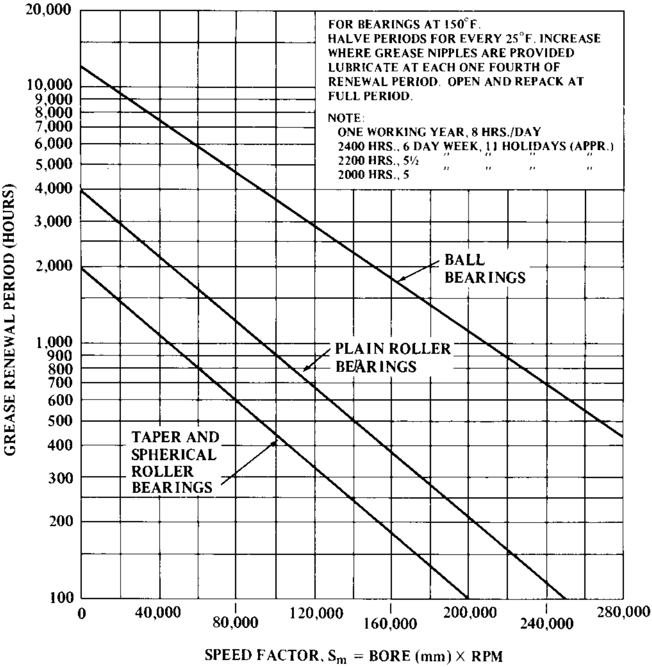
Conservative grease renewal periods will be found in the accompanying chart. Grease should not be allowed to remain in a bearing for longer than 48 months or if the service is very light and temperatures low, 60 months, irrespective of the number of hours' operation during that period as separation of the oil from the soap and oxidation continue whether the bearing is in operation or not.

Before renewing the grease in a hand-packed bearing, the bearing assembly should be removed and washed in clean kerosene, degreasing fluid or other solvent. As soon as the bearing is quite clean it should be washed at once in clean light mineral oil, preferably rust-inhibited. The bearing should *not be spun* before or while it is being oiled. Caustic solutions may be used if the old grease is hard and difficult to remove, but the best method is to soak the bearing for a few hours in light mineral oil, preferably warmed to about 130 degrees F., and then wash in cleaning fluid as described above. The use of chlorinated solvents is best avoided.

When replacing the grease, it should be forced with the fingers between the balls or rollers, dismantling the bearing, if convenient. The available space inside the bearing should be filled completely and the bearing then spun by hand. Any grease thrown out should be wiped off. The space on each side of the bearing in the housing should be not more than half-filled. Too much grease will result in considerable churning, high bearing temperatures and the possibility of early failure. Unlike any other kind of bearing, anti-friction bearings more often give trouble due to over-rather than to under-lubrication.

Grease is usually not very suitable for speed factors over 12,000 for S_i or 300,000 for S_m (although successful applications have been made up to an S_i of 50,000) or for temperatures much over 210 degrees F., 300 degrees F. being the extreme practical upper limit, even if synthetics are used. For temperatures above 210 degrees F., the grease renewal periods are very short.

Oil Lubrication: Oil lubrication is usually adopted when speeds and temperatures are high or when it is desired to adopt a central oil supply for the machine as a whole. Oil for anti-friction bearing lubrication should be well refined with high film strength and good resistance to oxidation and good corrosion protection. Anti-oxidation additives do no harm but are not really necessary at temperatures below about 200 degrees F. Anti-corrosion additives are always desirable. The accompanying table gives recommended viscosities of oil for ball bearing lubrication other than by an air-distributed oil mist. Within a given temperature and speed range, an oil towards the lighter end of the grade should be used, if convenient, as speeds increase. Roller bearings usually require an oil one grade heavier than do ball bearings for a given speed and temperature range. Cooled oil is sometimes circulated through an anti-friction bearing to carry off excess heat resulting from high speeds and heavy loads.



Oil Viscosities and Temperature Ranges for Ball Bearing Lubrication

Maximum Temperature Range Degrees F.	Optimum Temperature Range, Degrees F.	Speed Factor, S_i[a] Under 1000	Speed Factor, S_i[a] Over 1000
		Viscosity	
− 40 to + 100	− 40 to − 10	80 to 90 SSU[b]	70 to 80 SSU[b]
− 10 to + 100	− 10 to + 30	100 to 115 SSU[b]	80 to 100 SSU[b]
+ 30 to + 150	+ 30 to + 150	SAE 20	SAE 10
+ 30 to + 200	+ 150 to + 200	SAE 40	SAE 30
+ 50 to + 300	+ 200 to + 300	SAE 70	SAE 60

[a] Inner race bore diameter (inches) × RPM.
[b] At 100 deg. F.

Not applicable to air-distributed oil mist lubrication.

Aerodynamic Lubrication

A natural extension of hydrodynamic lubrication consists in using air or some other gas as the lubricant. The viscosity of air is 1,000 times smaller than that of a very thin mineral oil. Consequently, the viscous resistance to motion is very much less. However, the distance of nearest approach, i.e. the closest distance between the shaft and the bearing is also correspondingly smaller, so that special precautions must be taken.

To obtain full benefit from such aerodynamic lubrication, the surfaces must have a very fine finish, the alignment must be very good, the speeds must be high and the loading relatively low. If all these conditions are fulfilled extremely successful bearing system can be made to run at very low coefficients of friction. They may also operate at very high temperatures since chemical degradation of the lubricant need not occur. Furthermore, if air is used as the lubricant, it costs nothing. This type of lubrication mechanism is very important for oil-free compressors and gas turbines. Another area of growing application for aerodynamic bearings is in data recording heads for computers. Air is used as the lubricant for the recording heads which are designed to be separated from the magnetic recording disc by a thin air film. The need for high recording densities in magnetic discs necessitates the smallest possible air film thickness between the head and disc. A typical thickness is around 1μm.

The analysis of aerodynamic bearings is very similar to liquid hydrodynamic bearings. The main difference, however, is that the gas compressibility is now a distinctive feature and has to be incorporated into the analysis.

Elastohydrodynamic Lubrication.—In the arrangement of the shaft and bearing it is usually assumed that the surfaces are perfectly rigid and retain their geometric shape during operation. However, a question might be posed: what is the situation if the geometry or mechanical properties of the materials are such that appreciable elastic deformation of the surfaces occurs? Suppose a steel shaft rests on a rubber block. It deforms the block elastically and provides an approximation to a half-bearing (see Figure 1 a).

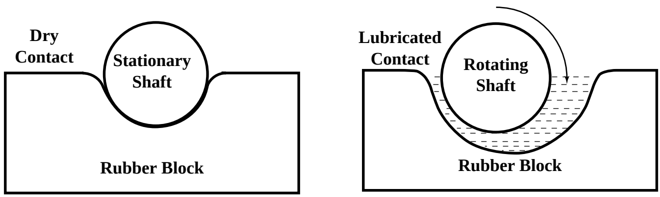


Fig. 1a. Fig. 1b.

If a lubricant is applied to the system it will be dragged into the interface and, if the conditions are right, it will form a hydrodynamic film. However, the pressures developed in the oil film will now have to match up with the elastic stresses in the rubber. In fact the shape of the rubber will be changed as indicated in Figure 1 b.

This type of lubrication is known as elastohydrodynamic lubrication. It occurs between rubber seals and shafts. It also occurs, rather surprisingly, in the contact between a windshield wiper blade and a windshield in the presence of rain. The geometry of the deformable member, its elastic properties, the load, the speed and the viscosity of the liquid and its dependence on the contact pressure are all important factors in the operation of elastohydrodynamic lubrication.

With conventional journals and bearings the average pressure over the bearing is of the order of 7×10^{-6} N/rn^2. With elastohydrodynamic bearings using a material such as rubber the pressures are perhaps 10 to 20 times smaller. At the other end of pressure spectrum, for instance in gear teeth, contact pressures of the order of $700\text{x}10^6$ N/in^2 may easily be reached. Because the metals used for gears are very hard this may still be within the range of elastic deformation. With careful alignment of the engaging gear teeth and appropriate surface finish, gears can in fact run successfully under these conditions using an ordinary mineral oil as the lubricant. If the thickness of the elastohydrodynamic film formed at such pressures is calculated it will be found that it is less than an atomic diameter. Sincc even the smoothest metal surfaces are far rougher than this (a millionth of an inch is about 100 atomic diameters) it seems hard to understand why lubrication is effective in these circumstances.

The explanation was first provided by A.N.Grubin in 1949 and a little later (1958) by A.W.Crook. With most mineral oils the application of a high pressure can lead to an enormous increase in viscosity. For example, at a pressure of 700×10^6 N/rn^2 the viscosity may be increased 10,000-fold. The oil entering the gap between the gear teeth is trapped between the surfaces and at the high pressures existing in the contact region behaves virtually like a solid separating layer. This process explains why many mechanisms in engineering practice operate under much severer conditions than the classical theory would allow.

