Application of Extreme Pressure Lubricants in the Oil Field

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Whenever two solid surfaces slide upon one another under load, the movement is accompanied by an undesirably high amount of friction. And as part of the power being transmitted, energy is comsumed by the high solid-to-solid friction, while wear and heat result and alter the load bearing surfaces and render them useless. Furthermore, if the wear does not completely destroy the mating surfaces, their operation becomes objectionably rough and noisy. Sliding friction is encountered by all load-bearing surfaces common to all machine elements whether they be plain or anti-friction bearings, gears, or ways.

The purpose of lubrication is to reduce friction and protect load-bearing surfaces so they may continue to function smoothly and continuously as designed. Ideally, this function is achieved by interposing a friction reducing film of some sort between the surfaces and completely separating the asperities of one surface from those of the other, for friction is produced by the contact between these minute peaks that are present on even highly polished surfaces.

Optimum lubrication depends on both tne correct selection and the proper application of a lubricant. This paper discusses the factors to be considered in selecting the correct lubricant. The factors which influence the selection of the lubricant are for the most part common to all machine elements regardless of the equipment of which they are a part. The lubricant in an anti-friction bearing does not care whether it is part of a precision machine tool or a twenty-ton hydraulic press.

The factors which must be taken into consideration in the selection of a lubricant are as follows:

(1) Type of lubrication, full film or boundary depending on the design, loads, speeds and sliding action of the machine element.
(2) Surrounding conditions such as operating temperatures, contamination with water or foreign matter.
(3) Service conditions, such as frequency of equipment operation, method of applying the lubricant, anticipated longevity of lubricant service.

Type of Lubrication

Conventional lubrication of moving parts is generally governed by one of two principles: full fluid film (hydrodynamic) lubrication and boundary (partial) lubrication. Full film librication is the more common and is applicable to nearly all types of continuous sliding action in which extreme pressures are not involved. Generally, it exists when all asperities of the lubricated surfaces are separated by a fluid film, and with no metal to metal contact, there is no wear. Friction is dependent primarily upon the fluid viscosity and running speed; however, under sufficient pressure, any type of grease or fluid supporting the surfaces is forced out. And, as will be discussed later, the surface asperities came within molecular contact, and boundary lubrication occurs.

Full film lubrication depends on sliding action whether it be with flat surfaces as in shoe-type thrust bearings or with cylindrical surfaces as in the case of journal bearings. The sliding action of gear teeth as they mesh with one another can promote full film lubrication. An examination of a Kingsbury thrust bearing will reveal the nature of full film lubrication (Fig. 1). If hydrodynamic lubrication is to be effected, and oil of the correct

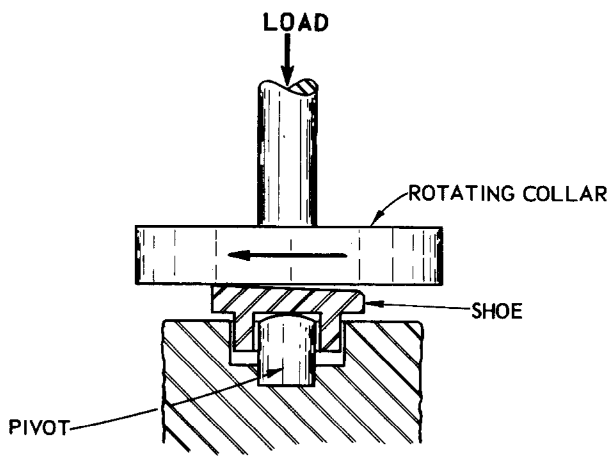

Fig. 1
DESIGN OF A KINGSBURY THRUST BEARING

viscosity must be applied at the leading edge of the shoe and incorporated into the shoe must be three design factors:

(1) The leading edge must not be sharp, but must be rounded to allow entry of the oil.
(2) The shoe must be mounted on a pivot so it is free to assume an angle which conforms to the oil wedge.
(3) The shoes of the bearing must have sufficient area and width to allow the oil film to carry the load. Insufficient area would result in a pressure too great for the full film to exist.

As the thrust collar rotates, the leading edge of the shoe encounters the supply of oil. However, if the oil is of the correct viscosity, it offers sufficient resistance to flow and is not wholly displaced by the shoe. Instead, a thin layer of oil remains on the collar surface above the shoe and, because of its rounded edge, the shoe permits the oil to enter. The shoe tilts to conform to the oil wedge, and if the surface asperities are completely separated, lubrication is occurring by the principle of hydrodynamic lubrication.

On the other hand, when surface asperities come within molecular contact, boundary lubrication occurs. The establishment of the full film may be impossible due to such factors as excessive bearing loads, insufficient running speeds, intermittent or reciprocal motion, shape and action of the sliding surfaces.

