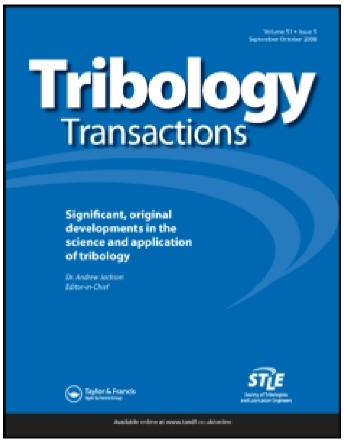

Tribology Transactions
Publication details, including instructions for authors and subscription information:
http://www.informaworld.com/smpp/title~content=t713669620

Energy Efficiency of Industrial Oils

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First published on: 01 October 1999

To cite this Article Bronshteyn, Lev A. and Kreiner, Jesa H.(1999) 'Energy Efficiency of Industrial Oils', Tribology Transactions, 42: 4, 771 — 776, First published on: 01 October 1999 (iFirst)
To link to this Article: DOI: 10.1080/10402009908982281
URL: http://dx.doi.org/10.1080/10402009908982281

Vol. 42 (1999), 4, 771-776
TRIBOLOGY TRANSACTIONS

Energy Efficiency of Industrial Oils©

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Lubricants influence energy efficiency mainly through reducing energy losses, which include churning losses and friction losses in hydrodynamic, elastohydrodynamic and boundary lubrication regimes. The total energy loss depends on lubricant viscosity and chemical composition. Different sources of lubricant-related power losses in industrial systems are described. The dependence of churning and friction losses on oil properties is analyzed.

Viscosity shear-thinning and pressure-thickening effects and their dependence on base oil and viscosity index improver chemical composition are examined. The role of pressure-viscosity relationships is emphasized. Some aspects of oil compressibility and viscoelasticity are discussed in regard to oil energy efficiency. The mechanism and role of friction modifiers in industrial oil formulations are described.

Significant savings in machinery energy consumption can be achieved by using energy-efficient lubricants. Methods for improving industrial lubricant energy efficiency are discussed and potential savings in energy consumption are estimated.

KEY WORDS

Energy Conservation; Gear Lubricants; Hydraulic Fluids; Friction Modifying Additives; Viscosity-Pressure Behavior

INTRODUCTION

How much energy can be conserved by using energy-efficient lubricants in comparison with conventional ones? It has been estimated that approximately 11 percent of the total energy annually consumed in the U.S. in the four major areas of transportation, turbomachinery, power generation and industrial processes can be saved through new developments in lubrication and tribology *(1)*. A simple analysis reveals that supplying all of the worm gear drives in the U.S. with a lubricant that increases mechanical efficiency five percent in comparison to a conventional mineral oil would result in an annual saving of $6 billion *(2)*. Industrial surveys report that average electrical energy savings of eight percent were achieved by changing lubricants in compressors from petroleum to synthetic diester oils of identical ISO viscosity grades *(3)*. More recent studies have shown energy savings of five percent in screw, vane and reciprocating compressors switched from mineral to synthetic oil *(4)*.

The majority of contemporary industrial oils have an additive package containing rust and oxidation inhibitors, defoamant, pour point depressant, and sometimes antiwear and/or EP additives, blended in mineral oil basestocks of different viscosities. The energy efficiency of these oils can generally be improved either by using additional supplements, separately purchased and blended into the oils, or by replacing a conventional oil with a new, fully-formulated, more efficient industrial oil.

Fuel-saving benefits obtained with synthetic and hydrotreated engine oils and transmission fluids have been extensively reported *(5)-(11)*. It was supposed that two main factors contributed to the fuel economy properties of engine oils: the high-temperature high-shear viscosity (HTHSV) and the boundary friction coefficient.