This type of elastohydrodynamic lubrication becomes apparent only when the film thickness is less than about 0.25 to 1 μm. To be exploited successfully it implies that the surfaces must be very smooth and very carefully aligned. If these conditions are met systems such as gears or cams and tappets can operate effectively at very high contact pressures without any metallic contact occurring. The coefficient of friction depends on the load, contact geometry, speed, etc., but generally it lies between about $\mu = 0.01$ at the lightest pressures and $\mu = 0.1$ at the highest pressures. The great success of elastohydrodynamic theory in explaining effective lubrication at very high contact pressures also raises a problem that has not yet been satisfactorily resolved: why do lubricants ever fail, since the harder they are squeezed the harder it is to extrude them? It is possible that high temperature flashes are responsible; alternatively the high rates of shear can actually fracture the lubricant film since when it is trapped between the surfaces it is, instantaneously, more like a wax than an oil.

It is clear that in this type of lubrication the effect of pressure on viscosity is a factor of major importance. It turns out that mineral oils have reasonably good pressure-viscosity characteristics. It appears that synthetic oils do not have satisfactory pressure-viscosity characteristics.

In engineering, two most frequently encountered types of contact are line contact and point contact.

The film thickness for line contact (gears, cam-tappet) can be estimated from:

$$h_o = 2.65 \frac{\alpha^{0.54} (\eta_o U)^{0.7} R_e^{0.43}}{w^{0.13} E_e^{0.03}}$$

In the case of point contact (ball bearings), the film thickness is given by:

$$h_o = 0.84 \alpha \eta_o U^{0.74} 0.41 R_e \left(\frac{E_e}{W} \right)^{0.074}$$

In the above equations the symbols used are defined as:

- α = the pressure-viscosity coefficient. A typical value for a mineral oil is 1.8x1 0-8 m^2/N
- ν = the viscosity of the lubricant at atmospheric pressure Ns/m^2
- U = the entraining surface velocity, $U = (U_A + U_B)/2$ m/s, where the subscripts A and B refer to the velocities of bodies 'A' and 'B' respectively.
- W = the load on the contact, N
- w = the load per unit width of line contact, N/m

E_O = the reduced Young's modulus $\frac{1}{E_e} = \frac{1}{2}\left(\frac{1-\upsilon_A^2}{E_A} + \frac{1-\upsilon_B^2}{E_B}\right)$ N/m² where 'ν_A and ν_B are the Poisson's ratios of the contacting bodies 'A' and 'B' respectively; E_A and E_B are the Young's moduli of the contacting bodies 'A' and 'B' respectively.

R_e = - is the reduced radius of curvature (meters) and is given by different equations for different contact configurations.

In ball bearings (see Figure 2) the reduced radius is given by:

- contact between the ball and inner race: $R_e = \frac{rR_1}{R_1 + r}$
- contact between the ball and outer race: $R_e = \frac{r(R_1 + 2r)}{R_1 + r}$

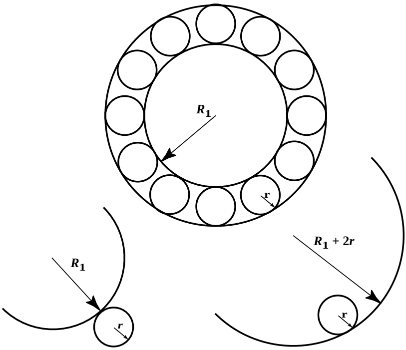


Fig. 2.

For involute gears it can readily be shown that the contact at a distance s from the pitch point can be represented by two cylinders of radii $R_{1,2}$ sinψ + s rotating with the angular velocity of the wheels (see Figure 3). In the expression below R_1 or R_2 represent pitch radii of the wheels and ψ is the pressure angle. Thus,

$$R_e = \frac{(R_1 \sin\psi + s)(R_2 \sin\psi + s)}{(R_1 + R_2)\sin\psi}$$

The thickness of the film developed in the contact zone between smooth surfaces must be related to the topography of the actual surfaces. The most commonly used parameter for this purpose is the specific film thickness defined as the ratio of the minimum film thickness for smooth surfaces (given by the above equations) to the roughness parameter of the contacting surfaces.

$$\lambda = \frac{h_o}{\sqrt{R_{m1}^2 + R_{m2}^2}}$$

where $R_m = 1.11R_a$ is the root-mean-square height of surface asperities, and R_a is the centre-line-average height of surface asperities.

If λ is greater than 3 then it is usually assumed that there is full separation of contacting bodies by an elastohydrodynamic film.

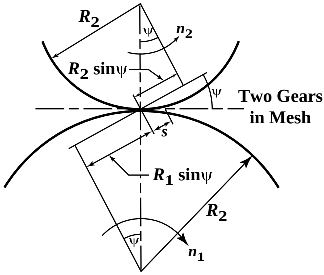


Fig. 3a.

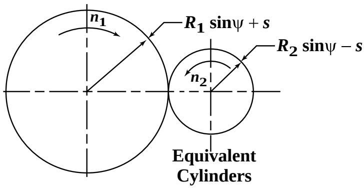


Fig. 3b.

Viscosity-pressure relationship.—Lubricant viscosity increases with pressure. For most lubricants this effect is considerably larger than the effect of temperature or shear when the pressure is appreciably above atmospheric. This is of fundamental importance in the lubrication of highly loaded concentrated contacts such as in rolling contact bearings, gears and cam-tappet systems.

The best known equation to calculate the viscosity of a lubricant at moderate pressures is the Barus equation.

$$\eta_p = \eta_o e^{\alpha p}$$

where η is the viscosity at pressure p (Ns/m^2), η_0 is the viscosity at atmospheric pressure (Ns/in^2), α is the pressure-viscosity coefficient (m^2/N) which can be obtained by plotting the natural logarithm of dynamic viscosity η measured at pressure p. The slope of the graph is α and p is the pressure of concern (N/m^2).

Values of dynamic viscosity η and pressure-viscosity coefficient α for most commonly used lubricants are given in Table 1.

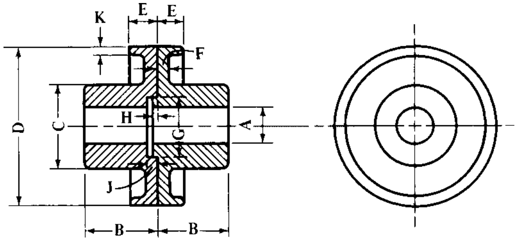
Table 1. Dynamic Viscosity η and Pressure-viscosity Coefficient α for Lubricants

Lubricant	Dynamic viscosity η measured at atmospheric pressure and room temperature $\eta \times 10^{-3}$ Ns/m^2	Pressure-viscosity coefficient α measured at room temperature $\alpha \times 10^{-3}$ m^2/N
Light machine oil	45	28
Heavy machine oil	153	23.7
Cylinder oil	810	34
Spindle oil	18.6	20
Medicinal whale oil	107	29.5
Castor oil	360	15.9
Glycerol (glycerine)	535	5.9

COUPLINGS AND CLUTCHES

Connecting Shafts.—For couplings to transmit up to about 150 horsepower, simple flange-type couplings of appropriate size, as shown in the table, are commonly used. The design shown is known as a safety flange coupling because the bolt heads and nuts are shrouded by the flange, but such couplings today are normally shielded by a sheet metal or other cover.