The Frictional Curve

The conditions of oil viscosity, speed, and pressure can be related to the coefficient of friction by the frictional curve. Let Z be the viscosity of the lubricant, N the sliding speed, and P the pressure (load per unit area). The coefficient of friction is the friction per unit load and is represented by the symbol μ. These factors can be related by the following equation:

$$\mu = f\left(\frac{ZN}{P}\right)$$

The above relationship usually takes the form of the frictional curve in Figure 2. Under the conditions of full film lubrication, friction is small, and if the pressure

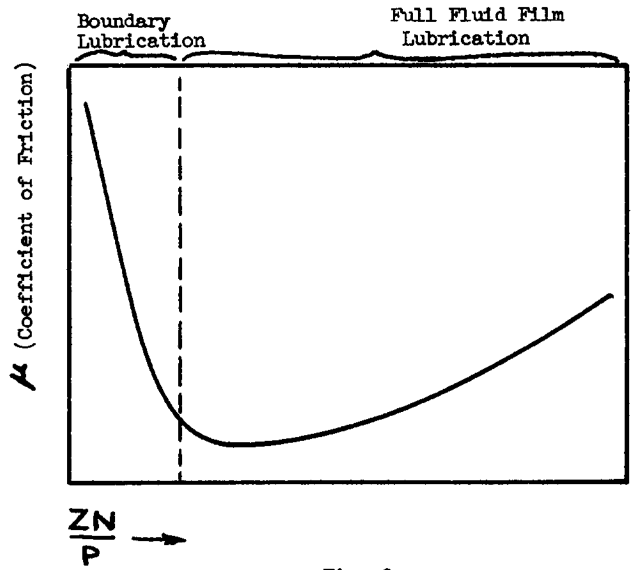

Fig. 2
A TYPICAL FRICTIONAL CURVE

is increased or viscosity or speed is reduced or a combination thereof occurs, the full film begins to deteriorate. This deterioration constitutes a movement to the left along the curve toward lower ZN/P values and into the region of boundary lubrication and results in a rapid rise in the coefficient of friction.

Journal or shoe type thrust bearings are designed to operate in the full film region of the frictional curve. Within limits, the value of ZN/P can be adjusted to keep friction low: for heavier loads more surface area is provided to maintain the same pressure; for slow speeds, an oil of higher viscosity is used to keep lubrication in the full fluid range. A straight mineral oil of the right viscosity will provide optimum full film lubrication.

Most types of gears can be lubricated in this region. In gearing, the action of the meshing teeth produces a combination of sliding and rolling contact. With spur, helical, spiral bevel, and herringbone gears, the sliding action is such that it promotes the formation of the fluid wedge. With worm and hypoid type gearing, the high rate of side slide wipes away the oil film and these gears usually operate in the region of boundary lubrication. The question of oil type, however, is not as clear in the case of journal and shoe-type thrust bearings. Other factors which influence gear lubrication are pinion speed, ratio of reduction, horsepower transmitted, nature of the load, and type of drive.

Pinion Speed

In the selection of the lubricant for speed reducers, each of these factors should be given due consideration. Higher pinion speeds require oils with lower viscosity since there is a greater tendency to produce the full fluid film, and more oil is drawn into the pressure area. Also the time available for oil to be squeezed from between the meshing teeth at the point of zero slide is exceedingly short. Furthermore, heavy bodied oils with high speed will produce excessive friction and tend to overheat the unit. At low pinion speeds, however, more time is available for the oil to be squeezed from between the teeth and less oil is drawn into the pressure area. Therefore, the lower the speed of the pinion, the heavier should be the body of the lubricating oil.

Ratio of Reduction

When the ratio of reduction is less than ten to one, a single reduction gear set is usually used and the oil can be selected on the basis of the speed of the driving pinion. On the other hand, when the reduction of speed is greater than ten to one, a multiple reduction gear set is usually employed. In this case, the oil viscosity should be selected to meet the requirements of the low speed pinion of the last reduction, for it is here that the film forming tendency will be the least.

Horsepower Transmitted

The horsepower transmitted dictates the tooth pressure which acts to squeeze the separating oil film from between the contact areas. Tooth pressure, therefore, is a factor to be considered in selecting proper body. The greater the pressure the more viscous the oil must be to resist the squeezing action and thus to maintain an effective oil film. The total force acting on gear teeth depends on the power transmitted and on the diameter and speed of the pinion. Formulae for the design of gears are fairly standardized and the maximum permissible unit loading for tooth surfaces largely depends on the Brinell hardness of the gear metals. Generally, gears of low horsepower rating are constructed with narrow face, small diameter, and small tooth size, but these dimensions are proportionately larger for higher horsepower ratings. The overall effect of scaling up to higher power capacities is that tooth pressures are held more or less constant even though higher horsepower rated gears result. However, there is more surface area for the production of friction and there is greater churning action in the reservoir. These contribute to the power loss converted into heat. Therefore, the higher the horsepower rating of the gear set the hotter it will operate. And since oils thin out at higher temperatures, it is advisable to offset this condition by using heavier bodied oils for the larger and hotter units.

