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## EFFECT OF LUBRICANT PROPERTIES ON EFFICIENCY OF REFRIGERATION COMPRESSORS

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### ABSTRACT

Energy efficiency rating (EER) is an increasingly important performance characteristic for all refrigeration systems. Correct lubricant selection is not only critical to long term reliability in operation, but also can have a significant impact on overall energy efficiency. This paper reviews the mechanisms by which the lubricant can contribute to optimising overall energy efficiency, and presents friction data from lubricant bench tests, and energy efficiency data from appliance operation, in order to evaluate the relative contributions of these mechanisms and show how lubricant basefluid design, and appropriate formulation, can be used to optimise energy efficiency rating.

### INTRODUCTION

Following the Kyoto conference on climate change, energy efficiency is becoming an important performance characteristic for all refrigeration and air-conditioning systems. Performance or labelling requirements are increasingly defined by legislation, such as the European Refrigerator Labelling Directive, the US DOE Energy Efficiency standards, and the Japanese Energy Conservation Law.

Correct selection of compressor lubricant can have a significant impact on the energy efficiency performance of a system. Over recent years, the main focus of lubricant development has been to ensure reliability of operation in systems operating with HFCs and other alternative refrigerants required by implementation of the Montreal protocol. However, now that reliable lubricant technologies, eg highly optimised polyol esters (HOPOES) for HFC systems, are well established, the focus is changing to delivery of improved energy efficiency.

Use of lubricants leading to reduced energy consumption can obviously deliver a direct consumer and environmental benefit in terms of reduced cost of operation and greenhouse gas emission. In addition, such lubricants may be used by an OEM to permit other desirable changes to compressor or system engineering, for example in order to achieve reduced noise level or reduced component cost, without incurring an overall energy consumption penalty.

Lubricant selection can impact compressor energy efficiency in a number of ways. This paper will focus on the energy losses arising from frictional losses in lubricated contacts and lubricant transport processes. However, it is important to recognise that lubricant properties can also impact energy efficiency in other ways. For example, use of a lubricant with too low viscosity, or too high refrigerant gas solubility, can reduce volumetric efficiency in some compressor types. Lubricant properties can also affect evaporator surface heat transfer rates [1, 2]. Such effects are generally highly system or equipment specific, and will not be discussed further here.

### LUBRICANT TRANSPORT

In general, for all lubricated compressor types, mechanical energy is used to convey lubricant from a sump or reservoir to the lubricated contacts, at a flow rate sufficient to ensure adequate lubrication. The amount of energy required for maintaining the required lubricant flows is a function of compressor design, and to establish the absolute value for a given compressor would require either experimental measurement, or a detailed fluid dynamics simulation of the system. However, for the purposes of establishing the relationship between energy requirement and lubricant characteristics, it can be noted that to a first approximation, the flow patterns can generally be represented as some combination of agitation, flow through channels and flow over surfaces.

For agitation of a fluid reservoir by a rotating mechanical element, the power requirement  $P$  is given by;

$$P = k_1 n^2 d^3 \eta$$

where  $k_1$  is a system constant,  $n$  is the rotational velocity,  $d$  the diameter of the agitator and  $\eta$  the kinematic viscosity.

For flow in enclosed tubes, the force,  $F$ , needed to attain a specific average fluid velocity is given by;

$$F = 8\eta V/R^2$$

where  $V$  is the fluid velocity and  $R$  the diameter of the channel.

For flow over surfaces, the drag force,  $F$ , exerted is given by;

$$F = k_2 V \eta,$$

where  $k_2$  is a system constant.

All of these simple flow processes separately exhibit a linear dependence on the fluid kinematic viscosity. Therefore, to a first approximation, the overall energy requirement for lubricant transport will also be linearly proportional to the lubricant kinematic viscosity at the operating conditions, as long as the Reynolds number for the system is  $< 100$ , so that the flow remains fully laminar. (At the limit of low viscosity or high fluid flow rates flows may become turbulent and these relationships no longer apply) [3, 4, 5].

### FRICITION IN LUBRICATED CONTACTS

The magnitude of the friction coefficient in lubricated contacts depends on the lubrication regime. Figure 1 shows a schematic diagram of variation of friction coefficient with the factor  $\eta N/P$ , where  $N$  is the relative velocity, and  $P$  the applied load. This type of graph, generally referred to as a Stribeck curve, illustrates that three different lubrication regimes can be identified [6].

