1992

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THE EFFECT OF REFRIGERANTS ON THE LUBRICATION OF ROLLING ELEMENT BEARINGS USED IN SCREW COMPRESSORS

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ABSTRACT

The formation of sufficiently thick lubricant films at the highly stressed contacts within a rolling bearing is a prerequisite for reliable operation. These films are a function of the lubricant's physical parameters: viscosity, η and pressure-viscosity coefficient, α. In the screw compressor bearing application, either or both parameters may be adversely affected by the presence of refrigerant. Experimental measurements of η and α are presented for oil-rich solutions of R22 and a naphthenic oil and of R134a and ester oils. These measurements, obtained with a rotating cylinder viscometer, are supported by measurements of film thickness on an optical disc test rig and of effective lubricant viscosity on a dynamic bearing test rig.

NOTATION

h_o	Central film thickness
k	Ellipticity parameter
G	Dimensionless materials parameter
H_c	Dimensionless central film thickness
R_x	Effective radius of surfaces in direction of motion (L)
U	Dimensionless speed parameter
u	Rolling speed (T^{-1})
W	Dimensionless load parameter
α	Pressure-viscosity coefficient ($LM^{-1}T^{-2}$)
η	Viscosity, dynamic ($MT^{-1}L^{-1}$)
υ	Viscosity, kinematic (L^2T^{-1})

INTRODUCTION

On screw compressors used in refrigeration and air conditioning equipment, the bearing environ-ment is governed by the refrigeration cycle, temperatures and pressures are determined by compressor operating characteristics and the lubricant is invariably a solution or mixture of oil and refrigerant. By necessity the latter is a volatile fluid with an extremely low viscosity and therefore its presence in the lubricating oil may be expected to influence bearing performance and reliability.

For oil lubrication the thickness of films formed at the rolling contacts within a bearing are a function of the oil's physical parameters - viscosity, η, and pressure viscosity coefficient, α. Thus, in a lubricant system in which the oil can become contaminated with refrigerant, a first requirement for estimating the effect on bearing performance is to know the proportion of refrigerant present at the bearing position and its effect on both η and α. In principle, this information should be sufficient to

design out unacceptable bearing lubrication conditions.

The proportion of refrigerant present in the lubricating oil is a complex function of the lubricant system dynamics. A recirculating supply is normally used in which the lubricant undergoes different temperatures and pressures as it travels around the cycle. Since the solubility of the refrigerant in oil is a function of temperature and pressure, its concentration also varies around the cycle and at any one point will depend not just on the above parameters but also on factors such as flow rates, energy trans-fer rates, absorption and desorption rates. Thus, from a practical design standpoint, bearing lubrication cannot be considered in isolation from the lubricant supply. Indeed, observation of the proportion of refrigerant in the lubricant reaching the bearings and the dynamic effects of either refrigerant boiling off or refrigerant going into solution can only be made on a test apparatus capable of simulating in-service conditions. However, such a rig cannot provide measurements of both η and α as well; a simpler laboratory viscometer adapated to accommodate solutions of refrigerant and oil is also required.

These rate-dependent effects and the dependence of α upon refrigerant concentration are aspects of lubrication specifically relevant to rolling bearings but which have not been considered in past investigations. Past research has concentrated largely upon the physical chemistry and thermodynamic behaviour of refrigerant-oil solutions, although the effect of refrigerant concentration on oil viscosity has been determined under essentially static conditions with a laboratory-type apparatus [1-5]. The results of this work are presented in the Refrigeration Handbook [6] in the form of a design guide. It is mostly relevant to the hydrodynamic lubrication of machine elements.

In this paper, an experimental investigation into rolling bearing lubrication in the presence of refrigerants is described. It represents a first examination of the problem and therefore concentrates upon the development of experimental techniques for measuring η and α for solutions of refrigerant and oil and for observing the lubricant films developed by a ball bearing operating as part of a dynamic refrigeration cycle. Experimental measurements are presented for R22 and a naphthenic oil and for R134a and ester oils.

