Large Scale Model Railway Engineering

Many times in the past year or so we have talked about friction either how to eliminate it or how to increase it. In the case of a bearing we would like to minimize it and perhaps in the case of a drive wheel, maximize it to increase our pulling power. This month I thought I would talk about it in a little more detail.

Friction is the resistance to motion which takes place when one surface is moved upon another. The value of this friction force is usually determined by multiplying the normal force between the two surfaces by a value called the coefficient of friction. The coefficient of friction can be defined as follows.

If a body is placed on an inclined plane, the friction between the body and the plane will prevent it from sliding down the inclined surface, provided the angle of the plane with the horizontal is not too great. There will be a certain angle at which the body will just barely be able to remain stationary, the frictional resistance being very nearly overcome by the tendency of the body to slide down. This angle is termed the angle of repose, and the tangent of this angle is the coefficient of friction.

Some general rules can be formulated when dealing with friction that holds true most of the time. Some of the more important ones that apply to non lubricated surfaces are as follows:

1) For lower pressures the friction is directly proportional to the normal pressure between the two surfaces. As the pressure increases to a high value, the friction increase at a rapid rate until seizing takes place.

2) The friction is independent of the area in contact, so long as the total pressure stays the same.

3) At very low velocities the friction is independent of the velocity of rubbing. As the velocity increases, the friction decreases.

For well lubricated surfaces, the laws of friction are considerably different from those of dry or poorly lubricated surfaces.

1) The frictional resistance is almost independent of the pressure per square inch if the surfaces are flooded with oil.

2) The friction varies directly as the speed, at low pressure; but for high pressures the friction is very great at low velocities, approaching a minimum at about two feet per second linear velocity, and afterwards

increasing approximately as the square root of the speed.

3) For well lubricated surfaces the frictional resistance depends, to a very great extent, on the temperature.

4) If the bearing surfaces are flooded with oil, the friction is almost independent of the nature of the material of the surface in contact.

Friction is often classified as either static or dynamic. Static friction usually has a higher value than dynamic friction because it takes a greater force to start a body moving than to keep it moving once it is started.

Some typical values for static coefficient of friction are:

Material	Clean	Lubricated
steel on steel	.8	.16
steel on cast iron	.35	.20
steel on aluminum	.35	.16
steel on bronze		.13
steel on Teflon	.04	.04
steel on Polystyrene	.32	.30
aluminum on aluminum	1.35	.30
iron on iron	1.00	.18
leather on metal(clean)	.6	
brake material on steel	.4	

Values of coefficients of friction depends on conditions surrounding tests and on test procedures. Even slight changes in these conditions and procedures often results in significant variations in coefficients. Therefore, when using coefficients of friction in design calculations, due allowance must be made for possible differences between published values and the values that might occur in the application.

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Section 5 Bearings

This month we are going to talk about bearings and how to correctly apply them. The first installment will be about plain bearings (bushings) and next month we will cover anti-friction bearings.

There is two general type of bearings, plain bearings and anti-friction type bearings. Plain bearings rely on an oil film or more recently a slippery plastic type material to support the load, were as the anti-friction bearing uses rolling elements. The advantages of the plain bearing are that they require less space, are quite in operation, have good rigidity and their life is not limited by fatigue. The disadvantages are that the Plain bearing has a higher coefficient of friction, are susceptible to damage from foreign material and are very sensitive to lubrication. Modern plastic bearings have overcome some of the lubrication problems but have problems of their own such as the inability to operate at high temperatures.

Most plain bearings uses a fluid film between its moving surfaces. The formation of this film is dependent, in part on the design of the bearing and the speed of rotation or sliding. It results in three modes of operation called full film, mixed film and boundary lubrication. Full film, or hydrodynamic lubrication produces a complete separation of the sliding surfaces. Boundary lubrication takes place when the sliding surfaces are rubbing together with only a thin film of lubrication between them. Mixed-film mode of operation fall between the full film and the boundary modes. A journal bearing in starting up and accelerating to its operating speed passes through all three modes. The friction and thus wear is highest for the boundary lubrication and lowest for full film. This explains why oscillation type applications are the most difficult application to design a bearing for. In this case the relative velocity between the moving surfaces are constantly changing and therefore a good lubrication film is not able to established, thus the bearing operates in the boundary mode most of the time causing rapid wear.