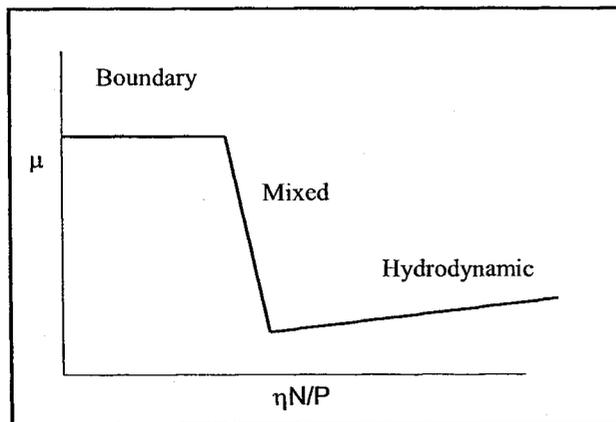


Figure 1. Schematic Stribeck curve showing lubrication regimes.

At high values of  $\eta N/P$ , corresponding to relatively high speed, lightly loaded contacts, the system is operating under conditions of full fluid film or hydrodynamic lubrication. The relative velocity of the surfaces is sufficient to entrain a lubricant film, which supports the load and is thicker than the characteristic roughness of the surfaces. Thus, the surfaces are fully separated by a film of lubricant, there is no asperity contact and no wear occurs.

Under hydrodynamic lubrication conditions, the only energy lost in the contact is that required overcoming viscous drag in the lubricant film. For a simple contact such as a plain bearing, the friction force  $F$  is approximately given by;

$$F = \eta \cdot A \cdot N / d$$

where  $A$  is the effective area of the contact, and  $d$  is the mean film thickness.

Thus, energy losses in hydrodynamic lubrication have a linear dependence on the fluid kinematic viscosity at the operating conditions, similar to that for lubricant transport processes [3, 6].

At low values of  $\eta N/P$ , corresponding to low speed, highly loaded contacts, the system is operating under conditions of boundary lubrication. Here, the relative velocity is insufficient to entrain a load-supporting hydrodynamic film. There is asperity contact between the surfaces, and the load is mainly carried through these solid contacts.

Under boundary lubrication a range of physical mechanisms which may contribute to frictional losses come into play. Although a partial fluid film is present which undergoes viscous shearing, this is only a minor contribution to the overall friction coefficient, and boundary friction coefficients generally show little dependence on lubricant viscosity. In the boundary regime, the major contribution to the frictional force is the energy required for deformation of contacting asperities [3, 7]. Lubricant basefluids or additives which form an adsorbed surface layer can modify the boundary friction coefficient [8]. In particular, components which form a coherent chemisorbed or physisorbed layer which deforms more readily than the underlying metal or metal oxide surface may reduce the boundary friction coefficients.

Lubricants having a high polarity or affinity for metal oxide surfaces, such as the polyol ester (POE) lubricants generally used with HFC refrigerant gases, have a greater tendency to form such adsorbed layers than less polar fluids such as mineral oils or synthetic hydrocarbons, and therefore have lower boundary friction coefficients. POE basefluids containing predominantly linear alkyl substituents (HOPOEs) can form a more coherently packed adsorbed film, and consequently show lower boundary friction coefficients, than those with branched alkyl substituents (HSPOEs) [9, 10]. This principle can be extended to use of components with longer linear alkyl chains and polar headgroups. The physical properties of such materials do not permit their use as lubricant basefluids, but they are widely used as friction modifying additives in a range of lubrication applications, particularly in low polarity basefluids such as hydrocarbons [11]. Delivering this effect in polar basefluids such as POEs is more challenging, as the polar basefluid can prevent surface adsorption of the additive by competing for surface adsorption sites.

At intermediate values of  $\eta N/P$ , the mixed lubrication regime, the load is mainly carried by the fluid film, but surface asperities may come into contact. The friction coefficient is intermediate between that characteristic of the boundary and hydrodynamic regimes. In the mixed lubrication regime, unlike the hydrodynamic regime, the friction coefficient increases as the lubricant viscosity decreases (at constant load and velocity) or the velocity decreases (at constant load and viscosity), due to an increasing contribution from the higher boundary friction coefficient.