VISCOMETER FOR MEASUREMENT OF REFRIGERANT-OIL PROPERTIES, η AND α

This apparatus was originally designed by Isaksson [7] to measure rheological properties of water-based hydraulic fluids at pressures up to 20 MPa. It consists mainly of one rotating inner cylinder and an outer cylinder suspended in two deep groove ball bearings, Fig. 1. The outer cylinder is radially fixed to the rigid frame through a thin plate, strain gauged for torque measurements. The gap between the cylinders is 0.178 mm and the mean radius of the gap is 45.048 mm. The viscometer is placed in a pressure chamber which has a pressure sealed shaft packing.

The shear stress, shear rate relationship for the test fluid can be measured by giving the inner cylinder a constant angular speed and measuring the torque on the outer cylinder using the strain gauged plate. The signal from the strain gauge is continuously recorded with a pen recorder.

The viscometer, Fig. 2, is pressurized by an arrangement of two hydraulic cylinders. The pressure chamber is connected to both sides of a double acting cylinder with a through piston rod. The piston rod is connected to a second cylinder driven by a hydraulic pump. The pressure of the test fluid can be adjusted with a pressure regulator.

Liquid refrigerant is measured with a filling device originally designed for servicing small refrigeration plants. The filling device consists of a pressure vessel which is equipped with a level gauge and a heating device, which make it possible to increase the pressure in the vessel.

VISCOSITY AND PRESSURE-VISCOSITY MEASUREMENTS

Viscosity and pressure-viscosity measurements were carried out for a pure ester oil (referred to as oil A, Table 1) and for mixtures of 14%, 23.6% and 31.9% by weight dissolved R134a in the oil. Measurements were also obtained for a 10% concentration of R22 in a mineral oil. The concentrations were determined by weighing the oil container and the filling device before and after

addition of oil or refrigerant.

Preparation of Oil-Refrigerant Mixture

The arrangement for preparing the oil-refrigerant mixtures is shown in Fig. 2. Before the system was filled with oil it was evacuated down to a pressure below 1 MPa to ensure that possible solvent residues were removed from the system. After that the vacuum pump was disconnected from the system, valves V5 and V6 were opened so that the atmospheric pressure forced the test oil from the oil container into the system. In order to completely fill the system, the piston was driven back and forth until no air bubbles appeared in the oil container. Thus, when refrigerant was added to the system, an amount of oil was displaced. The amount of refrigerant in the oil could be determined by heating a sample of the mixture to a temperature of 50 °C whilst vigorously stirring so that the dissolved refrigerant was caused to boil off. The weight of discharged refrigerant was determined from the weight of pure oil left and the prior concentration of refrigerant in the system. After refrigerant had been added to the system the viscometer was started and the fluids were mixed together by pumping them from the bottom to the top of the viscometer with the hydraulic cylinders and by-pass valve V1. The mixture was homogenised when the reading from the strain gauge was stable.

Measurement Procedure and Evaluation of Data

The angular speed was adjusted to give the desired shear rate, then the viscometer was stopped and the pen recorder started, after which the pressure was adjusted and the viscometer was started again. When the reading on the strain gauge bridge had stabilized, the pressure, speed and strain were noted by the pen recorder. The pressure was increased in steps of 5 MPa from vapour pressure to 20 MPa after which the viscosity calculated from the torque and angular speed was plotted against pressure. For R134a and oil A, and for R22 and naphthenic oil, the pressure-viscosity coefficient was calculated by a least squares fit, as seen in Fig. 3. For R134a and oil A, ambient viscosity and pressure-viscosity coefficient are plotted against the concentration of refrigerant in Fig. 4.

To illustrate how the film thickness is theoretically affected by the decrease in viscosity and pressure-viscosity coefficient, an oil film thickness coefficient is shown plotted against pressure in Fig. 5. The oil film thickness coefficient is the product of the viscosity and the pressure-viscosity coefficient using the same exponents as in the Hamrock and Dowson formula [10] for the central film thickness, where $h \propto \eta^{0.67} * \alpha^{0.53}$.