Safety Flange Couplings



										Bolts	
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	No.	Dia.
1	1 3/4	2 1/4	4	11/16	5/16	1 1/2	1/4	9/32	1/4	5	3/8
1 1/4	2 3/16	2 3/4	5	13/16	3/8	1 7/8	1/4	9/32	1/4	5	7/16
1 1/2	2 5/8	3 3/8	6	15/16	7/16	2 1/4	1/4	9/32	1/4	5	1/2
1 3/4	3 1/16	4	7	1 1/16	1/2	2 5/8	1/4	9/32	1/4	5	9/16
2	3 1/2	4 1/2	8	1 3/16	9/16	3	1/4	9/32	5/16	5	5/8
2 1/4	3 15/16	5 1/8	9	1 5/16	5/8	3 3/8	1/4	9/32	5/16	5	11/16
2 1/2	4 3/8	5 5/8	10	1 7/16	11/16	3 3/4	1/4	9/32	5/16	5	3/4
2 3/4	4 13/16	6 1/4	11	1 9/16	3/4	4 1/8	1/4	9/32	5/16	5	13/16
3	5 1/4	6 3/4	12	1 11/16	13/16	4 1/2	1/4	9/32	3/8	5	7/8
3 1/4	5 11/16	7 3/8	13	1 13/16	7/8	4 7/8	1/4	9/32	3/8	5	15/16
3 1/2	6 1/8	8	14	1 15/16	15/16	5 1/4	1/4	9/32	3/8	5	1
3 3/4	6 9/16	8 1/2	15	2 1/16	1	5 5/8	1/4	9/32	3/8	5	1 1/16
4	7	9	16	2 1/4	1 1/8	6	1/4	9/32	7/16	5	1 1/8
4 1/2	7 7/8	10 1/4	18	2 1/2	1 1/4	6 3/4	1/4	9/32	7/16	5	1 1/4
5	8 3/4	11 1/4	20	2 3/4	1 3/8	7 1/2	1/4	9/32	7/16	5	1 3/8
5 1/2	8 3/4	11 1/4	20	2 3/4	1 3/8	7 1/2	1/4	9/32	7/16	5	1 3/8
6	10 1/2	12 3/8	22	2 15/16	1 1/2	8 1/4	5/16	11/32	1/2	5	1 7/16
6 1/2	11 3/8	13 1/2	24	3 1/8	1 5/8	9	5/16	11/32	1/2	5	1 1/2
7	12 1/4	14 5/8	26	3 1/4	1 3/4	9 3/4	5/16	11/32	9/16	6	1 1/2
7 1/2	13 1/8	15 3/4	28	3 7/16	1 7/8	10 1/2	5/16	11/32	9/16	6	1 9/16
8	14	16 7/8	28	3 1/2	2	10 7/8	5/16	11/32	5/8	7	1 1/2
8 1/2	14 7/8	18	30	3 11/16	2 1/8	11 1/4	5/16	11/32	5/8	7	1 9/16
9	15 3/4	19 1/8	31	3 3/4	2 1/4	11 5/8	5/16	11/32	11/16	8	1 1/2
9 1/2	16 5/8	20 1/4	32	3 15/16	2 3/8	12	5/16	11/32	11/16	8	1 9/16
10	17 1/2	21 3/8	34	4 1/8	2 1/2	12 3/4	5/16	11/32	3/4	8	1 5/8
10 1/2	18 3/8	22 1/2	35	4 1/4	2 5/8	13 1/8	5/16	11/32	3/4	10	1 5/8
11	19 1/4	23 5/8	36	4 7/16	2 3/4	13 1/2	5/16	11/32	7/8	10	1 11/16
11 1/2	20 1/8	24 3/4	37	4 5/8	2 7/8	13 7/8	5/16	11/32	7/8	10	1 3/4
12	21	25 7/8	38	4 13/16	3	14 1/4	5/16	11/32	1	10	1 13/16

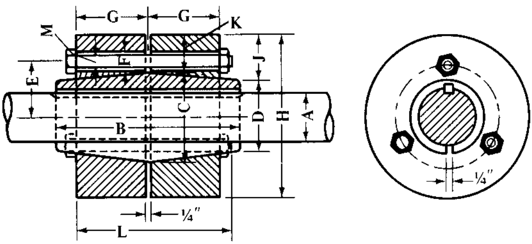
For small sizes and low power applications, a setscrew may provide the connection between the hub and the shaft, but higher power usually requires a key and perhaps two setscrews, one of them above the key. A flat on the shaft and some means of locking the setscrew(s) in position are advisable. In the AGMA Class I and II fits the shaft tolerances are –0.0005 inch from ½ to 1 ½ inches diameter and -0.001 inch on larger diameters up to 7 inches.

Class I coupling bore tolerances are + 0.001 inch up to 1 ½ inches diameter, then + 0.0015 inch to 7 inches diameter. Class II coupling bore tolerances are + 0.002 inch on sizes up to 3 inches diameter, + 0.003 inch on sizes from 3 ¼ through 3¾ inches diameter, and + 0.004 inch on larger diameters up to 7 inches.

Interference Fits.—Components of couplings transmitting over 150 horsepower often are made an interference fit on the shafts, which may reduce fretting corrosion. These couplings may or may not use keys, depending on the degree of interference. Keys may range in size from ⅛ inch wide by 1⁄16 inch high for ½-inch diameter shafts to 1 ¾ inches wide by ⅞ inch high for 7-inch diameter shafts. Couplings transmitting high torque or operating at high speeds or both may use two keys. Keys must be a good fit in their keyways to ensure good transmission of torque and prevent failure. AGMA standards provide recommendations for square parallel, rectangular section, and plain tapered keys, for shafts of 5⁄16 through 7 inches diameter, in three classes designated commercial, precision, and fitted. These standards also cover keyway offset, lead, parallelism, finish and radii, and face keys and splines. (See also ANSI and other Standards in Keys and Keyways section of this Handbook.)

Double-cone Clamping Couplings.—As shown in the table, double-cone clamping couplings are made in a range of sizes for shafts from 1 7⁄16 to 6 inches in diameter, and are easily assembled to shafts. These couplings provide an interference fit, but they usually cost more and have larger overall dimensions than regular flanged couplings.

Double-cone Clamping Couplings



A	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	*K*	*L*	*M*	No. of Bolts	No. of Keys
17⁄16	5¼	2¾	2⅛	1⅝	⅝	2⅛	4¾	1⅛	1	5	½	3	1
115⁄16	7	3½	2⅞	2⅛	⅝	2¾	6¼	1⅛	1⅜	6¼	½	3	1
27⁄16	8¾	45⁄16	3⅝	3	¾	3½	713⁄16	1⅞	1¾	7⅞	⅝	3	1
3	10½	5½	43⁄32	3½	¾	43⁄16	9	2¼	2	9½	⅝	3	1
3½	12¼	7	5⅜	4⅜	⅞	51⁄16	11¼	2⅝	2⅛	11¼	¾	4	1
4	14	7	5½	4¾	⅞	5½	12	3¾	2½	12	¾	4	1
4½	15½	8	6⅞	5¼	⅞	6¾	13½	3¾	2¾	14½	¾	4	1
5	17	9	7¼	5¾	⅞	7	15	3¾	3	15¼	¾	4	1
5½	17½	9½	7¾	6¼	1	7	15½	3¾	3	15¼	⅞	4	1
6	18	10	8¼	6¾	1	7	16	3¾	3	15¼	⅞	4	2

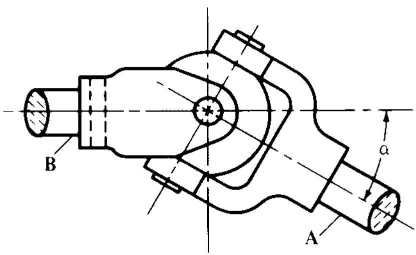
Flexible Couplings.—Shafts that are out of alignment laterally or angularly can be connected by any of several designs of flexible couplings. Such couplings also permit some degree of axial movement in one or both shafts. Some couplings use disks or diaphragms to transmit the torque. Another simple form of flexible coupling consists of two flanges connected by links or endless belts made of leather or other strong, pliable material. Alternatively, the flanges may have projections that engage spacers of molded rubber or other flexible materials that accommodate uneven motion between the shafts. More highly developed flexible couplings use toothed flanges engaged by correspondingly toothed elements, permitting relative movement. These couplings require lubrication unless one or more of the elements is made of a self-lubricating material. Other couplings use diaphragms or bellows that can flex to accommodate relative movement between the shafts.

The Universal Joint.—This form of coupling, originally known as a Cardan or Hooke's coupling, is used for connecting two shafts the axes of which are not in line with each other, but which merely intersect at a point. There are many different designs of universal joints or couplings, which are based on the principle embodied in the original design. One well-known type is shown by the accompanying diagram.

As a rule, a universal joint does not work well if the angle α (see illustration) is more than 45 degrees, and the angle should preferably be limited to about 20 degrees or 25 degrees, excepting when the speed of rotation is slow and little power is transmitted.

Variation in Angular Velocity of Driven Shaft: Owing to the angularity between two shafts connected by a universal joint, there is a variation in the angular velocity of one shaft during a single revolution, and because of this, the use of universal couplings is sometimes prohibited. Thus, the angular velocity of the driven shaft will not be the same at all points of the revolution as the angular velocity of the driving shaft. In other words, if the driving shaft moves with a uniform motion, then the driven shaft will have a variable motion and, therefore, the universal joint should not be used when absolute uniformity of motion is essential for the driven shaft.

Determining Maximum and Minimum Velocities: If shaft *A* (see diagram) runs at a constant speed, shaft *B* revolves at maximum speed when shaft A occupies the position shown in the illustration, and the minimum speed of shaft *B* occurs when the fork of the driving shaft *A* has turned 90 degrees from the position illustrated. The maximum speed of the driven shaft may be obtained by multiplying the speed of the driving shaft by the secant of angle α. The minimum speed of the driven shaft equals the speed of the driver multiplied by cosine α. Thus, if the driver rotates at a constant speed of 100 revolutions per minute and the shaft angle is 25 degrees, the maximum speed of the driven shaft is at a rate equal to $1.1034 \times 100 = 110.34$ R.P.M. The minimum speed rate equals $0.9063 \times 100 = 90.63$; hence, the extreme variation equals $110.34 - 90.63 = 19.71$ R.P.M.



