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Marks' Standard Handbook for Mechanical Engineers

Revised by a staff of specialists

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d the average pressure, lb/in^2 , on the projected area of the bearings, where P is the load, lb , l the length of bushing, in, Z the absolute viscosity of the oil in centipoises (see p. 3-35). Approximate values of Z at 160 (130°F) are as follows: light machine oil, 30 (16); medium machine oil, 60 (25); medium-heavy machine oil, 120 (40); heavy machine oil, 160 (60).

For purposes of design of ordinary machinery with bearing pressures from 50 to 300 lb/in^2 (344.7 to 2,068 kN/m^2) and speeds of 100 to 3,000 rpm, values for the coefficient of journal friction can be taken from 0.008 to 0.020.

Thrust Bearings

Frictional Resistance Step bearings or pivots may be used to resist the end thrust of shafts. Let L = total load, lb in the direction of the shaft axis; dA = an elementary area of the thrust-bearing surface, in^2 ; y = distance of the area dA from its axis of revolution, in; p = pressure on dA due to load L , lb/in^2 ; and f = coefficient of sliding friction. Then, **moment of thrust friction** = $M = \int p f y dA$ in $\text{in}\cdot\text{lb}$, and the **work expended in friction** per min at n r/min = $W_f = 2\pi M n$ in $\text{in}\cdot\text{lb}$.

For a ring-shaped flat step bearing such as that shown in Fig. 12 (or a collar bearing), $M = \frac{1}{4} f L (\frac{1}{2} D^3 - \frac{1}{2} d^3)$, where D and d are in in. For a flat circular step bearing, $d = 0$, and $M = \frac{1}{8} f L D^3$.

The value of the coefficient of sliding friction is 0.08 to 0.15 when the speed of rotation is very slow. At higher velocities when a collar or step bearing is used, $f = 0.04$ to 0.06. If the design provides for the formation of a load-carrying oil film, as in the case of the Kingsbury thrust bearing, the coefficient of friction has values $f = 0.001$ to 0.0025.

Where oil is supplied from an external pump with such pressure as to separate the surfaces and provide an oil film of thickness b (Fig. 12), the frictional moment is

$$M = \frac{Z \times n (D^4 - d^4)}{67 \times 10^7 \times b}$$

where b is the film thickness, in, Z is viscosity of lubricant in centipoises, and n is rotation speed, r/min. With this kind of lubrication the frictional moment depends upon the speed of rotation of the shaft and actually approaches zero for zero shaft

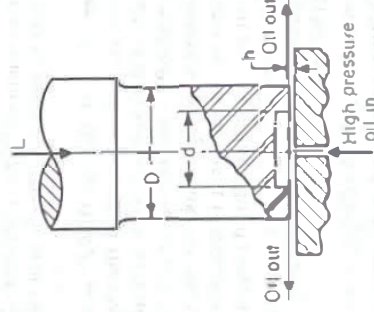


Fig. 12

speeds. The thrust load will be carried on a film of oil regardless of shaft rotation for as long as the pump continues to supply the required volume and pressure (see also Sec. 8).

EXAMPLE A hydrostatic thrust bearing carries 101,000 lb , D is 16

in, d is 10 in, oil-film thickness b is 0.006 in, oil viscosity Z , 30 centipoises at operating temperature, and n is 750 r/min. Substituting these values, the frictional torque M is 310 $\text{in}\cdot\text{lb}$ (358 $\text{cm}\cdot\text{kg}$). The oil supply pressure was 82.5 lb/in^2 (569 kN/m^2); the oil flow, 12.2 gal/min (46.2 liters).