OPTICAL FILM THICKNESS MEASUREMENTS

Optical interferometry was also used to evaluate film thickness with and without refrigerant. This method was first presented by Cameron and Gohar [8] in 1966. In Fig. 6 the main parts of the test rig are shown. A steel ball is loaded against a chromium covered glass plate using a lever and a spring. The contact is illuminated with white light which is reflected by the 200 Å chromium layer on the lower side of the glass disc and by the surface of the ball, thus causing constructive interference. This is observed as different colours in the contact and by knowing the refractive index of the lubricant can be used to evaluate the topography of the contact area. A description of the rig can be found in Åström et al [9]. The surface roughness of the ball and glass plate was 0.04 µm rms and 0.009 µm rms, respectively. Thus it was ensured that no asperity interaction could take place and a full film situation prevailed.

The disc was driven by an electric motor via a chain and a variable speed drive. Optical counters measured the speeds of both disc and ball and pure rolling prevailed. Since the oil-refrigerant solution was fed to the contact under pressure, the ball retainer was designed as shown in Fig. 7. An O-ring was used as a seal between the glass disc and cap mounted on top of the retainer. The shaft supporting the ball was mounted in two self-aligning ball bearings in a type of cradle arrangement, and a sealing cage together with a radial seal was fitted on the shaft. Thus, the retainer was completely sealed off and could be filled to a desired volume with the pressurized oil-refrigerant solution which was then drawn into the contact by the rotation of the ball. The load was applied to the outer bearing using a spring-loading device. Since the inner bearing was of a self-aligning type, the load was transmitted to the ball-disc contact. The applied load was measured by a load cell at the end of the *

spring. Temperature and pressure in the ball retainer were registered using a Pt-100 thermoresistance and a pressure gauge.

The contact was illuminated with white light through a fibre optics bundle and a 90° glass prism. A video camera (VHS) with a shutter speed of 1/1000 second and a microscope were used to record the colours in the contact and the readings on the gauges.

Oil-refrigerant solution was supplied to the ball retainer via a hose from a steel cylinder, Fig. 8, which was pressurized with nitrogen. The pressure was always kept above the vapour pressure of the oil-refrigerant solution - in these tests, 1.0 MPa. The steel cylinder had glass gables and was also used to determine the refractive index of the oil-refrigerant solution. A He-Ne laser was used to send a beam of light through the filled cylinder, and the distance D (Fig. 8) was then used to calculate the refractive index. The system was calibrated using pure oil with a refractive index measured by an Abbe refrac-tometer. For ester oil A (Table 1), the refractive index was 1.457 and for oil A+9% by weight R134a, 1.42. The tests were run at room temperature, 21 °C, and a maximum Hertzian pressure of 587 MPa.

Results and discussion

Figs. 9-11 show the central film thicknesses as a function of rolling speed. They are also compared to the theoretical values calculated from Hamrock and Dowson [10]:

$$h_c = H_c R_x = 2.69\ R_x\ U^{0.67}\ G^{0.53} W^{-0.067}(1-0.61e^{-0.73k})$$

In the above equation, viscosity and viscosity-pressure coefficients were taken from measurements in the viscometer, Fig. 1. For pure ester oil A: $\eta = 0.34$ Pas, $\alpha = 2.4*10^{-8} Pa^{-1}$, and for ester oil A+9% R134a: $\eta = 0.27$ Pas, $\alpha = 2.35*10^{-8}$ Pa^{-1}.