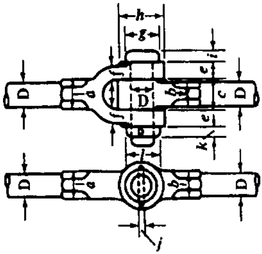
Use of Intermediate Shaft between Two Universal Joints.—The lack of uniformity in the speed of the driven shaft resulting from the use of a universal coupling, as previously explained, is objectionable for some forms of mechanisms. This variation may be avoided if the two shafts are connected with an intermediate shaft and two universal joints, provided the latter are properly arranged or located. Two conditions are necessary to obtain a constant speed ratio between the driving and driven shafts. First, the shafts must make the same angle with the intermediate shaft; second, the universal joint forks (assuming that the fork design is employed) on the intermediate shaft must be placed relatively so that when the plane of the fork at the left end coincides with the center lines of the intermediate shaft and the shaft attached to the left-hand coupling, the plane of the right-hand fork must also coincide with the center lines of the intermediate shaft and the shaft attached to the right-hand coupling; therefore the driving and the driven shafts may be placed in a variety of positions. One of the most common arrangements is with the driving and driven shafts parallel. The forks on the intermediate shafts should then be placed in the same plane.

This intermediate connecting shaft is frequently made telescoping, and then the driving and driven shafts can be moved independently of each other within certain limits in longitudinal and lateral directions. The telescoping intermediate shaft consists of a rod which enters a sleeve and is provided with a suitable spline, to prevent rotation between the rod and sleeve and permit a sliding movement. This arrangement is applied to various machine tools.

Knuckle Joints.—Movement at the joint between two rods may be provided by knuckle joints, for which typical proportions are seen in the table *Proportions of Knuckle Joints* that follows.

Friction Clutches.—Clutches which transmit motion from the driving to the driven member by the friction between the engaging surfaces are built in many different designs, although practically all of them can be classified under four general types, namely, conical clutches; radially expanding clutches; contracting-band clutches; and friction disk clutches in single and multiple types. There are many modifications of these general classes, some of which combine the features of different types. The proportions of various sizes of cone clutches are given in the table “Cast-iron Friction Clutches.” The multicone friction clutch is a further development of the cone clutch. Instead of having a single cone-shaped surface, there is a series of concentric conical rings which engage annular grooves formed by corresponding rings on the opposite clutch member. The internal-expanding type is provided with shoes which are forced outward against an enclosing drum by the action of levers connecting with a collar free to slide along the shaft. The engaging shoes are commonly lined with wood or other material to increase the coefficient of friction. Disk clutches are based on the principle of multiple-plane friction, and use alternating plates or disks so arranged that one set engages with an outside cylindrical case and the other set with the shaft. When these plates are pressed together by spring pressure, or by other means, motion is transmitted from the driving to the driven members connected to the clutch. Some disk clutches have a few rather heavy or thick plates and others a relatively large number of thinner plates. Clutches of the latter type are common in automobile transmissions. One set of disks may be of soft steel and the other set of phosphor-bronze, or some other combination may be employed. For instance, disks are sometimes provided with cork inserts, or one set or series of disks may be faced with a special friction material such as asbestos-wire fabric, as in “dry plate” clutches, the disks of which are not lubricated like the disks of a clutch having, for example, the steel and phosphor-bronze combination. It is common practice to hold the driving and driven members of friction clutches in engagement by means of spring pressure, although pneumatic or hydraulic pressure may be employed.

Proportions of Knuckle Joints



For sizes not given below:

a = 1.2 D	h = 2 D
b = 1.1 D	i = 0.5 D
c = 1.2 D	j = 0.25 D
e = 0.75 D	k = 0.5 D
f = 0.6 D	l = 1.5 D
g = 1.5 D	

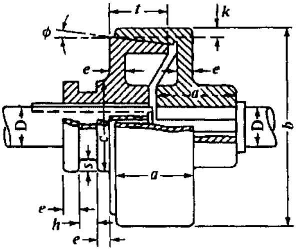
D	*a*	*b*	*c*	*e*	*f*	*g*	*h*	*i*	*j*	*k*	*l*
1/2	5/8	9/16	5/8	3/8	5/16	3/4	1	1/4	1/8	1/4	3/4
3/4	7/8	3/4	7/8	9/16	7/16	1 1/8	1 1/2	3/8	3/16	3/8	1 1/8
1	1 1/4	1 1/8	1 1/4	3/4	5/8	1 1/2	2	1/2	1/4	1/2	1 1/2
1 1/4	1 1/2	1 3/8	1 1/2	15/16	3/4	1 7/8	2 1/2	5/8	5/16	5/8	1 7/8
1 1/2	1 3/4	1 5/8	1 3/4	1 1/8	7/8	2 1/4	3	3/4	3/8	3/4	2 1/4
1 3/4	2 1/8	2	2 1/8	1 5/16	1 1/16	2 5/8	3 1/2	7/8	7/16	7/8	2 5/8
2	2 3/8	2 1/4	2 3/8	1 1/2	1 3/16	3	4	1	1/2	1	3
2 1/4	2 3/4	2 1/2	2 3/4	1 11/16	1 3/8	3 3/8	4 1/2	1 1/8	9/16	1 1/8	3 3/8
2 1/2	3	2 3/4	3	1 7/8	1 1/2	3 3/4	5	1 1/4	5/8	1 1/4	3 3/4
2 3/4	3 1/4	3	3 1/4	2 1/16	1 5/8	4 1/8	5 1/2	1 3/8	11/16	1 3/8	4 1/8
3	3 5/8	3 1/4	3 5/8	2 1/4	1 13/16	4 1/2	6	1 1/2	3/4	1 1/2	4 1/2
3 1/4	4	3 5/8	4	2 7/16	2	4 7/8	6 1/2	1 5/8	13/16	1 5/8	4 7/8
3 1/2	4 1/4	3 7/8	4 1/4	2 5/8	2 1/8	5 1/4	7	1 3/4	7/8	1 3/4	5 1/4
3 3/4	4 1/2	4 1/8	4 1/2	2 13/16	2 1/4	5 5/8	7 1/2	1 7/8	15/16	1 7/8	5 5/8
4	4 3/4	4 3/8	4 3/4	3	2 3/8	6	8	2	1	2	6
4 1/4	5 1/8	4 3/4	5 1/8	3 3/16	2 9/16	6 3/8	8 1/2	2 1/8	1 1/16	2 1/8	6 3/8
4 1/2	5 1/2	5	5 1/2	3 3/8	2 3/4	6 3/4	9	2 1/4	1 1/8	2 1/4	6 3/4
4 3/4	5 3/4	5 1/4	5 3/4	3 9/16	2 7/8	7 1/8	9 1/2	2 3/8	1 3/16	2 3/8	7 1/8
5	6	5 1/2	6	3 3/4	3	7 1/2	10	2 1/2	1 1/4	2 1/2	7 1/2

Power Transmitting Capacity of Friction Clutches.—When selecting a clutch for a given class of service, it is advisable to consider any overloads that may be encountered and base the power transmitting capacity of the clutch upon such overloads. When the load varies or is subject to frequent release or engagement, the clutch capacity should be greater than the actual amount of power transmitted. If the power is derived from a gas or gasoline engine, the horsepower rating of the clutch should be 75 or 100 per cent greater than that of the engine.

Power Transmitted by Disk Clutches.—The approximate amount of power that a disk clutch will transmit may be determined from the following formula, in which H = horsepower transmitted by the clutch; μ = coefficient of friction; r = mean radius of engaging surfaces; F = axial force in pounds (spring pressure) holding disks in contact; N = number of frictional surfaces; S = speed of shaft in revolutions per minute:

$$H = \frac{\mu r F N S}{63{,}000}$$

Cast-iron Friction Clutches



For sizes not given below:

$a = 2D$
$b = 4$ to $8D$
$c = 2\frac{1}{4}D$
$t = 1\frac{1}{2}D$
$e = \frac{3}{8}D$
$h = \frac{1}{2}D$
$s = \frac{5}{16}D$, nearly
$k = \frac{1}{4}D$