Nature of the Load

The nature of the load has an important influence on the selection of the oil. Uniform loads produce tooth pressures that are uniform; but excessive tooth pressures, produced by erratic or shock loads, tend to rupture the lubricating film that would otherwise separate the gear teeth if the load were more uniform. If erratic and shock loads have not been considered in the design of the gear set, an oil of heavier body is required to prevent film rupture. In some cases, this condition may be still more severe because of overloads or because of a combination of heavy and extreme shock loads, e.g., with fluid pounds or with gears starting under heavy load. In such cases, it may be impossible to maintain a full film and boundary lubrication may prevail; so excessive wear will result unless there is used a special oil that possesses extreme pressure or film strength characteristics.

In instances when gears are so heavily loaded that the full fluid film cannot be maintained, metal to metal contact can occur. Welding and scoring of the tooth surfaces occur at the intense localized pressures and high spot temperatures between the surfaces in contact (Fig. 3), and the mating surfaces are destroyed and rendered useless. Under these conditions, special chemical additives called "extreme pressure agents" are

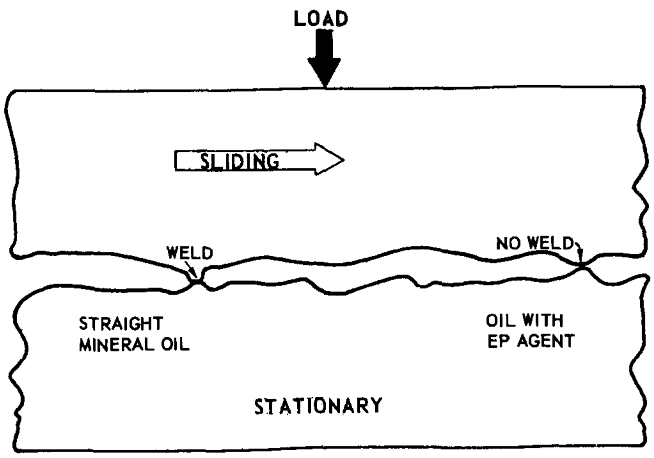

Fig. 3
EXTREME PRESSURE AGENTS PREVENT WELDING

required to control welding and reduce wear to an acceptable limit. For milder extreme pressure conditions such as might be encountered in heavily or shock loaded spur, bevel, and helical gear drives, somewhat less active compounds are appropriate. Tricresyl phosphate, sulfurized sperm oil, lead oleate, or lead naphthenate are operative at more moderate local temperatures. For severe pressure conditions such as those encountered in automobile hypoid gears, more active chemicals like chlorine derivatives are required.

Type of Drive

The nature of the drive also determines the uniformity of the pressures on the gear teeth. The uniform torque of an electric motor throws no additional load on the teeth of the gears transmitting the load. Such units as reciprocating engines may require heavier bodied oils or extreme pressure additives for the protection of the mating surfaces.

Surrounding Conditions

In selecting the lubricant, the conditions under which it is to perform must be taken into consideration. The surrounding conditions include ambient temperature, sun, airborne sand, dust, and humidity. Some gases can have a deleterious effect on the oil or contribute to corrosion.

The operating temperature is an important consideration because it dictates the apparent viscosity of the lubricant. As previously pointed out, when gears operate heat is generated by friction and the churning of the oil. In a fully loaded spur, helical, or bevel gear set, the temperature rise will be approximately 50 F over ambient. Thus, for these gear sets, operating temperatures should not exceed about 160 F. Hypoid and worm gears normally operate at higher temperatures and a 90 F rise when loaded to capacity is not uncommon for these gears: thus, about 200 F maximum operating temperature is normal for worm and hypoid gears. If for some reason higher ambient temperatures or greater rises than normal are anticipated, a heavier bodied lubricant may be required.

Bearings do not normally produce high operating temperatures. However, here again the ambient temperature must be taken into consideration and this consideration is especially valid with bearings lubricated with grease which does not flow away from the heat and which offers a high resistance to heat transfer. High temperatures may cause excessive oxidation resulting in bearing failure; therefore, when high temperatures are encountered, more frequent lubrication should be practiced.

Water and foreign matter can and often do find their way into the lubricating system of enclosed equipment. Water may come from cooling coils or may result from condensation of moisture in the atmosphere. Foreign matter such as dirt or dust can work their way into bearings or accumulate in the oil and scratch mating surfaces. The importance of cleaning zerk fittings before greasing cannot be over-emphasized.