Fig. 9 shows that the experimental values for pure ester oil A indicate a thicker lubricating film than given by theory. The difference is about 20% and agrees very well with the results of other investigators, e.g. Koye and Winer [11], who found a 23% difference in minimum film thickness for mineral oils. When 9% R134a is added, the film thickness drops to approximately 75% of the theoretical value, see Fig. 10. It must be remembered that the theoretical film thickness itself is reduced owing to the change of η and α. Theoretically, the change in film thickness should be:

$$\frac{h_{c9}}{h_{c0}} = \left(\frac{0.27}{0.34}\right)^{0.67}\left(\frac{2.35\times10^{-8}}{2.4\times10^{-8}}\right)^{0.53} = 0.85$$

The real drop in film thickness, however, was much larger, down to about 55%; see Fig. 11. Oil A was known to have poor mixibility with R134a and under the test conditions, two liquid phases, one oil-rich, the other, refrigerant-rich, would have existed.

In this situation, it is possible that differences in homogeneity occurred between the fluid in the viscometer and that in the optical test rig. Thus, the values of η and α taken from viscometer measure-ments may not have characterised the latter fluid accurately. This observation requires further investi-gation to determine whether the effect occurs over a full range of oil-refrigerant concentrations, for different shear rates and heating effects.

BEARING TEST APPARATUS

First objectives were to determine the effective viscosity of refrigerant-oil solutions experienced by rolling bearings as part of a refrigeration cycle, to develop appropriate measurement techniques and to observe any rate-dependent effects produced by refrigerant concentrations varying around the cycle. These requirements governed the overall configuration of the test apparatus which comprised a bearing test head. a refrigeration test stand and instrumentation for monitoring oil films in the test bearing, Fig. 12.

One of the compromises of the rig was the mode of bearing lubrication. A bath lubrication system was used in which the loaded rolling elements of the test bearing were immersed and for which the refrigerant-oil composition could be determined by drawing off fluid samples into an evacuated bottle. This method of lubrication is untypical of screw compressor bearings but it simplifies the measurement technique and, at least for an initial examination, provides a bearing environment sufficiently close to reality to enable dynamic effects to be observed.

The method of monitoring bearing oil films which was used was a development of a standard technique published in 1982 [12] and only required the rig to have a continuously variable speed drive and electrical insulation of the bearing from its environment. However, it was important to be able to perform reference measurements with a known oil and this necessitated a means of isolating the bearing test head from the refrigeration test stand - a feature which also had the advantage of enabling measure-ments to be performed under static conditions.
The main features of the test apparatus are described in the following sub-sections.

Overall Arrangement

The overall arrangement is shown diagrammatically in Fig. 12. The main elements are the bearing test head and the refrigeration test stand comprising: compressor, oil separator, oil-refrigerant mixing circuit and liquid-gas refrigerant mixing circuit. Liquid supplied to the bearing test head could be varied in composition and flow rate, from nominally 100% oil to 100% refrigerant, with flows to a maximum of approximately 4 l/min. The set flow was continuously monitored by the flow meters shown in Fig. 12.

Refrigerant gas was also supplied to the bearing test head but via a separate feed. A test head bypass was taken from this line direct to the compressor suction line in order to avoid high flow impedances and hence maintain good thermal stability of the system under a full range of operating conditions. Fluid composition in both the test head and the suction line could be determined by sampling. Pressures and temperatures could also be measured at these two points.

Test Head

The test head and bearing arrangement is shown in Fig. 13. Important features are a magnetic drive coupling located at one end of the shaft which permits the head to be hermetically sealed, a large viewing window for observation of the lubricating fluid, and a simple bearing arrangement in which a central test bearing is electrically isolated from its surroundings and made part of an electrical circuit. Both the test bearing and the two outboard slave bearings were 6204 deep groove ball bearings. All were radially loaded through a spring attached to the test bearing housing and reacted against the main housing.

The arrangement of the bearing lubrication is shown in Fig. 14. Liquid and gas feeds from the compressor stand enter the bearing cavity above the bearing axis whilst fluid level is controlled by a drain positioned on the axis. Under normal conditions the loaded rolling elements, which are below the bearing centre, are completely immersed in fluid. A sampling port is situated well below the fluid level.