## EXPERIMENTAL DETERMINATION OF FRICTION COEFFICIENTS

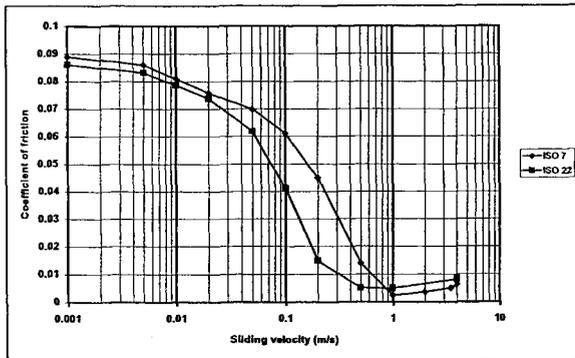
To illustrate the frictional loss behaviour of lubricated contacts across the range of lubrication regimes, this section reports friction coefficient data obtained using a conforming-pin-on-ring tribometer of inhouse design.

In this device, a softer metal pin is loaded against the edge of a rotating hardened steel ring. The pin wears as the system is run in until the pin surface forms a conforming contact against the edge of the ring. The relative hardnesses of the materials are selected such that wear is essentially confined to the surface of the pin and running in can be completed without scoring or scuffing damage to either surface.

Once the pin has been run in, the pin load is adjusted to an appropriate value, and the rotational velocity of the ring is varied through a programmed range of values from 0.001 to 4 m/s. For most lubricants of interest, this velocity range encompasses both hydrodynamic and boundary lubrication regimes. At each velocity stage, the velocity is maintained constant for 15 minutes or 1500 revolutions, while the frictional force is recorded via a torque transducer mounted on the shaft. A correction for parasitic friction effects is made by subtracting the results of a reference run at zero load, and the friction coefficient calculated. The rotational velocity program is repeated until reproducible results are obtained on at least 3 consecutive runs. The bulk oil temperature is controlled throughout the cycle. All data reported here were recorded at  $40 \pm 2^\circ\text{C}$ .

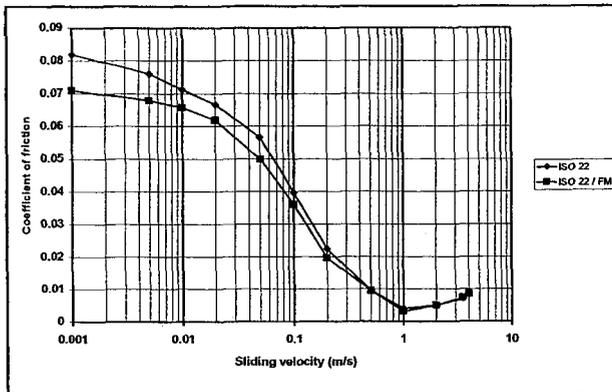
Figure 2 compares the run in friction coefficient versus velocity curves for an ISO 22 and an ISO 7 HOPOE lubricant at the same load. It is evident from the figure that, as expected, the ISO 7 fluid shows the transition to mixed and boundary lubrication occurring at higher velocity than the ISO 22. If these friction coefficient data are

replotted against the product of velocity and viscosity (as the loads are identical this is qualitatively equivalent to plotting against the  $\eta N/P$  parameter), the two traces are nearly coincident.



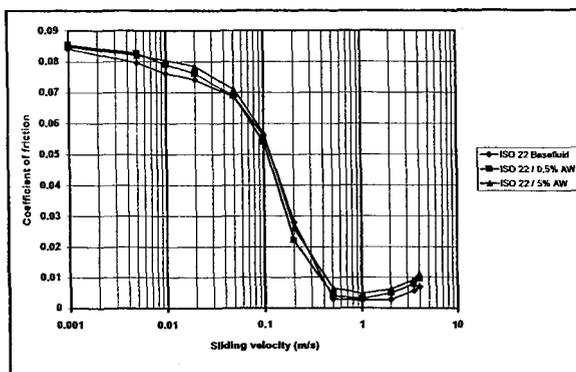
**Figure 2. Comparison of friction coefficients for ISO 22 and ISO 7 HOPOE lubricants.**

As noted above, boundary friction coefficients can be modified by the action of lubricant additives. Figure 3 illustrates the effect of an organic friction modifying additive in an ISO 22 HOPOE. It is evident that the additive slightly reduces the boundary friction coefficient, but has little or no effect on the location of the mixed lubrication transition, or on the friction coefficient under hydrodynamic conditions.