Selecting and sizing plain bearings is relatively straight forward. The two factors that are usually used in selecting a bearing is the projected bearing area and the relative velocity between the moving surfaces.

The projected bearing area is found by multiplying the bearings inside diameter by its length. The load can then be divided by the projected bearing area to obtain the projected bearing pressure. The relative velocity is found by multiplying the inside diameter by .261 and multiplying the result by the rpm of the rotating part. The velocity caused by oscillating motion can be calculated by multiplying inside dia of the bearing by two times the included angle of motion, times the cycles per minute and then this quantity divided by 1380.

If we multiply the velocity by the projected bearing area we get a factor called the PV factor. The PV factor is often used to determine the maximum load that a bearing can carry. Table 1 shows the operation limits of some common bearing materials.

The choice of bearing material is determined by the operating conditions and the fabrication method desired. In our model work bronze is usually the material of choice, either solid material or sintered material that is oil impregnated. Sintered bronze is compressed from bronze powder and then fused together in a furnace. This leaves a porous structure that can be saturated with oil so that the bearing becomes self lubricating. Either type works well assuming proper lubrication is maintained.

COMPARISON OF NONLUBRICATED BEARINGS AND THEIR OPERATING LIMITS*

	Load Capacity (psi)	Max Temp (F)	Max Speed (fpm)	PV Limit (P=psi load) (V=surface fpm)
Porous bronze	4500	160	1500	50,000
Porous Iron	8000	160	800	50,000
Phenolics	6000	200	2500	15,000
Nylon	1000	200	1000	3,000
TFE	500	500	50	1,000
Reinforced teflon	2500	500	1000	10,000
TFE fabric	60,000	500	50	25,000
Polycarbonate	1000	220	1000	3,000
Acetal	1000	180	1000	3,000
Carbon-graphite	600	750	2500	15,000
Rubber	50	150	4000	-
Wood	2000	160	2000	12,000

Figure 1

PERMISSIBLE UNIT LOADING ON LUBRICATED JOURNAL BEARINGS

Type of Bearing	Viscosity SUS at 100^{o}F	Unit Load, psi of Projected Area
Diesel engines:		
Main bearings............................	250-850	800-1,500
Crankpin.................................	250-850	1,000-2,000
Wristpin.................................	250-850	1,800-2,000
Electric-motor bearings..................	122-180	100-200
Marine diesel engines:		
Main bearings............................	250-500	400-600
Crankpin.................................	250-500	1,000-1,400
Steam turbines and reduction gears......	122-470	100-220
Automotive engines:		
Main bearings............................	150-850	500-600
Crankpin.................................	150-850	1,500-2,000
Air compressors:		
Main bearings............................	150-1,700	120-240
Crankpin.................................	150-1,700	240-400
Aircraft-engine crankpin.................	150-500	700-2,000
Centrifugal pumps........................	122-180	80-100
Miscellaneous bearings...................	122-250	80-150
Automotive transmissions.................	800-1,500	80-150

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* Machine Design - 1970 Bearing Reference Issue

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Section 5 Bearings

As promised last month we will finish up this section by covering the class of bearing called a roller bearing. Some of the variety of standard roller bearings are shown in figure 1.

Straight roller bearings will carry a greater load than ball bearings of the same size because of the greater contact area. However, they have the disadvantage of requiring perfect geometry of the raceways and rollers because they are not self-aligning. A slight misalignment will cause the rollers to skew and get out of line which leads to increased friction and wear. Straight roller bearings will not take thrust loads which may or may not be a problem.

Helical rollers are made by winding rectangular material into rollers, after which they are hardened and ground. This gives the bearing the capability to accept some misalignment. This type of bearing is not very common but you might run across one some day.

The spherical roller bearing is very common and has the greatest load carrying ability of any roller bearing. They have great radial load capacity and in fact the contact area between the races and the rollers increase under heavy loading conditions. Although primarily used for radial load application, the spherical roller bearing can take some axial load as well.

The needle bearing is simular to the roller bearing with the great advantage that they are available in much smaller sizes than the typical roller bearing. This makes them an excellent choice for model engineering applications. They can also be used without an inner race, the rollers riding directly on the shaft provided that the shaft is hard enough to support the rollers. Normally 50 to 60 on the Rockwell "C" scale although for light loads a softer surface can sometimes be used. Mild steel generally is not adaquite.