Bearing and lubrication bath temperatures were measured at the points indicated in Fig. 14. Thermocouple connections and the electrical connections across the bearing were made through hermetically sealed plugs mounted in the test head wall. Ambient pressure in the head was monitored with a gauge.

Measurement of Lubricant Films

Of primary interest is the effective viscosity of oil-refrigerant solutions. Since attention is confined to oil-rich solutions (i.e., less than 20% refrigerant by weight) for which the pressure-viscosity coefficient has been shown to be close to that of oil only, measurement of lubricant film thickness generated at rolling element-raceway contacts would enable effective viscosity to be deduced. The film thickness measurement technique developed is based upon capacitance and follows that reported in [12]. Particular attention was given to the problem posed by refrigerant and oil having

different dialectric constants and a procedure in which this effect could be minimised was adopted.

EXPERIMENTAL MEASUREMENTS

Dynamic Effects

Being able to run the bearing test head with and without the refrigeration test stand in circuit enabled two basic types of measurement to be performed. With the test head isolated, different oil-refrigerant solutions could be obtained by first adding oil and then by filling the head to a known pressure with refrigerant gas. Under these conditions, measurements could be obtained close to static Pressure-Temperature-Solubility (PTS) equilibria corresponding to ambient temperature. With the refrigeration stand in circuit the measurements were dynamic although at nominally constant pressure corresponding to compressor suction pressure.

A logical first step to the experimental work was to observe how refrigerant concentration varied at positions in the circuit local to the test head. Fig. 15 shows examples of measurements taken from the lubricant bath and the return line plotted against the refrigerant concentration set in the delivery line, for R22 and a naphthenic oil. The results confirm that large variations in refrigerant concentration occur; furthermore, the variations were observed to be dependent upon flow rates and temperatures. It was therefore important to be able to determine the composition of the lubriant in the bath as this provided the lubrication to the test bearing. How this composition related to set flow, temperatures and pressures was not of central importance to the work reported here and is not considered further. In general, the composition of the fluid actually lubricating the bearing may be expected to be a function of the lubricating system in addition to operating conditions and is therefore a point to be examined in relation to practical designs of lubrication systems.

PTS equilibrium data is often used to estimate refrigerant concentrations in the lubricating oil. Fig. 16 compares the concentration of refrigerant in the lubricant bath with that taken from PTS data. In this particular case, the lubricating fluid was oil-rich relative to the PTS equilibrium and would therefore be of a higher viscosity. The implication is that actual lubricant film thicknesses developed in the bearing would be greater than theoretically estimated.

R22 and Naphthenic Oil

The effective viscosity of R22 and Naphthenic oil solutions measured under static equilibrium conditions is shown in Fig. 17. These results correspond to a constant ambient temperature of 24 °C and show log(viscosity) to reduce linearly with increasing refrigerant content.

Equilibrium PTS conditions took an appreciable time to reach, particularly for the higher concentrations of refrigerant. In the interval between initial charging with refrigerant and stable equilibrium, viscosity measurements were time-dependent. Viscosity appeared to be very low immediately after charging but then increased slowly with time until equilibrium was achieved.
Fig. 17 shows the minimum viscosity recorded at 10 and 12% refrigerant concentrations as well as the stable value recorded at 10% concentration. The results were contrary to expectations. Since refrigerant was added in the form of gas then the concentration in the lubricant bath would increase with time and hence viscosity would be expected to reduce. A possible explanation is that the refrigerant initially mixed as a gas and then slowly transformed to liquid. Gas bubbles in solution would increase compress-ibility of the lubricant and reduce shear strength, which in a bearing would result in reduced oil film thickness, in turn appearing as a reduced effective viscosity in these measurements.

However, the phenomenon did appear consistent with the results of Albright and Lawyer [2] who found that the viscosity of a refrigerant-oil solution reduced with increasing vapour pressure. In the measurements described above, pressure in the test head was initially high after charging and decreased slowly to an equilibrium value.