Note:— The angle φ of the cone may be from 4 to 10 degrees

D	*a*	*b*	*c*	*t*	*e*	*h*	*s*	*k*
1	2	4 – 8	2¼	1½	⅜	½	5/16	¼
1¼	2½	5–10	2⅞	1⅞	½	⅝	⅜	5/16
1½	3	6–12	3⅜	2¼	⅝	¾	½	⅜
1¾	3½	7–14	4	2⅝	⅝	⅞	⅝	7/16
2	4	8–16	4½	3	¾	1	⅝	½
2¼	4½	9–18	5	3⅜	⅞	1⅛	⅝	9/16
2½	5	10–20	5⅝	3¾	1	1¼	¾	⅝
2¾	5½	11–22	6¼	4⅛	1	1⅜	⅞	11/16
3	6	12–24	6¾	4½	1⅛	1½	⅞	¾
3¼	6½	13–26	7⅜	4⅞	1¼	1⅝	1	13/16
3½	7	14–28	7⅞	5¼	1⅜	1¾	1	⅞
3¾	7½	15–30	8½	5⅝	1⅜	1⅞	1¼	15/16
4	8	16–32	9	6	1½	2	1¼	1
4¼	8½	17–34	9½	6⅜	1⅝	2⅛	1⅜	1 1/16
4½	9	18–36	10¼	6¾	1¾	2¼	1⅜	1⅛
4¾	9½	19–38	10¾	7⅛	1¾	2⅜	1½	1 3/16
5	10	20–40	11¼	7½	1⅞	2½	1½	1¼
5¼	10½	21–42	11¾	7⅞	2	2⅝	1⅝	1 5/16
5½	11	22–44	12⅜	8¼	2	2¾	1¾	1⅜
5¾	11½	23–46	13	8⅝	2¼	2⅞	1¾	1 7/16
6	12	24–48	13½	9	2¼	3	1⅞	1½

Frictional Coefficients for Clutch Calculations.—While the frictional coefficients used by designers of clutches differ somewhat and depend upon variable factors, the following values may be used in clutch calculations: For greasy leather on cast iron about 0.20 or 0.25, leather on metal that is quite oily 0.15; metal and cork on oily metal 0.32; the same on dry metal 0.35; metal on dry metal 0.15; disk clutches having lubricated surfaces 0.10.

Formulas for Cone Clutches.—In cone clutch design, different formulas have been developed for determining the horsepower transmitted. These formulas, at first sight, do not seem to agree, there being a variation due to the fact that in some of the formulas the friction clutch surfaces are assumed to engage without slip, whereas, in others, some allowance is made for slip. The following formulas include both of these conditions:

H.P. = horsepower transmitted

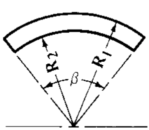
N = revolutions per minute
r = mean radius of friction cone, in inches
r_1 = large radius of friction cone, in inches
r_2 = small radius of friction cone, in inches
R_1 = outside radius of leather band, in inches
R_2 = inside radius of leather band, in inches
V = velocity of a point at distance r from the center, in feet per minute
F = tangential force acting at radius r, in pounds
P_n = total normal force between cone surfaces, in pounds
P_s = spring force, in pounds
α = angle of clutch surface with axis of shaft = 7 to 13 degrees
β = included angle of clutch leather, when developed, in degrees
f = coefficient of friction = 0.20 to 0.25 for greasy leather on iron
p = allowable pressure per square inch of leather band = 7 to 8 pounds
W = width of clutch leather, in inches

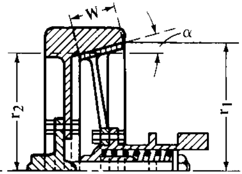
$$R_1 = \frac{r_1}{\sin\alpha} \qquad R_2 = \frac{r_2}{\sin\alpha}$$

$$\beta = \sin\alpha \times 360 \qquad r = \frac{r_1 + r_2}{2}$$

$$V = \frac{2\pi r N}{12}$$

DEVELOPMENT OF CLUTCH LEATHER





$$F = \frac{\text{HP} \times 33{,}000}{V} \qquad W = \frac{P_n}{2\pi r p} \qquad \text{HP} = \frac{P_n f r N}{63{,}025}$$

For engagement with some slip:

$$P_n = \frac{P_s}{\sin\alpha} \qquad P_s = \frac{\text{HP} \times 63{,}025 \sin\alpha}{frN}$$

For engagement without slip:

$$P_n = \frac{P_s}{\sin\alpha + f\cos\alpha} \qquad P_s = \frac{\text{HP} \times 63{,}025(\sin\alpha + f\cos\alpha)}{frN}$$

Angle of Cone.—If the angle of the conical surface of the cone type of clutch is too small, it may be difficult to release the clutch on account of the wedging effect, whereas, if the angle is too large, excessive spring force will be required to prevent slipping. The minimum angle for a leather-faced cone is about 8 or 9 degrees and the maximum angle about 13 degrees. An angle of 12 ½ degrees appears to be the most common and is generally considered good practice. These angles are given with relation to the clutch axis and are one-half the included angle.

Magnetic Clutches.—Many disk and other clutches are operated electromagnetically with the magnetic force used only to move the friction disk(s) and the clutch disk(s) into or out of engagement against spring or other pressure. On the other hand, in a magnetic particle clutch, transmission of power is accomplished by magnetizing a quantity of metal particles enclosed between the driving and the driven components. forming a bond between them. Such clutches can be controlled to provide either a rigid coupling or uniform slip, useful in wire drawing and manufacture of cables.

Another type of magnetic clutch uses eddy currents induced in the input member which interact with the field in the output rotor. Torque transmitted is proportional to the coil current, so precise control of torque is provided. A third type of magnetic clutch relies on the hysteresis loss between magnetic fields generated by a coil in an input drum and a close-fitting cup on the output shaft, to transmit torque. Torque transmitted with this type of clutch also is proportional to coil current, so close control is possible.

Permanent-magnet types of clutches also are available, in which the engagement force is exerted by permanent magnets when the electrical supply to the disengagement coils is cut off. These types of clutches have capacities up to five times the torque-to-weight ratio of spring-operated clutches. In addition, if the controls are so arranged as to permit the coil polarity to be reversed instead of being cut off, the combined permanent magnet and electromagnetic forces can transmit even greater torque.

Centrifugal and Free-wheeling Clutches.—Centrifugal clutches have driving members that expand outward to engage a surrounding drum when speed is sufficient to generate centrifugal force. Free-wheeling clutches are made in many different designs and use balls, cams or sprags, ratchets, and fluids to transmit motion from one member to the other. These types of clutches are designed to transmit torque in only one direction and to take up the drive with various degrees of gradualness up to instantaneously.

Slipping Clutch/Couplings.—Where high shock loads are likely to be experienced, a slipping clutch or coupling or both should be used. The most common design uses a clutch plate that is clamped between the driving and driven plates by spring pressure that can be adjusted. When excessive load causes the driven member to slow, the clutch plate surfaces slip, allowing reduction of the torque transmitted. When the overload is removed, the drive is taken up automatically. Switches can be provided to cut off current supply to the driving motor when the driven shaft slows to a preset limit or to signal a warning or both. The slip or overload torque is calculated by taking 150 per cent of the normal running torque.

Wrapped-spring Clutches.—For certain applications, a simple steel spring sized so that its internal diameter is a snug fit on both driving and driven shafts will transmit adequate torque in one direction. The tightness of grip of the spring on the shafts increases as the torque transmitted increases. Disengagement can be effected by slight rotation of the spring, through a projecting tang, using electrical or mechanical means, to wind up the spring to a larger internal diameter, allowing one of the shafts to run free within the spring.

Normal running torque T_r in lb-ft = (required horsepower × 5250) ÷ rpm. For heavy shock load applications, multiply by a 200 per cent or greater overload factor. (See Motors, factors governing selection.)

The clutch starting torque T_c, in lb-ft, required to accelerate a given inertia in a specific time is calculated from the formula:

$$T_c = \frac{WR^2 \times \Delta N}{308t}$$

where WR^2 = total inertia encountered by clutch in lb-ft^2 (W = weight and R = radius of gyration of rotating part)

ΔN = final rpm — initial rpm; 308 = constant (see Motors, factors governing selection)

t = time to required speed in seconds

Example: If the inertia is 80 lb-ft^2, and the speed of the driven shaft is to be increased from 0 to 1500 rpm in 3 seconds, find the clutch starting torque in lb-ft.

$$T_c = \frac{80 \times 1500}{308 \times 3} = 130 \text{ lb-ft}$$

The heat E, in BTU, generated in one engagement of a clutch can be calculated from the formula:

$$E = \frac{T_c \times WR^2 \times (N_1^2 - N_2^2)}{(T_c - T_1) \times 4.7 \times 10^6}$$

where: WR^2 = total inertia encountered by clutch in lb-ft.^{2}
N_1 = final rpm
N_2 = initial rpm
Tc = clutch torque in lb-ft
T_1 = torque load in lb-ft

Example: Calculate the heat generated for each engagement under the conditions cited for the first example.

$$E = \frac{130 \times 80 \times (1500)^2}{(130 - 10) \times 4.7 \times 10^6} = 41.5 \text{ BTU}$$

The preferred location for a clutch is on the high- rather than on the low-speed shaft because a smaller-capacity unit, of lower cost and with more rapid dissipation of heat, can be used. However, the heat generated may also be more because of the greater slippage at higher speeds, and the clutch may have a shorter life. For light-duty applications, such as to a machine tool, where cutting occurs after the spindle has reached operating speed, the calculated torque should be multiplied by a safety factor of 1.5 to arrive at the capacity of the clutch to be used. Heavy-duty applications such as frequent starting of a heavily loaded vibratory-finishing barrel require a safety factor of 3 or more.