The best energy efficiency can be achieved when using all possibilities for improvement, including boundary friction modification and viscosity optimization. Friction modifiers, being added to the oil additive package in low concentrations (about one to two percent), reduce boundary friction. Such concentrations have very little effect on the oil viscometric behavior. Therefore, these two ways of obtaining energy efficiency improvements can be considered and utilized almost independently.

SOURCES OF POWER LOSS IN INDUSTRIAL SYSTEMS

Energy loss in industrial equipment consists of friction losses in the bearings, gear mesh, vane-ring, piston-liner or other friction couplings as well as churning losses, seal losses, and losses in the auxiliaries, such as valves, filters, etc.

The numerical distribution of power losses is load dependent.

**Presented at the 54th Annual Meeting
Las Vegas, Nevada
May 23-27, 1999
Final manuscript approved March 12, 1999**

TABLE 1—MECHANICAL EFFICIENCY OF GEAR TRANSMISSION FOR DIFFERENT REGIMES *(5)*

Parameter/Regime	Low Load, Low Speed	High Load, Low Speed	Low Load, High Speed	High Load, High Speed
Mechanical efficiency, %	91.14	94.95	87.33	95.02
Relative churning losses (if 100 assigned to high load, high speed regime)	75	40	165	100
Temperature of oil in steady state condition, °C	50	85	66	108

The total power loss in a system P_{TOTAL} has two parts, i.e., the power loss without load P_0 and the increment of power loss due to the loading P_{LOAD}:

$$P_{TOTAL} = t_0 \bullet P_0 + (1 - t_0) \bullet P_{LOAD} \quad [1]$$

where t_0 is the fraction of time when the system is unloaded.

The no-load power loss P_0 consists of two parts:

$$P_0 = P_{CH0} + P_{F0} + P_{LO} \quad [2]$$

where P_{CH0} is the churning losses, P_{F0} is the friction losses and P_{LO} is the leakage losses under no-load conditions.

The no-load churning losses are caused by churning, squeezing, ventilation, acceleration and deceleration of lubricant in pumps, gears and other devices. The no-load friction and leakage loss can often be neglected because their values are very small.

The load-dependent power loss P_{LOAD} consists of four parts:

$$P_{LOAD} = P_{CH} + P_F + P_{AX} + P_L \quad [3]$$

where P_{CH} is the churning losses, P_F is the friction losses, P_{AX} is the losses in auxiliaries, and P_L is the leakage losses under load conditions.

Since the churning and friction losses represent the most significant fraction of the total power losses, the authors will concentrate on them in the analysis. Thus, because the values of P_{F0}, F_{AX}, P_{LO} and F_L are negligibly small in comparison with P_F, the authors obtain

$$P_{TOTAL} - t_0 \bullet P_0 + (1 - t_0) \bullet P_{LOAD} - t_0 \bullet P_{CH0} + (1 - t_0) \bullet (P_{CH} + P_F) \quad [4]$$

The difference between P_{CHO} and P_{CH} in Eq. [4] is that the viscosity of the compressed oil under load conditions in hydraulic systems will be higher, resulting in higher churning losses in comparison with no-load conditions. For example, in contemporary hydraulic systems, pressures of 35 MPa (5000 psi) can be reached, approximately doubling the viscosity of the oil. The pressure in hydrodynamic journal bearings may be twice as high as in a bulk fluid (approximately 70-100 MPa) resulting in a four to five times viscosity increase compared to unload conditions. This may contribute additional friction losses due to increased viscous drag in bearings. In addition, the bearing power losses depend on lubricant viscosity behavior at high share rates, which occur in thin film bearing lubrication typical of the hydrodynamic or elastohydrodynamic regime.

Churning losses in gear systems, sometimes called splashing losses, can be estimated with Eq. [5]:

$$P_{CH} = 10^{-18} \bullet g \bullet \rho \bullet \eta^{0.6} \bullet \upsilon^{2.15} \bullet r \bullet b \quad [5]$$

where g is the coefficient depending on gear wheels, ρ is the oil density, η is the kinematic viscosity, υ is the peripheral speed of the gear, r is the pitch radius, and b is the wheel width.