Frictional Forces in Pin Joints of Mechanisms

In the absence of friction, or when the effect of friction is negligible, the force transmitted by the link b from the driver a to the driven link c (Figs. 13 and 14) acts through the centerline OO of the pins connecting the link b with links a and c . With friction, this line of action shifts to the line AA , tangent to small circles of diameter d . The diameter d of the circle, called the **friction circle**, for each individual joint, is equal to fD , where D is the diameter of the pin and f is the coefficient of friction between the pin and the link. The choice of the proper disposition of the tangent AA with respect to the two friction circles is dictated by the consideration that friction always opposes the action of the linkage. The force f opposes the motion of a ; therefore, with friction it acts on a longer lever than without friction (Figs. 13 and 14). On the other hand, the force F drives the link c ; friction hinders its

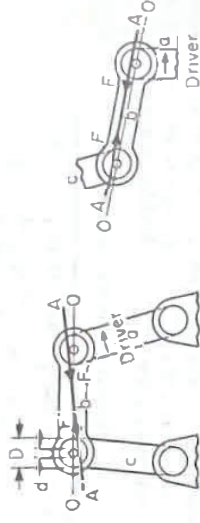


Fig. 14

action, and the equivalent lever is shorter with friction than without friction; the friction throws the line of action toward the center of rotation of link c .

EXAMPLE. An engine eccentric (Fig. 15) is a joint where the friction loss may be large. For the dimensions shown and with a torque of 250 $\text{in}\cdot\text{lb}$ applied to the rotating shaft, the resultant horizontal force, with no friction, will act through the center of the eccentric and be $250/(2.5 \sin 60)$ or 115.5 lb . With friction coefficient 0.1, the resultant force (which for a long rod remains approximately horizontal) will be tangent to the friction circle of radius 0.1×5 , or 0.5 in, and have a magnitude of $250/(2.5 \sin 60 + 0.5)$, or 93.8 lb (42.6 kg).

Tension Elements

Frictional Resistance In Fig. 16, let T_1 and T_2 be the tensions with which a rope, belt, chain, or brake band is strained over a drum, pulley, or sheave, and let the rope or

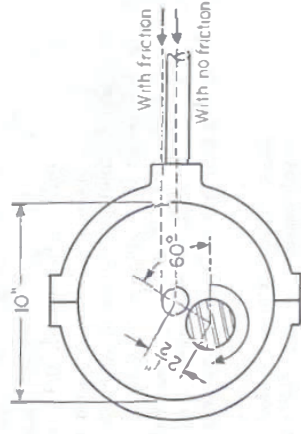


Fig. 15

Table 3. Values of e^{μ}

$\frac{e^{\mu}}{360^{\circ}}$	f									
	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5	
0.1	1.06	1.1	1.13	1.17	1.21	1.25	1.29	1.33	1.37	
0.2	1.13	1.21	1.29	1.37	1.46	1.55	1.65	1.76	1.87	
0.3	1.21	1.32	1.45	1.60	1.76	1.93	2.13	2.34	2.57	
0.4	1.29	1.46	1.65	1.87	2.12	2.41	2.73	3.10	3.51	
0.425	1.31	1.49	1.70	1.95	2.23	2.55	2.91	3.33	3.80	
0.45	1.33	1.53	1.76	2.03	2.34	2.69	3.10	3.57	4.11	
0.475	1.35	1.56	1.82	2.11	2.45	2.84	3.30	3.83	4.45	
0.5	1.37	1.60	1.87	2.19	2.57	3.00	3.51	4.11	4.81	
0.525	1.39	1.64	1.93	2.28	2.69	3.17	3.74	4.41	5.20	
0.55	1.41	1.68	2.00	2.37	2.82	3.35	3.98	4.74	5.63	
0.6	1.46	1.76	2.13	2.57	3.10	3.74	4.52	5.45	6.59	
0.7	1.55	1.93	2.41	3.00	3.74	4.66	5.81	7.24	9.02	
0.8	1.65	2.13	2.73	3.51	4.52	5.81	7.47	9.60	12.35	
0.9	1.76	2.34	3.10	4.11	5.45	7.24	9.60	12.74	16.90	
1.0	1.87	2.57	3.51	4.81	6.59	9.02	12.35	16.90	23.14	
1.5	2.57	4.11	6.59	10.55	16.90	27.08	43.38	69.49	111.32	
2.0	3.51	6.59	12.35	23.14	43.38	81.31	152.40	285.68	535.49	
2.5	4.81	10.55	23.14	50.75	111.32	244.15	535.49	1,174.5	2,575.9	
3.0	6.59	16.90	43.38	111.32	285.68	733.14	1,881.5	4,828.5	12,391	
3.5	9.02	27.08	81.31	244.15	733.14	2,199.90	6,610.7	19,851	59,608	
4.0	12.35	43.38	152.40	535.49	1,881.5	6,610.7	23,227	81,610	286,744	