**Figure 3. Effect of friction modifier on friction coefficient for ISO 22 HOPOE.**

In many cases, for systems which operate fully or partially in the boundary regime, antiwear additives are used in the lubricant. It is sometimes suggested that such additives, which operate by chemical modification of the surfaces, may have an effect on boundary friction coefficients, and thus on energy efficiency. However, friction and wear are essentially independent, albeit related, phenomena under boundary conditions. Thus Figure 4 compares the friction versus velocity curves for an ISO 22 POE containing a highly active antiwear additive at 0, 0.5 and 5% concentration. The graph illustrates that this additive, which is highly effective at limiting wear, has essentially no effect on the boundary friction coefficient.

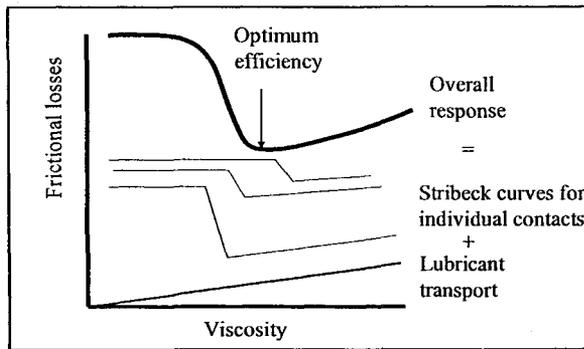


**Figure 4. Effect of antiwear additive on friction coefficient. ISO 22 HOPOE.**

There has also been some discussion in the literature about the extent, if any, to which the friction and wear behaviour of refrigeration compressor systems is affected by the presence of the refrigerant gas [12, 13]. The effect of replacement of air by an atmosphere of HFC R134a was investigated and found to have no significant effect on the frictional properties determined in this test.

### OPTIMISING ENERGY EFFICIENCY

The results in the previous section illustrate the basic trends which will govern the behaviour at each of the lubricated contacts in a machine such as a refrigeration compressor. In general, any compressor design will contain a number of such contacts, all lubricated by the same fluid, but running at different loads and sliding velocities, and therefore operating at different positions on the Stribeck curve. For the machine as a whole, the power required to overcome frictional losses will be represented by the sum of the contributions of each of the individual contacts, plus the contribution from lubricant transport, and will therefore show a U-shaped dependence when plotted against the lubricant viscosity. This is represented schematically in figure 5. The challenge for the design engineer is to select a lubricant formulation so as to position the system as near as possible to the minimum of this curve, which represents the optimum position for energy efficiency.



**Figure 5. Schematic diagram of overall frictional losses as a function of lubricant viscosity**

It is normally necessary to operate to the right hand side of the optimum energy efficiency position, as the exact minimum of the curve will represent conditions under which the most highly loaded or lowest velocity contacts are in the boundary or mixed lubrication regime, and wear is occurring. These most sensitive contacts must be protected by use of highly optimised basefluids or antiwear additives in the lubricant formulation in order to permit a closer approach to the energy efficiency optimum.

In summary then, it is predicted that energy efficiency performance of most compressor types will be dominated by losses arising from lubricant transport and hydrodynamic friction losses, and that the most significant impact on energy efficiency will be obtained by reducing the lubricant viscosity.

Use of antiwear additives may be necessary to permit use of lower viscosity fluids, but do not themselves impact on energy efficiency. Use of friction modifier additives will contribute to energy efficiency only if a significant proportion of power losses occur at boundary lubricated contacts. (It should be noted that even if a contact operates in the hydrodynamic regime at standard operating conditions, it must pass temporarily through the boundary regime at start up and shutdown. This is particularly important at startup, when the contacts may be lubricant-starved due to drainage occurring during while the equipment is stationary. Antiwear or friction modifying additives may be required in a lubricant formulation simply to limit wear and surface damage occurring at this stage).

### APPLIANCE ENERGY EFFICIENCY RESULTS

In order to test the expected relationship between lubricant viscosity and energy consumption, refrigerator efficiency tests based on the conditions of ISO 5155 were carried out on standard commercial refrigerator and freezer cabinets, retrofitted with a range of HOPOE lubricants spanning the viscosity range from ISO 22 to ISO 3.

The appliances used for this test were standard commercially available units purchased in the UK, operating on R134a refrigerant gas.