Taking into consideration only lubricant-related parameters, the authors obtain

$$P_{CH} = K \bullet \rho \bullet \eta^{0.6} \bullet \upsilon^{2.15} \quad [6]$$

where K $10^{-18} \bullet g \bullet r \bullet b$ is a coefficient depending on gear wheel parameters.

It is important to note that churning losses in gear systems depend on oil density with exponent one, viscosity with exponent 0.6 and speed with an exponent of about 2. This dependence is not likely to be changed much for no-load hydraulic pumps because the nature of the process remains the same. But in loaded hydraulic systems, higher viscosity and density of a compressed fluid will contribute to higher churning losses.

Table 1 illustrates the mechanical efficiency and churning losses experimentally measured for a car's manual gear transmission with a reference mineral oil operating in four regimes with different levels of load and speed *(5)*. These data indicate that the relative contribution of churning losses to total losses is maximum for low-load, high-speed operating conditions that result in the minimum mechanical efficiency of the transmission.

In spray-lubricated gear transmissions, no-load losses are lower than in bath-lubricated systems, but the viscosity dependence of losses is approximately the same. A four-times increase in viscosity has caused approximately a 1.6 to 1.7 times increase in no-load power losses for spray-lubricated gears *(12)*, which roughly corresponds with the exponential dependence shown in Eq. [5].

It has been shown *(13)* that friction losses for gear transmissions are proportional to the fiction coefficient that is measured independently for the same friction conditions. The dependence of friction losses on load, gear tooth surface roughness and oil viscosity was expressed by exponential correlations *(13)*

$$P_F \propto (P_N/b)^{\delta} \cdot R_{MAX}^{\beta} \cdot \eta_{90}^{\varphi} \qquad [7]$$

where P_N is the normal load on the tooth surface, b is the width of the gear tooth, R_{MAX} is the maximum surface roughness and η_{90} is the oil kinematic viscosity at 90°C.

The values of the exponents in Eq. [7] are $\delta = 0.1$, $\beta = 0.08$, $\varphi = 0.05\text{-}0.12$. The value of φ is dependent on load: the small value is used for high load and the large value is used for low-load conditions. But even for low-load conditions, boundary or EHD lubrication is supposed to exist in a sliding gear contact because Eq. [7] is based on the assumption that the friction coefficient decreases while sliding velocity increases. This is a characteristic of boundary and EHD lubrication. Hydrodynamic lubrication exists in the range of sliding velocities greater than 1 m/s *(13)*.

It is interesting to note that even the large value of $\varphi = 0.12$ in Eq. [7] is considerably less than 0.6 which is used in Eq. [6] for the churning losses-viscosity dependence. That means that viscosity has a much stronger effect on churning losses than on friction losses. For high-load conditions $\varphi \rightarrow 0$ and friction losses become almost independent of oil viscosity.

ROLE OF OIL VISCOSITY

Energy losses are defined by initial oil viscosity and its temperature, pressure and shear rate dependence. The initial oil viscosity is usually chosen based on considerations other than energy efficiency such as flow rate, temperature range, wear, leakage, etc. Conventional industrial hydraulic fluids for plant indoor applications do not usually contain viscosity index improvers (VII) as the fluids are not supposed to be used at sub-zero temperatures.

The shear-thinning effect due to temporary mechanical realignment of a polymeric viscosity index improver (VII) has been observed for multigrade engine oils to give better fuel economy than single-grade oils. This reversible behavior reduces the effective viscosity of the lubricant under high shear operating conditions resulting in a reduction of viscous drag and an improvement in fuel economy. Simultaneously, this non-Newtonian oil behavior causes the EHD film thickness drop to the value defined by the effective high shear viscosity. Thus, the potential negative effect of viscosity thinning turns out to be beneficial for energy conservation, providing that the effect is reversible.