The dynamic experiments were performed at essentially constant pressure - the suction pressure of the compressor. Thus, with increasing refrigerant concentration, bearing test head temperatures

decreased to give an effective viscosity-refrigerant relationship with a low slope, Fig. 18. To bring this back to the same basis as the curve of Fig. 17, a correction was applied to account for the increase in the viscosity of the oil with reducing temperature. The correction was approximate, based on the assumption that the viscosity reduction produced by the refrigerant was independent of initial oil viscosity. This amounts to factoring the measured viscosity by the ratio of the viscosity of the oil at 24 °C to that at the test temperature.

Both dynamic and static results are shown in Fig. 19 and compared to published data taken from [1]. The results are presented in terms of the H parameter defined by Bell and Sharp [13] where H is a function of viscosity given by:

$$H = 870 * [\text{Log}(\text{Log}(\upsilon + 0.8))] + 154$$

and υ = viscosity in mm^2/s.

Fig. 19 shows that static and dynamic measurements are consistent in terms of the reduction in H due to the presence of refrigerant. Agreement with published data is also reasonable, the difference corresponding to approximately a one percent error in measurement of refrigerant concentration. This error is within the accuracy to which the composition of test head fluid samples could be determined.

R134a and Polyol Ester Oil

Two polyol ester oils were evaluated under static and dynamic conditions on the bearing test rig and viscosity measurements were also obtained on the rotating cylinder viscometer. The oil properties are summarised in Table 1. Fig. 20 shows the viscosity-weight% refrigerant relationship for the two oils. For oil B, static and dynamic tests on the bearing rig show a consistent relationship between viscosity reduction and weight% refrigerant. The level of reduction is low compared to that found for R22 and Naphthenic oil. Furthermore, oil B exhibited very rapid adsorption of R134a in the static tests, with equilibrium being reached in just a few minutes or so. This oil was known to have good miscibility characteristics.

TABLE 1: Properties of ester oil

	LUBRICANT	
	A	B
Type	PE Modified	PE Modified
Viscosity (mm^2/s): - 40°C - 100°C	 123.9 14.7	 64.1 8.9
Viscosity Index	120	113
Pour Point (°C)	-35	-43

In contrast, oil A exhibited extremely long adsorption times to the point where reliable measure-ments could not be obtained over a full range of refrigerant concentrations. For all refrigerant concentrations the static tests produced a time-dependent viscosity which was characterised by a low value immediately after adding R134a and a higher equilibrium value. Above 5% concentration of R134a, the time to reach an equilibrium was several days and reliable measurements were masked by interim fluctuations in ambient temperatures. In contrast to oil B, this oil was known to have poor miscibility with R134a and non-homogeneity of the test fluid may be a possible explanation for the low values of viscosity measured.

The dynamic tests were also limited in range. Weight% refrigerant below 29% could not be achieved, a phenomenon attributed to the oil separator system being unsuitable for this lubricant. The same lubricant was used in the viscometer measurements reported in Section 3 and this data is also

shown in Fig. 20. A comparison between the two methods of measurement suggests that the viscometer records slightly higher values of viscosity than the bearing test rig for a given concentration of refrigerant. The vigorous stirring action inherent in the viscometer measurement method also made it less susceptible to the stability problems encountered with the bearing test rig.

DISCUSSION

The work described in this paper is based on the hypothesis that lubricant films generated in rolling element bearings operating in the presence of refrigerants are dependent upon the same viscosity and pressure-viscosity mechanisms as for oil only. At least for oil-rich solutions is the hypothesis supported by the measurements of α shown in Fig. 4 together with the measurements of effective lubricant vis-cosity shown in Figs. 17 and 20. The possibility of solid film deposition was considered during the course of bearing tests but could be dismissed as some lubricants were known to be free of additives, whilst for others rolling surfaces were examined following short duration tests.