Positive Clutches.—When the driving and driven members of a clutch are connected by the engagement of interlocking teeth or projecting lugs, the clutch is said to be “positive” to distinguish it from the type in which the power is transmitted by frictional contact. The positive clutch is employed when a sudden starting action is not objectionable and when the inertia of the driven parts is relatively small. The various forms of positive clutches differ merely in the angle or shape of the engaging surfaces. The least positive form is one having planes of engagement which incline backward, with respect to the direction of motion. The tendency of such a clutch is to disengage under load, in which case it must be held in position by axial pressure.

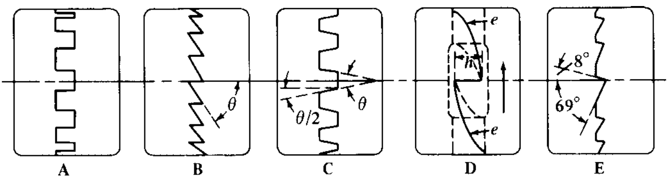


Fig. 1. Types of Clutch Teeth

This pressure may be regulated to perform normal duty, permitting the clutch to slip and disengage when over-loaded. Positive clutches, with the engaging planes parallel to the axis of rotation, are held together to obviate the tendency to jar out of engagement, but they provide no safety feature against over-load. So-called “under-cut” clutches engage more tightly the heavier the load, and are designed to be disengaged only when free from load. The teeth of positive clutches are made in a variety of forms, a few of the more common styles being shown in Fig. 1. Clutch *A* is a straight-toothed type, and *B* has angular or saw-shaped teeth. The driving member of the former can be rotated in either direction: the latter is adapted to the transmission of motion in one direction only, but is more readily engaged. The angle θ of the cutter for a saw-tooth clutch *B* is ordinarily 60 degrees. Clutch *C* is similar to *A*, except that the sides of the teeth are inclined to facilitate engagement and disengagement. Teeth of this shape are sometimes used when a clutch is required to run in either direction without backlash. Angle θ is varied to suit requirements and should not exceed 16 or 18 degrees. The straight-tooth clutch A is also modified to make the teeth engage more readily, by rounding the corners of the teeth at the top and bottom. Clutch *D* (commonly called a "spiral-jaw" clutch) differs from *B* in that the surfaces *e* are helicoidal. The driving member of this clutch can transmit motion in only one direction.

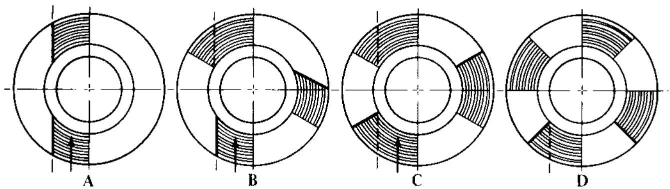


Fig. 2. Diagrammatic View Showing Method of Cutting Clutch Teeth

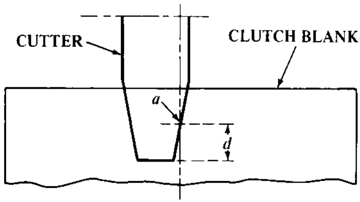


Fig. 3.

Clutches of this type are known as right- and left-hand, the former driving when turning to the right, as indicated by the arrow in the illustration. Clutch *E* is the form used on the back-shaft of the Brown & Sharpe automatic screw machines. The faces of the teeth are radial and incline at an angle of 8 degrees with the axis, so that the clutch can readily be disengaged. This type of clutch is easily operated, with little jar or noise. The 2-inch diameter size has 10 teeth. Height of working face, $\frac{1}{8}$ inch.

Cutting Clutch Teeth.—A common method of cutting a straight-tooth clutch is indicated by the diagrams *A*, *B* and *C*, Fig. 2, which show the first, second and third cuts required for forming the three teeth. The work is held in the chuck of a dividing-head, the latter being set at right angles to the table. A plain milling cutter may be used (unless the corners of the teeth are rounded), the side of the cutter being set to exactly coincide with the center-line. When the number of teeth in the clutch is odd, the cut can be taken clear across the blank as shown, thus finishing the sides of two teeth with one passage of the cutter. When the number of teeth is even, as at *D*, it is necessary to mill all the teeth on one side and then set the cutter for finishing the opposite side. Therefore, clutches of this type commonly have an odd number of teeth. The maximum width of the cutter depends upon the width of the space at the narrow ends of the teeth. If the cutter must be quite narrow in order to pass the narrow ends, some stock may be left in the tooth spaces, which must be removed by a separate cut. If the tooth is of the modified form shown at *C*, Fig. 1, the cutter should be set as indicated in Fig. 3; that is, so that a point a on the cutter at a radial distance *d* equal to one-half the depth of the clutch teeth lies in a radial plane. When it is important to eliminate all backlash, point *a* is sometimes located at a radial distance *d* equal to six-tenths of the depth of the tooth, in order to leave clearance spaces at the bottoms of the teeth; the two clutch members will then fit together tightly. Clutches of this type must be held in mesh.

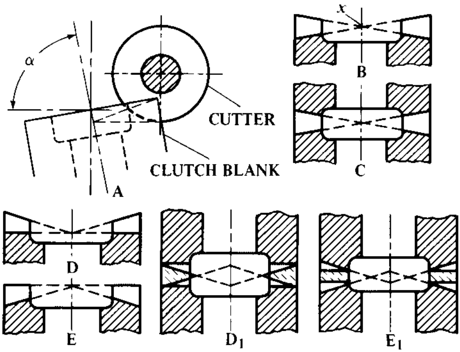
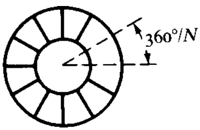


Fig. 4.

Angle of Dividing-head for Milling V-shaped Teeth with Single-angle Cutter



$$\cos\alpha = \frac{\tan(360°/N) \times \cot\theta}{2}$$

α is the angle shown in Fig. 4 and is the angle shown by the graduations on the dividing head. θ is the included angle of a single cutter, see Fig. 1.

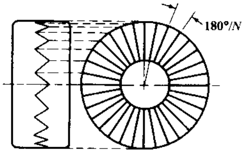
No. of Teeth, *N*	Angle of Single-angle Cutter, θ						No. of Teeth, *N*	Angle of Single-angle Cutter, θ					
	60°		70°		80°			60°		70°		80°	
	Dividing Head Angle, α							Dividing Head Angle, α					
5	27°	19.2′	55°	56.3′	74°	15.4′	18	83°	58.1′	86°	12.1′	88°	9.67′
6	60		71	37.6	81	13	19	84	18.8	86	25.1	88	15.9
7	68	46.7	76	48.5	83	39.2	20	84	37.1	86	36.6	88	21.5
8	73	13.3	79	30.9	84	56.5	21	84	53.5	86	46.9	88	26.5
9	75	58.9	81	13	85	45.4	22	85	8.26	86	56.2	88	31
10	77	53.6	82	24.1	86	19.6	23	85	21.6	87	4.63	88	35.1
11	79	18.5	83	17	86	45.1	24	85	33.8	87	12.3	88	38.8
12	80	24.4	83	58.1	87	4.94	25	85	45	87	19.3	88	42.2
13	81	17.1	84	31.1	87	20.9	26	85	55.2	87	25.7	88	45.3
14	82	.536	84	58.3	87	34	27	86	4.61	87	31.7	88	48.2
15	82	36.9	85	21.2	87	45	28	86	13.3	87	37.2	88	50.8
16	83	7.95	85	40.6	87	54.4	29	86	21.4	87	42.3	88	53.3
17	83	34.7	85	57.4	88	2.56	30	86	28.9	87	47	88	55.6

Cutting Saw-tooth Clutches: When milling clutches having angular teeth as shown at *B*, Fig. 1, the axis of the clutch blank should be inclined a certain angle α as shown at *A* in Fig. 4. If the teeth were milled with the blank vertical, the tops of the teeth would incline towards the center as at *D*, whereas, if the blank were set to such an angle that the tops of the teeth were square with the axis, the bottoms would incline upwards as at *E*. In either case, the two clutch members would not mesh completely: the engagement of the teeth cut as shown at *D* and *E* would be as indicated at D_1 and E_1 respectively. As will be seen, when the outer points of the teeth at D_1 are at the bottom of the grooves in the opposite member, the inner ends are not together, the contact area being represented by the dotted lines. At E_1 the inner ends of the teeth strike first and spaces are left between the teeth around the outside of the clutch. To overcome this objectionable feature, the clutch teeth should be cut as indicated at *B*, or so that the bottoms and tops of the teeth have the same inclination, converging at a central point *x*. The teeth of both members will then engage across the entire width as shown at *C*. The angle α required for cutting a clutch as at *B* can be determined by the following formula in which α equals the required angle, *N* = number of teeth, θ = cutter angle, and 360°/*N* = angle between teeth:

$$\cos\alpha = \frac{\tan(360°/N) \times \cot\theta}{2}$$

The angles α for various numbers of teeth and for 60-, 70- or 80-degree single-angle cutters are given in the table on page 2335. The following table is for double-angle cutters used to cut V-shaped teeth.