Hydraulic pumps operate at high fluid shear rates. An oil formulation that shear thins dramatically may have a much higher viscosity at low shear rates but would pump better and require less driving power in actual practice.

There is a great number of synthetic fluids as well as conventional mineral blends that can be used as base fluids for contemporary industrial oils. The most common alternatives to conventional mineral-based oils are synthetic polyalphaolefins, polyglycols and esters, as well as so-called "very high viscosity index" (VHVI) petroleum hydrogenated base oils produced either by severe hydrocracking or by paraffin isomerization processes developed in recent years.

It is assumed that petroleum base oils do not experience any shear-thinning effects within a reasonable range of shear rates. However, when studying ultrathin liquid films *(14)*, the shear thinning effect was found to exist even for pure hydrocarbon fluids like hexadecane that was previously considered to be Newtonian. Shear-thinning was observed starting from a shear rate as low as 10 s^{-1} for layers of one-to-three molecular dimensions confined between atomically smooth plates. The onset of shear thinning implies the existence of a characteristic rheological relaxation time of about 0.1 s. Since the relaxation time of single hexadecane molecules is well-known to be less than 10^{-10} s, this finding may reflect collective motion rather than the motion of single molecules.

It is interesting to point out that, in this case, it is not necessary that the liquid boundary structure be an ordered structure similar to an ionic solid. The ordering is expected for polar long-chain adsorbed molecules, such as stearic acid, but not for hexadecane. This structure may be analogous to an amorphous glass state, which is well-known to occur for liquid lubricants under glass transition pressure or at very low temperatures. Contrary to an ordered structure with more expressed viscoelasticity, an amorphous structure behaves like a plastic solid.

Results closely associated to those mentioned above for hexadecane were found for highly pressurized liquid lubricants *(15)*, *(16)*. These findings confirmed that the viscosity is very sensitive to fluid structure, which is quite different in boundary layers from that in the bulk because of the influence of the solid surface. The phenomena of surface vitrification and the aligning effects of solid surfaces may be contributing factors in determining differences in the performance of petroleum and synthetic base oils.

Another factor is oil viscosity-pressure behavior. It is described by pressure-viscosity coefficient α, the value of which reflects the rate of oil viscosity increase under increasing pressure. In comparison with conventional mineral oils, highly paraffinic polyalphaolefin and hydrotreated base oils have lower α and their viscosity is less dependent on pressure. Introduction of a DI additive package into base oil decreases its pressure-viscosity dependence as well *(7)*.

Compressibility is vitally important for hydraulic fluids, one of the main functions of which is to transmit power. Precise power transmission in high pressure hydraulic systems requires the fluid viscosity (as well as volume) to be the least sensitive to pressure or have the smallest pressure-viscosity coefficient α. Low viscosity at high pressure is also beneficial for the reduction of viscosity-related churning power losses under load conditions. However, a certain compressibility can be convenient in that it dampens pressure surges caused by switching and thus provides smoother operation.

Pressure-viscosity and temperature-viscosity properties are known to be loosely related. High viscosity index oils tend to be less compressible. Both these properties of liquid hydrocarbons are strongly dependent on molecular structure. According to existing theories of liquid structure, the more degrees of freedom in the molecular structure of a liquid, the higher its compressibility *(17)*.

The decrease in free molecular space under high pressure results in a higher tendency of structuring, and shear thinning starts at lower shear rates. When the molecules of the liquid have made all the steric adjustments possible, further increase of pressure works against intermolecular repulsive forces. For very high pressures, a glass transition to solid-like behavior occurs and a