In each case, prior to testing, refrigerant was removed via the service port of the compressor, followed by removal of the original fill lubricant from the sump pipe. The discharged lubricant was weighed. The system was evacuated and recharged with the same weight of the first test lubricant, then re-evacuated and charged with the required amount of refrigerant gas, as vapour. The cabinet was then restarted. This procedure was repeated for each of the series of test lubricants. The actual viscosity of each test lubricant was checked after discharge from the appliance.

The cabinets were filled with synthetic food packages to simulate an average loading and operated under controlled conditions over a 14 day period for each test lubricant. Ambient temperature was controlled, and energy consumption of each unit was recorded.

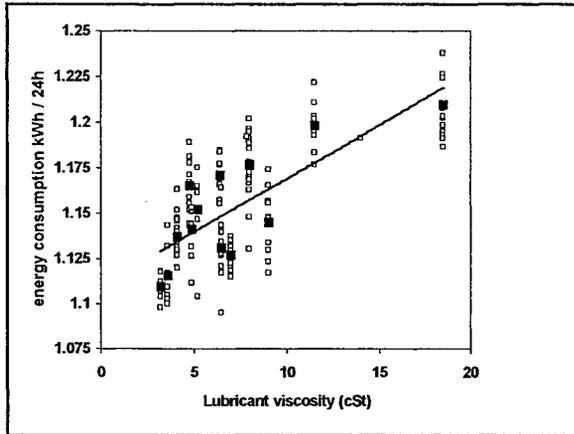


Figure 6. Power consumption versus lubricant viscosity. Refrigerator.

Figure 6 shows the daily power consumption of the refrigerator unit plotted against lubricant viscosity. For each test lubricant viscosity, the solid point represents the average value for the run and the hollow points represent the individual daily readings. Separately, the dependence of daily energy consumption on minor variations of ambient temperature was determined, and a good correlation was found between the spread of daily values, and the slight fluctuation in ambient temperature over the test. This underlines the critical importance of accurate control of ambient temperature in this type of test.

Figure 7 shows the same data for the freezer.

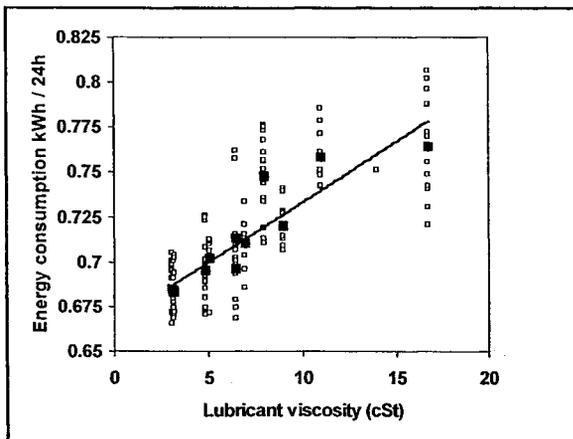


Figure 7. Power consumption versus lubricant viscosity. Freezer.

It is apparent in both cases that reducing the lubricant viscosity leads to a significant reduction in energy consumption, corresponding to an approximately 2% drop in energy consumption for each ISO grade step decrease in lubricant viscosity for the refrigerator.

It can therefore be concluded that the operating conditions for these appliances lie well over to the right hand side of the friction loss versus lubricant viscosity schematic diagram shown in Figure 5, and therefore that the overall power losses are dominated by lubricant transport and viscous drag across the range of lubricant viscosities tested here.

#### CONCLUSIONS.

From the data presented above it is clear that appropriate selection of lubricant can have a significant impact on the overall efficiency of operation of domestic appliances, and other refrigeration and air conditioning systems. In most cases, the most significant lubricant property is the viscosity, which impacts on the energy required for lubricant transport and overcoming viscous drag in hydrodynamically lubricated contacts.

Operating data shows that under standard operating conditions, appliances can function using lubricants having viscosities significantly lower than those in typical current commercial use, and that this can deliver a significant reduction in energy consumption. However, the requirement to demonstrate reliability over the exacting range of test conditions normally used during the process of lubricant approval will mean that it will be some time before fluids of this viscosity are in routine use.

The contribution of friction in boundary lubrication to overall energy efficiency is likely to be significantly lower for most compressor types, but use of friction modifying or antiwear additives may be beneficial in limiting wear at the most sensitive contacts and thereby permitting the use of a lower viscosity lubricant in the system.

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