Angle of Dividing-head for Milling V-shaped Teeth with Double-angle Cutter



$$\cos\alpha = \frac{\tan(180°/N) \times \cot(\theta/2)}{2}$$

This is the angle (α, Fig. 4) shown by graduations on the dividing-head. θ is the included angle of a double-angle cutter, see Fig. 1.

No. of Teeth, *N*	Included Angle of Cutter, θ				No. of Teeth, *N*	Included Angle of Cutter, θ			
	60°		90°			60°		90°	
	Dividing Head Angle, α					Dividing Head Angle, α			
10	73°	39.4′	80°	39′	31	84°	56.9′	87°	5.13′
11	75	16.1	81	33.5	32	85	6.42	87	10.6
12	76	34.9	82	18	33	85	15.4	87	15.8
13	77	40.5	82	55.3	34	85	23.8	87	20.7
14	78	36	83	26.8	35	85	31.8	87	25.2
15	79	23.6	83	54	36	85	39.3	87	29.6
16	80	4.83	84	17.5	37	85	46.4	87	33.7
17	80	41	84	38.2	38	85	53.1	87	37.5
18	81	13	84	56.5	39	85	59.5	87	41.2
19	81	41.5	85	12.8	40	86	5.51	87	44.7
20	82	6.97	85	27.5	41	86	11.3	87	48
21	82	30	85	40.7	42	86	16.7	87	51.2
22	82	50.8	85	52.6	43	86	22	87	54.2
23	83	9.82	86	3.56	44	86	26.9	87	57
24	83	27.2	86	13.5	45	86	31.7	87	59.8
25	83	43.1	86	22.7	46	86	36.2	88	2.4
26	83	57.8	26	31.2	47	86	40.6	88	4.91
27	84	11.4	86	39	48	86	44.8	88	7.32
28	84	24	86	46.2	49	86	48.8	88	9.63
29	84	35.7	86	53	50	86	52.6	88	11.8
30	84	46.7	86	59.3	51	86	56.3	88	14

The angles given in the table above are applicable to the milling of V-shaped grooves in brackets, etc., which must have toothed surfaces to prevent the two members from turning relative to each other, except when unclamped for angular adjustment

FRICTION BRAKES

Formulas for Band Brakes.—In any band brake, such as shown in Fig. 1, in the tabulation of formulas, where the brake wheel rotates in a clockwise direction, the tension in that part of the band marked x equals $P\frac{1}{e^{\mu\theta}-1}$

The tension in that part marked y equals $P\frac{e^{\mu\theta}}{e^{\mu\theta}-1}$.

P = tangential force in pounds at rim of brake wheel

e = base of natural logarithms = 2.71828

μ = coefficient of friction between the brake band and the brake wheel

θ = angle of contact of the brake band with the brake wheel expressed in

$$\text{radians}\left(\text{one radian} = \frac{180\text{ deg.}}{\pi\text{ radians}} = 57.296\frac{\text{deg.}}{\text{radian}}\right).$$

For simplicity in the formulas presented, the tensions at x and y (Fig. 1) are denoted by T_1 and T_2 respectively, for clockwise rotation. When the direction of the rotation is reversed, the tension in x equals T_2, and the tension in y equals T_1, which is the reverse of the tension in the clockwise direction.

The value of the expression $e^{\mu\theta}$ in these formulas may be most easily found by using a hand-held calculator of the scientific type; that is, one capable of raising 2.71828 to the power $\mu\theta$. The following example outlines the steps in the calculations.

Table of Values of $e^{\mu\theta}$

Proportion of Contact to Whole Circumference, $\frac{\theta}{2\pi}$	Steel Band on Cast Iron, μ = 0.18	Leather Belt on				
		Wood	Cast Iron			
		Slightly Greasy; μ = 0.47	Very Greasy; μ = 0.12	Slightly Greasy; μ = 0.28	Damp; μ = 0.38
0.1	1.12	1.34	1.08	1.19	1.27
0.2	1.25	1.81	1.16	1.42	1.61
0.3	1.40	2.43	1.25	1.69	2.05
0.4	1.57	3.26	1.35	2.02	2.60
0.425	1.62	3.51	1.38	2.11	2.76
0.45	1.66	3.78	1.40	2.21	2.93
0.475	1.71	4.07	1.43	2.31	3.11
0.5	1.76	4.38	1.46	2.41	3.30
0.525	1.81	4.71	1.49	2.52	3.50
0.55	1.86	5.07	1.51	2.63	3.72
0.6	1.97	5.88	1.57	2.81	4.19
0.7	2.21	7.90	1.66	3.43	5.32
0.8	2.47	10.60	1.83	4.09	6.75
0.9	2.77	14.30	1.97	4.87	8.57
1.0	3.10	19.20	2.12	5.81	10.90

Formulas for Simple and Differential Band Brakes

F = force in pounds at end of brake handle; P = tangential force in pounds at rim of brake wheel; e = base of natural logarithms = 2.71828; μ = coefficient of friction between the brake band and the brake wheel; θ = angle of contact of the brake band with the brake wheel, expressed in radians (one radian = 57.296 degrees).

$$T_1 = P\frac{1}{e^{\mu\theta}-1} \qquad T_2 = P\frac{e^{\mu\theta}}{e^{\mu\theta}-1}$$

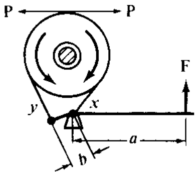


Fig. 1.

Simple band brake.
For clockwise rotation:

$$F = \frac{bT_2}{a} = \frac{Pb}{a}\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right)$$

For counter clockwise rotation:

$$F = \frac{bT_1}{a} = \frac{Pb}{a}\left(\frac{1}{e^{\mu\theta}-1}\right)$$

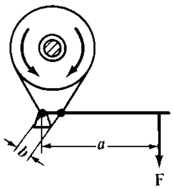


Fig. 2.

Simple band brake.
For clockwise rotation:

$$F = \frac{bT_1}{a} = \frac{Pb}{a}\left(\frac{1}{e^{\mu\theta}-1}\right)$$

For counter clockwise rotation:

$$F = \frac{bT_2}{a} = \frac{Pb}{a}\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right)$$

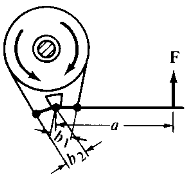


Fig. 3.

Differential band brake.
For clockwise rotation:

$$F = \frac{b_2T_2 - b_1T_1}{a} = \frac{P}{a}\left(\frac{b_2e^{\mu\theta}-b_1}{e^{\mu\theta}-1}\right)$$

For counter clockwise rotation:

$$F = \frac{b_2T_1 - b_1T_2}{a} = \frac{P}{a}\left(\frac{b_2-b_1e^{\mu\theta}}{e^{\mu\theta}-1}\right)$$

In this case, if b_2 is equal to, or less than, $b_1e^{\mu\theta}$, the force F will be 0 or negative and the band brake works automatically.

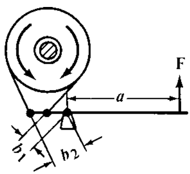


Fig. 4.

Differential band brake.
For clockwise rotation:

$$F = \frac{b_2T_2 + b_1T_1}{a} = \frac{P}{a}\left(\frac{b_2e^{\mu\theta}+b_1}{e^{\mu\theta}-1}\right)$$

For counter clockwise rotation:

$$F = \frac{b_1T_2 + b_2T_1}{a} = \frac{P}{a}\left(\frac{b_1e^{\mu\theta}+b_2}{e^{\mu\theta}-1}\right)$$

If $b_2 = b_1$, both of the above formulas reduce to $F = \frac{Pb_1}{a}\left(\frac{e^{\mu\theta}+1}{e^{\mu\theta}-1}\right)$.

In this case, the same force F is required for rotation in either direction.

In a band brake of the type in Fig. 1, dimension a = 24 inches, and b = 4 inches; force P = 100 pounds; coefficient μ = 0.2, and angle of contact = 240 degrees, or

$$\theta = \frac{240}{180} \times \pi = 4.18$$

The rotation is clockwise. Find force F required.

$$\begin{aligned}F &= \frac{Pb}{a}\left(\frac{e^{\mu\theta}}{e^{\mu\theta}-1}\right)\\ &= \frac{100\times 4}{24}\left(\frac{2.71828^{0.2\times 4.18}}{2.71828^{0.2\times 4.18}-1}\right)\\ &= \frac{400}{24}\times\frac{2.71828^{0.836}}{2.71828^{0.836}-1}\\ &= 16.66\times\frac{2.31}{2.31-1} = 29.4\end{aligned}$$

If a hand-held calculator is not used, determining the value of $e^{\mu\theta}$ is rather tedious, and the table on page 2337 will save calculations.