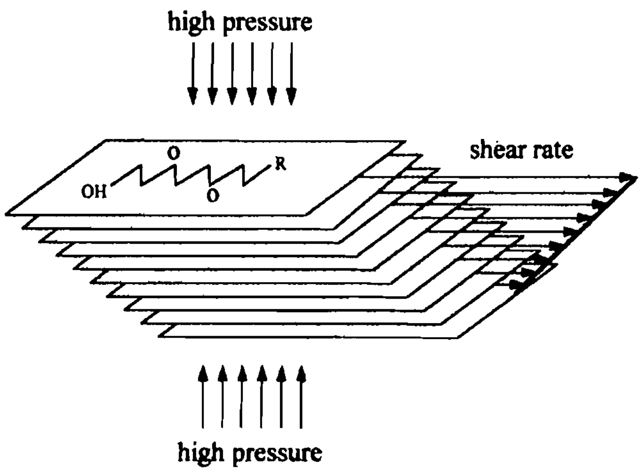

Fig. 1—Paper stack model of lubricant behavior in EHD contacts.

critical shear stress is required to initiate sliding. For mineral oils, the glass transition pressure is usually about 0.8 GPa at 30°C *(18)*.

In practice, a hydraulic fluid is often simultaneously exposed to temperature, pressure and shear stresses. Sharp pressure surges in a hydraulic system cause cyclical heating and cooling of oil due to energy dissipation. This energy dissipated in hydraulic fluids may be much higher than in gear oils, in which temperature is usually controlled by friction heat developed in the gear mesh. When pressure grows, causing oil thickening, the temperature rise coincidentally imposes the opposite effect of viscosity drop, compensating the final viscosity change. The dissipation process and energy loss is dependent on the relationship of dynamic parameters of the mechanical impact and fluid relaxation time which defines oil viscoelastic properties.

Besides the shear thinning effect, VI-improved oils exhibit another aspect of non-Newtonian behavior: viscoelasticity. The viscoelastic effect is considered to be important for energy conservation and friction reduction. The ability to store elastic energy and produce normal forces during steady flow has been measured in lubricants which contain high molecular weight VII. The authors of Ref. *(19)* reported that long-chain polymer VII created the "network" structure with rubber-like properties in a low molecular weight liquid medium. The viscoelastic effect of this structure results in normal stresses exerted on the bearing surfaces and contributes to the oil film thickness *(19)*, *(20)*. The authors of Ref. *(21)* showed that assuming relaxation time to be proportional to viscosity, viscoelastic normal stresses can result in increases of 20 percent and higher in load-bearing capacity. That, in turn, may contribute to friction reduction and energy conservation. The numerical modeling suggested that at high eccentricities in journal bearings (the most severe HD case, close to EHD), pressure-thickening dominates the viscosity behavior rather than shear-thinning or temperature-thinning. That means that lubricant rheology, as well as friction energy losses for highly loaded contacts, tends to be affected more by the pressure-viscosity relations than temperature or shear dependence.

ROLE OF OIL MOLECULAR STRUCTURE AND CHEMICAL COMPOSITION

While viscosity mainly controls the hydrodynamic behavior of lubricants, their molecular structure has a direct effect in EHD and boundary lubrication. The effect of a lubricant's molecular structure on its friction behavior can be illustrated by the so-called paper stack model shown in Fig. 1 *(22)*.

Each sheet of paper in the stack represents a lubricant molecule. The stack is compressed vigorously in a direction perpendicular to the paper plane and is subject to a rate of shear in trying to pull single sheets of paper out of the stack. This action will be more difficult and will require more energy the more uneven the paper surface is and/or the stronger the interaction between the individual paper planes. This corresponds to a high traction coefficient.

The single molecules are accelerated by a shear stress to a different extent depending on the distance from the contact surface. The resistance to a shearing stress depends on a shape of the molecule. Disk-shaped molecules with nonbranched hydrocarbon chains provide lower traction. This model concept also allows one to predict that molecules shaped like balls or ellipsoids should have lower friction because they are susceptible to rotating, sliding, rolling and spinning motions.