Water in severely oxidized or dirty oil usually forms a permanent emulsion which does not have load carrying capabilities; so the protective film between the loaded surfaces breaks down and excessive wear of the gear surfaces and bearings result. For this reason, an oil used in gear lubrication must have high emulsibility, i.e., the ability to separate quickly and completely from water. Water also causes corrosion of bearings and gear surfaces; therefore, oils that possess special anti-rust properties are required.

Service Conditions

The severity of the lubricant service has an important influence on the selection of the lubricating oil. Oils used for once-through lubrication do not need high oxidation resistance, high viscosity index, or special anti-rust character. On the other hand, the conditions imposed by enclosed systems require oils with a high degree of the aforementioned qualities. In an enclosed system the oil is repeatedly fed to the point of gear tooth mesh or any mating surfaces which it is to lubricate; and it is exposed to elevated temperatures for long periods of time and is expected to flow properly over a wide range of ambient temperatures, to accommodate climatic variations.

The lubricant in an enclosed or circulating system is subjected to high temperatures and intimate contact with air. Furthermore, it should serve for a reasonably long period of time. Therefore, it must possess good oxidation resistance or it will become unduly thick and cause unnecessary power losses or will form varnish and deposits which impair efficient lubrication and prohibit adequate heat transfer.

Very often gear installations are exposed to climatic conditions. During low ambient temperatures the oil must be fluid enough to distribute itself over the working areas of the gear teeth; on the other hand, when the unit has come up to its final operating temperature, the oil must not be unduly thin and be incapable of supporting the load. Oils used over wide ranges of ambient temperature must possess relatively constant viscosity - temperature characteristics (a high VI).

As previously mentioned, the oil in an enclosed system can become contaminated with water from condensation. This condition is apt to occur in gear sets operated intermittently when warm periods of operation alternate with cool periods of idleness. The system breathes and moisture in the air is condensed on cool interior surfaces; and if the periods of idleness are long, highly corrosive conditions can develop. Service conditions such as these require an oil with good anti-rust character. In addition to the possibility of rust on gear tooth surfaces leading to failure, rust and water in the oil have a catalytic effect on the rate of oil oxidation.

An Application for Extreme Pressure Lubricants

The problem of lubricating pumping unit speed reducers appears to be a good example of sorhe of the principles discussed above. These units probably experience considerable shock loading because of rough running, slow firing engines, sticking plungers, plungers bumping bottom, and gas and fluid pounds. It is also possible that they may be overloaded from poor counterbalance and at times intentionally overloaded to meet production schedules. Furthermore, operating schedules are far from the optimum. A reducer may operate for as little as 3 to 6 hr per day only 9 to 15 days of each month. This service can result in excessive corrosion which can significantly shorten the life of the reducer.

Most manufacturers follow the recommendations of API Standard RP 11G which indicates that straight mineral and crankcase oils may be used. These recommendations are contingent upon operation of the unit at rated capacity and in accordance with API standards; however, this situation is the exception rather than the rule. The speed reducer is almost never sized for the most severe operation anticipated, since this type sizing would result in excessive lifting cost per barrel. And still another common cause of gear failure is improper counterbalance which can impose overloads in excess of 100%.

In view of the above, a gear oil with mild extreme pressure and good anti-rust characteristics should result in increased reducer life over straight mineral gear oils or crankcase lubricants.

One type of mild EP oil is compounded with lead naphthenate and sulfurized sperm oil. It has wide acceptability by heavy industry and is recommended by AGMA Standard 252.01 which specifies that acceptable mild EP oils be non-separating, resistant to foaming, non-corrosive, clean, stable and rated at no less than 30 lb Timken OK rating. Oils formulated with lead naphthenate and sulfurized sperm oil meet these specifications. Additionally, a few of these products have a 60 lb Timken OK load rating for the most severe operations.

Experience with these lubricants has shown that when working loads exceed 1,000 lb per linear in. of gear tooth face, these oils are mandatory. However, between loadings of 500 and 1,000 lb per in. their use depends upon the nature of the operation such as shock loading and periodic over loads.

These oils are also particularly suitable for worm gear lubrication and are available in grades especially tailored for both undershot and overshot installations.

Oils compounded with these ingredients can tolerate considerably more water without an effect on the extreme pressure characteristics of the oil. This toleration is expecially important for reducers operating intermittently, because such operation is also conducive to the formation of rust on the gears during idle periods. Fortunately, lead naphthenate has excellent rust preventive characteristics even in the presence of very high percentages of water.

Summation

The final selection of a lubricant should be based on an evaluation of its performance in the application. The economies -- savings in repair and replacement costs, increased production, or reduced oil consumption -- resulting from a higher quality oil should be weighed against higher oil costs. Usually, if technical considerations indicate an advantage for an oil with specific characteristics the evaluation will confirm this advantage. In addition, large scale operations can realize significant savings from only minor improvements in oil quality since incramental savings per unit are multiplied by the number of units involved.