Coefficient of Friction in Brakes.—The coefficients of friction that may be assumed for friction brake calculations are as follows: Iron on iron, 0.25 to 0.3 leather on iron, 0.3; cork on iron, 0.35. Values somewhat lower than these should be assumed when the velocities exceed 400 feet per minute at the beginning of the braking operation.

For brakes where wooden brake blocks are used on iron drums, poplar has proved the best brake-block material. The best material for the brake drum is wrought iron. Poplar gives a high coefficient of friction, and is little affected by oil. The average coefficient of friction for poplar brake blocks and wrought-iron drums is 0.6; for poplar on cast iron, 0.35 for oak on wrought iron, 0.5; for oak on cast iron, 0.3; for beech on wrought iron, 0.5; for beech on cast iron, 0.3; for elm on wrought iron, 0.6; and for elm on cast iron, 0.35. The objection to elm is that the friction decreases rapidly if the friction surfaces are oily. The coefficient of friction for elm and wrought iron, if oily, is less than 0.4.

Calculating Horsepower from Dynamometer Tests.—When a dynamometer is arranged for measuring the horsepower transmitted by a shaft, as indicated by the diagrammatic view in the illustration on page 2340, the horsepower may be obtained by the formula:

$$HP = \frac{2\pi LPN}{33{,}000}$$

in which H.P. = horsepower transmitted; N = number of revolutions per minute; L = distance (as shown in illustration) from center of pulley to point of action of weight P, in feet; P = weight hung on brake arm or read on scale.

By adopting a length of brake arm equal to 5 feet 3 inches, the formula may be reduced to the simple form:

$$HP = \frac{NP}{1000}$$

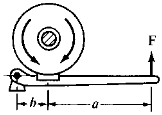
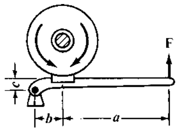
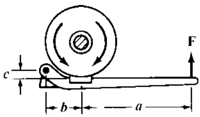
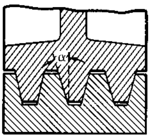
If a length of brake arm equal to 2 feet 7½ inches is adopted as a standard, the formula takes the form:

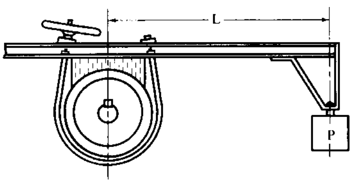
$$HP = \frac{NP}{2000}$$

The *transmission* type of dynamometer measures the power by transmitting it through the mechanism of the dynamometer from the apparatus in which it is generated, or to the apparatus in which it is to be utilized. Dynamometers known as *indicators* operate by simultaneously measuring the pressure and volume of a confined fluid. This type may be used for the measurement of the power generated by steam or gas engines or absorbed by refrigerating machinery, air compressors, or pumps. An electrical dynamometer is for measuring the power of an electric current, based on the mutual action of currents flowing in two coils. It consists principally of one fixed and one movable coil, which, in the normal position, are at right angles to each other. Both coils are connected in series, and, when a current traverses the coils, the fields produced are at right angles; hence, the coils tend to take up a parallel position. The movable coil with an attached pointer will be deflected, the deflection measuring directly the electric current.

Formulas for Block Brakes

F = force in pounds at end of brake handle;
P = tangential force in pounds at rim of brake wheel;
μ = coefficient of friction between the brake block and brake wheel.

 Fig. 1.	Block brake. For rotation in either direction: $F = P\frac{b}{a+b}\times\frac{1}{\mu} = \frac{Pb}{a+b}\left(\frac{1}{\mu}\right)$
 Fig. 2.	Block brake. For clockwise rotation: $F = \frac{\frac{Pb}{\mu} - Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} - \frac{c}{b}\right)$ For counter clockwise rotation: $F = \frac{\frac{Pb}{\mu} + Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} + \frac{c}{b}\right)$
 Fig. 3.	Block brake. For clockwise rotation: $F = \frac{\frac{Pb}{\mu} + Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} + \frac{c}{b}\right)$ For counter clockwise rotation: $F = \frac{\frac{Pb}{\mu} - Pc}{a+b} = \frac{Pb}{a+b}\left(\frac{1}{\mu} - \frac{c}{b}\right)$
 Fig. 4.	The brake wheel and friction block of the block brake are often grooved as shown in Fig. 4. In this case, substitute for μ in the above equations the value $\frac{\mu}{\sin\alpha + \mu\cos\alpha}$ where α is one-half the angle included by the facts of the grooves.



Friction Wheels for Power Transmission

When a rotating member is driven intermittently and the rate of driving does not need to be positive, friction wheels are frequently used, especially when the amount of power to be transmitted is comparatively small. The driven wheels in a pair of friction disks should always be made of a harder material than the driving wheels, so that if the driven wheel should be held stationary by the load, while the driving wheel revolves under its own pressure, a flat spot may not be rapidly worn on the driven wheel. The driven wheels, therefore, are usually made of iron, while the driving wheels are made of or covered with, rubber, paper, leather, wood or fiber. The safe working force per inch of face width of contact for various materials are as follows: Straw fiber, 150; leather fiber, 240; tarred fiber, 240; leather, 150; wood, 100 to 150; paper, 150. Coefficients of friction for different combinations of materials are given in the following table. Smaller values should be used for exceptionally high speeds, or when the transmission must be started while under load.

Horsepower of Friction Wheels.—Let D = diameter of friction wheel in inches; N = Number of revolutions per minute; W = width of face in inches; f = coefficient of friction; P = force in pounds, per inch width of face. Then:

$$\text{H.P.} = \frac{3.1416 \times D \times N \times P \times W \times f}{33{,}000 \times 12}$$

Assume

$$\frac{3.1416 \times P \times f}{33{,}000 \times 12} = C$$

then,

for $P = 100$ and $f = 0.20$, $C = 0.00016$

for $P = 150$ and $f = 0.20$, $C = 0.00024$

for $P = 200$ and $f = 0.20$, $C = 0.00032$

Working Values of Coefficient of Friction

Materials	Coefficient of Friction	Materials	Coefficient of Friction
Straw fiber and cast iron	0.26	Tarred fiber and aluminum	0.18
Straw fiber and aluminum	0.27	Leather and cast iron	0.14
Leather fiber and cast iron	0.31	Leather and aluminum	0.22
Leather fiber and aluminum	0.30	Leather and typemetal	0.25
Tarred fiber and cast iron	0.15	Wood and metal	0.25
Paper and cast iron	0.20		

The horsepower transmitted is then:

$$\text{HP} = D \times N \times W \times C$$

Example: Find the horsepower transmitted by a pair of friction wheels; the diameter of the driving wheel is 10 inches, and it revolves at 200 revolutions per minute. The width of the wheel is 2 inches. The force per inch width of face is 150 pounds, and the coefficient of friction 0.20.

$$\text{HP} = 10 \times 200 \times 2 \times 0.00024 = 0.96 \text{ horsepower}$$

Horsepower Which May be Transmitted by Means of a Clean Paper Friction Wheel of One-inch Face when Run Under a Force of 150 Pounds (Rockwood Mfg. Co.)

Dia. of Friction Wheel	Revolutions per Minute										
	25	50	75	100	150	200	300	400	600	800	1000
4	0.023	0.047	0.071	0.095	0.142	0.190	0.285	0.380	0.571	0.76	0.95
6	0.035	0.071	0.107	0.142	0.214	0.285	0.428	0.571	0.856	1.14	1.42
8	0.047	0.095	0.142	0.190	0.285	0.380	0.571	0.761	1.142	1.52	1.90
10	0.059	0.119	0.178	0.238	0.357	0.476	0.714	0.952	1.428	1.90	2.38
14	0.083	0.166	0.249	0.333	0.499	0.666	0.999	1.332	1.999	2.66	3.33
16	0.095	0.190	0.285	0.380	0.571	0.761	1.142	1.523	2.284	3.04	3.80
18	0.107	0.214	0.321	0.428	0.642	0.856	1.285	1.713	2.570	3.42	4.28
24	0.142	0.285	0.428	0.571	0.856	1.142	1.713	2.284	3.427	4.56	5.71
30	0.178	0.357	0.535	0.714	1.071	1.428	2.142	2.856	4.284	5.71	7.14
36	0.214	0.428	0.642	0.856	1.285	1.713	2.570	3.427	5.140	6.85	8.56
42	0.249	0.499	0.749	0.999	1.499	1.999	2.998	3.998	5.997	7.99	9.99
48	0.285	0.571	0.856	1.142	1.713	2.284	3.427	4.569	6.854	9.13	11.42
50	0.297	0.595	0.892	1.190	1.785	2.380	3.570	4.760	7.140	9.52	11.90