Even insignificant changes in molecular structure can have extraordinary effects, provided they take place near the "active molecular effect center" responsible for molecular interactions. Usually these centers are oxygen, nitrogen, sulfur or other nonhydrocarbon heteroatoms of functional groups attached to a molecular carbon frame. Minimizing functional groups involved in intermolecular interactions facilitates molecular motion and decreases friction.

Polycyclic fluids have higher traction coefficients than monocyclic *(22)*, *(23)*. It was also shown that the pressure-viscosity coefficient α has the predominating effect on the maximum friction coefficient. The degree of branching of the carbon chains in a molecule was defined as a critical parameter of aliphatic oil frictional behavior causing α to increase for more branched chains. The effects of molecular structure were shown to become more important at elevated pressures.

The effect of oil chemical structure on friction losses was studied in more detail in *(24)*. The comparison of losses measured for the high and low sliding velocities and load range showed that the benefits of synthetic fluids were more evident for the high-speed and high-load test conditions that characterize newer machinery designs. Two kinds of synthetic oils - polyalphaolefin and polyglycol - behaved alike: they gave 12-20 percent lower friction losses in comparison with a mineral oil for the high-speed and high-load conditions and showed no practical differences in losses for the low-speed and low-load test.

The hydrocracking process results in basestocks containing more paraffinic and less aromatic hydrocarbons while PAO consists of almost only paraffins. This molecular structure, as was discussed previously, has better temperature-viscosity and pressure-viscosity behavior, resulting in a high viscosity index and low pressure-viscosity coefficient α. The high pressure viscosity of the VHVI base stocks is substantially lower than that of the sol-

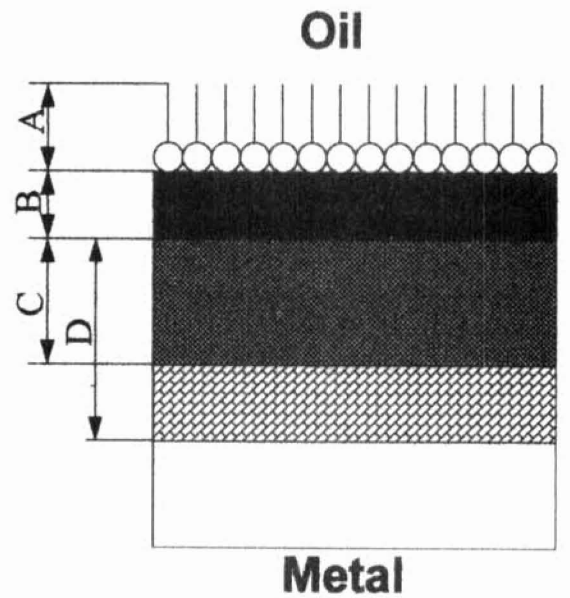

Fig. 2—Structure of metal friction surface layer.

vent-refined basestocks. Therefore, hydrocracked and PAO oils show significant advantages over conventional mineral oils in friction reduction under EHD lubrication conditions. It is important to note that PAOs are more effective than hydrocracked oils. Similar results were reported for automotive transmission and power steering fluids formulated with these basestocks *(11)*.

Energy-saving benefits have been obtained with synthetic industrial gear oils in comparison with paraffinic mineral oil *(12), (13), (24)*. Mesh power loss in a gear transmission was reduced by as much as 50 percent by using polyglycol-type lubricants *(12)*. Synthetic polyalphaolefin and polyglycol based oils have shown friction in the hydrodynamic lubrication regime approximately 12-20 percent lower than that using mineral oil *(24)*.

Conventional mineral-based hydraulic fluids which are used at plants usually do not contain VII because these fluids are not exposed to very low temperatures. Nevertheless, VII can help to utilize the shear-thinning effect for friction reduction and energy conservation as was described earlier. Polymers can also reduce leakage problems in hydraulic systems. Apparently, the best polymer for these purposes should significantly thicken oil at low shear rates but start thinning even at shear rates corresponding to churning of the oil.