Tapered roller bearings combine the advantages of ball and straight roller bearings since they can take either radial or thrust loads or any combination of the two, and, in addition, they have the high load carrying capacity of straight roller bearings. The tapered roller bearing is designed so that all elements in the roller surface and the raceway intersect at a common point on the bearing axis.

Most of the bearing types that we have covered in the last couple of

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Section 5 Bearings

This month we will cover antifriction bearings. The term antifriction is used to describe that class of bearing in which the load is transferred through elements in rolling contact rather than in sliding contact as in plain bearings. In a rolling bearing the starting friction and the running friction are about the same, and the effect of speed and temperature variation on the friction are small. It is probably a mistake to describe a rolling bearing as "antifriction" since some friction does exist, but the term is generally used.

Antifriction bearing can be divided up into two types. The first is the ball bearing which uses balls as the rolling element and the second is the roller bearing that uses rollers of various shapes to act as our rolling elements.

Ball bearings are designed to take pure radial loads, pure thrust loads, or a combination of the two and are composed of four essential parts. There is the outer race, the inner race, the balls and the separator. In low-priced bearings the separator is sometimes omitted, but it has the important function of separating the elements so that rubbing contact will not occur. Some of the most common types are shown in figure 1.

The single-row deep-groove ball bearing will take radial load as well as some thrust load. The balls are inserted into the grooves by moving the inner ring to an eccentric position. The balls are separated after loading, and the separator is then assembled.

The use of a filling notch in the inner and outer rings enables a greater number of balls to be inserted, thus increasing the load capacity. The thrust capacity is decreased because of the bumping of the balls against the edge of the notch when thrust loads are present.

The angular-contact bearing provides a greater thrust capacity. All these bearings may be obtained with shields on one or both sides. The shields are not complete closure but do offer a measure of protection against dirt. Seals are also available on many types of ball bearings. When seals are applied the bearing can be pre lubricated and further lubrication is not needed.

Single-row bearings will withstand a small amount of shaft misalignment or deflection, but where this is severe, self-aligning bearings can be used.

Assuming the bearing is properly lubricated, mounted, and sealed against the entrance of dirt, fatigue is the cause of failure. For this reason the life of an individual bearing is defined as the total number of revolutions or as the number of hours at a given constant speed at which the bearing runs before the first evidence of fatigue develops. The rated life of a ball bearing as seen in a bearing catalog is usually given as the number of hours a bearing will last at a given constant speed and load. Because the failure mode is caused by fatigue, there is a very wide variation of bearing life between identical bearings. Therefore statistical methods must be used be the bearing manufacturer to determine the rated life. Often the term B-10 life is used to set the load rating for ball bearings. The B-10 life is defined as the number of hours that 90% of a given lot of bearings will last under a given load and speed.

I am running out of space and Ken told me to keep it down to one page this month so that we don't exceed our postage limit, so will keep roller bearings for next month.

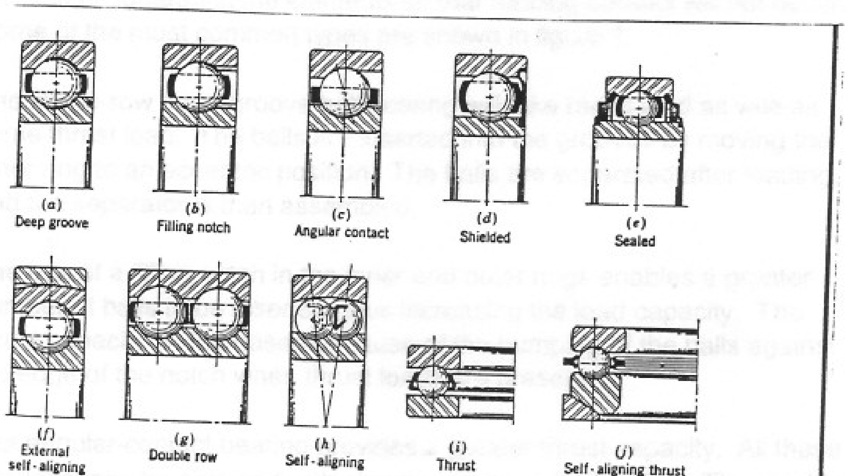

Figure 1