$$e^{\mu} = 23.1407. \quad \log e^{\mu} = 1.3643764.$$

belt be on the point of slipping from T_2 toward T_1 by reason of the difference of tension $T_1 - T_2$. Then $T_1 - T_2 =$ equal circumferential force P transferred by friction must be equal to the frictional resistance W of the belt, rope, or band on the drum or pulley. Also, let $\alpha =$ angle subtending the arc of contact between the drum and tension element, measured in radians. Then, disregarding centrifugal forces,

$$T_1 = T_2 e^{f\alpha} \quad \text{and} \quad P = (e^{\mu} - 1)T_1 / e^{\mu} = (e^{\mu} - 1)T_2 = W$$

where $e =$ base of the napierian system of logarithms $= 2.718+$.

f is the coefficient of friction of repose (f_0) when there is no slip of the belt or band on the drum and the coefficient of sliding friction (f) when slip takes place.

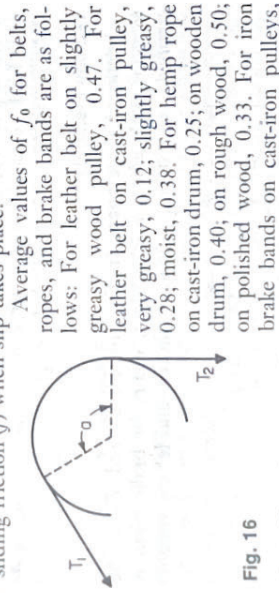


Fig. 16

0.18. For wire ropes, Tichinsky reports coefficients of static friction, f_0 , for a $\frac{5}{8}$ rope (8×19) on a worn-in cast-iron groove: 0.113 (dry); 0.124 (with fuller's earth); and 0.104 (when lubricated with Texaco Crater compound No. 1).

Effect of Friction and Stiffness in Tension Elements Let d

Rope diam., in (cm)	$\frac{5}{8}$ (1.59)
Fixed sheaves:	$e = 0.95-0.96$
Floating sheaves:	$e = 0.97$

$=$ the rope diameter, the diameter of the chain stock, or the diameter of the pin in link chains, and $R =$ the radius of the pitch circle in which the tension element travels, both in inches. The internal friction rigidity of the tension element causes a shortening at the driving end T_1 of the lever arm R an amount b_1 in inches, and at the following end a lengthening of the lever arm R an amount $= b_2$ in. Then, for simultaneous winding on and off, $T_1(R - b_1) = T_2(R + b_2)$. Approximately, $b_1 = b_2 = b$, whence $b_1 + b_2 = 2b$ (approx) and $T_1 = [1 + (2b/R)]T_2$.

This results in a frictional loss, and the efficiency of transmission is $e = 1 - (2b/R)$. If the tension element is only wound on the drum, $b_1 = 0$ and $T_1 = [1 + (b/R)]T_2$, with the corresponding efficiency $e = 1 - b/R$.

For chains the coefficient of friction between the link faces or pivots is $f = 0.2$ to 0.3 ; when $d =$ the diam of the link pin, $b = fdl/2$.

For hemp ropes, $b = 0.03d^2$ to $0.09d^2$ according to the construction, material, and condition of the rope.

The elastic rigidity of the material is not a factor in simultaneous winding on and off, since the lever arm R is increased equally at the points of winding on and off; the work expended in bending the tension element as it is wound on is recovered as it straightens out in unwinding. But if there is only winding on, the effect of the bending is to be taken into account.

Efficiency of rope and chain sheaves at low speeds, including journal friction (180° contact):

For fixed sheaves, chain, and wire rope, $e = 0.94$ to 0.96 .
For floating sheaves, chain, and wire rope, $e = 0.97$.

Hemp rope sheaves:

1 (2.54)	$1\frac{1}{2}$ (3.81)	$1\frac{3}{4}$ (4.44)	2 (3.08)
0.91-0.96	0.89-0.93	0.84-0.92	0.85-0.91
0.96	0.95	0.94	0.93