The most commonly used VIIs are polymethacrylates and olefin copolymers. Polymethacrylates have more stable viscosity-shear characteristics than olefin copolymers. The shear-thinning effect tends to be more pronounced for higher molecular weight polymers and higher polymer concentrations in oil *(25)*.

Friction modifiers create boundary layers on rubbing metal surfaces, physically adsorbing and chemically interacting with the surfaces. The structure of these boundary layers is different from the surface layers created by regular antiwear or EP sulfur, phosphorus or chlorine-containing additives. These additives reduce wear, but, as a rule do not decrease (and often increase) friction force and energy consumption.

There are two main groups of oil-soluble friction modifiers, which are proven to be effective in engine oils: organomolybdenum compounds and ashless organic ester- or acid-based additives like stearic acid or octadecanol. Organomolybdenum additives such as molybdenum dithiophosphate (MoDTP) or molybdenum dithiocarbamate (MoDTC) react chemically with surfaces, and are effective for boundary lubrication at high temperatures. Their friction reduction activity is based on the thermally activated chemical reaction with metal surfaces, and their effectiveness is higher at higher temperatures *(26)*. Long-chain polar organic acids or esters work at lower temperatures providing relatively thick antifriction films adsorbed on metal surfaces.

The antifriction efficiency of friction modifiers depends on the composition and thickness of the surface layers created on rubbing metal surfaces. The suggested structure of these surface layers is shown in Fig. 2 *(27)*.

The multilayer composition consists of A - a layer of adsorbed lubricant molecules; B - a resin-like film of oxidation and destruction products of the lubricant components; [this film contains elements (C, O, S, P, Ca, Zn, etc.) included in the lubricant composition]; C - a layer of secondary structures formed during the friction process as a result of diffusion of active components of the lubricant (mainly O and S) into the metal surface, with a subsequent formation of nonstoichiometric compounds with an oriented microcrystalline structure; and D - a layer formed as a result of the carbonization and hardening of the metal surface. The thickness of each layer depends on the lubricant composition and friction conditions and is approximately in the range of 10-50 nm for the B layer and 50-3500 nm for the C and D layers. The formation of the C and D layers occurs simultaneously. The final thickness of the D layer may be greater, equal to, or less than the thickness of the C layer depending on friction conditions. The better antifriction efficiency was observed to correspond to the thinner C and D layers allowing the localization of the tribo-chemical processes in smaller microvolumes. The thicker the layer of the material involved in friction process, the higher the friction coefficient.

CONCLUSIONS

1. Approximately five-to-eight percent energy savings are achievable by using energy-efficient industrial lubricants compared with current typical products.
2. Lubricant-related energy losses in a hydraulic system are load dependent and consist of churning and friction losses. Churning losses are mostly influenced by oil viscosity while friction losses largely depend on oil chemical composition. Boundary friction coefficient, high-temperature high-shear viscosity and pressure-viscosity coefficient are the most important oil properties defining oil energy efficiency.
3. The shear-thinning and viscoelastic effects of viscosity index-improved hydraulic fluids can be utilized for friction reduction and energy conservation. The best polymer for these objectives should significantly thicken oil; however it begins to thin at low shear rates corresponding to the oil

pumping process. In addition, viscosity index-improved hydraulic fluids can reduce leakage problems and improve low temperature fluid pumpability in hydraulic systems.

4. Lubricant rheology as well as friction energy losses for highly loaded contacts tend to be more affected by pressure-viscosity relationships than temperature or shear dependence. Since hydraulic fluids are usually subjected to high pressures, the pressure-viscosity behavior is one of the most significant fluid properties to be taken into account while selecting a hydraulic fluid. A minimal pressure-viscosity dependence, as shown for polyalphaolefins, is the most beneficial for energy conservation as well as for antiwear performance. Hydrogenated petroleum oils produced by severe hydrocracking processes are considerably cheaper than polyalphaolefins but show similar performance, and can be regarded as a good compromise between performance and cost.

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