For the lubricants tested, pressure-viscosity coefficient, α, generally decreased in value with increasing refrigerant content. In the oil-rich operating regime, applicable to oil separator lubrication systems (i.e., less than 20% by weight refrigerant), the reduction in α was small compared to the reduction in base viscosity and for practical purposes may be neglected in film thickness calculations. Further work is necessary to determine α at higher refrigerant concentrations and for more types of lubricant.

The reasonable agreement between the effective viscosity measurements for R22 and Naphthenic oil and published data verifies the measurement technique used on the bearing test apparatus, and that the design data published in the 1986 Refrigerant Handbook is applicable to the present test arrangement. A point of concern, however, is that in order to use this data, it is necessary to know the refrigerant concentration local to the bearing. For the lubrication system used, the latter was quite different to concentrations set in the delivery line or in the return line and had no simple correlation with either. Further work is necessary to examine whether this is the case on more practical lubrication systems and, if so, to develop a model for predicting refrigerant concentration from known system parameters.

The new refrigerant R134a used with polyolester oils appears to give a much lower reduction in lubricant viscosity than R22 and Naphthenic oils. The early results reported here are therefore encouraging from the point of view of generating full lubricant films in rolling element bearings, but again further work is necessary to identify the characteristics of other candidate oils.

CONCLUSIONS

The mechanism of lubrication of rolling element bearings operating in the presence of oil-refrigerant solutions is similar to that for oil only. However, refrigerant has the effect of reducing both viscosity and pressure-viscosity coefficients of the oil.

For oil-rich solutions (less than 20% refrigerant by weight), the effect of refrigerant upon α is small compared with the effect upon η. Thus, published data giving η as a function of refrigerant R22 concentration in mineral oil may be applied to rolling bearings.

The concentration of refrigerant in oil local to the bearing may be quite different to that delivered and there is no simple way of estimating this from known system parameters. This will currently limit the accuracy to which lubricant film thicknesses may be calculated in a practical bearing application.

ACKNOWLEDGEMENTS

The authors would like to thank the Managing Director of SKF Engineering & Research Centre, The Netherlands, for permission to publish this paper, the Trane Company, USA, for assistance with the refrigerant test stand, Art Butterworth of Trane for practical advice and Professor A. Cameron for many helpful discussions.

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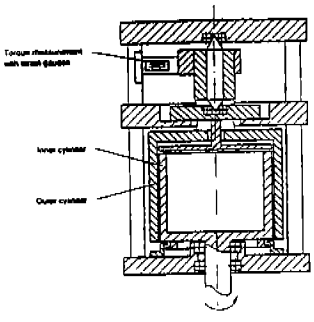

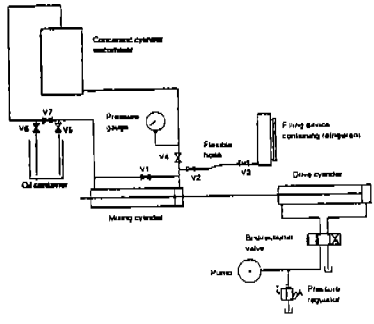

Fig. 2 Arrangement for measuring the viscosity

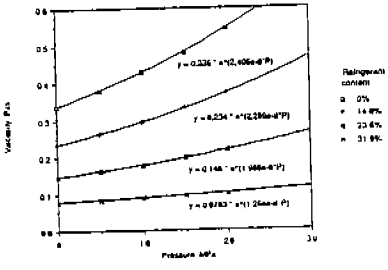

Fig. 3a Viscosity-pressure relationship for polyolester oil A and refrigerant R134a at different concentrations

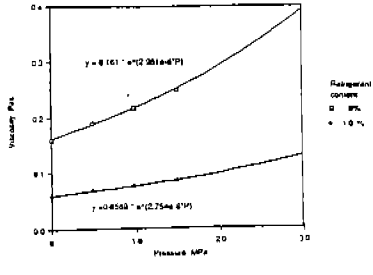

Fig. 3b Viscosity-pressure relationship of mineral oil and refrigerant R22 at 10% concentration

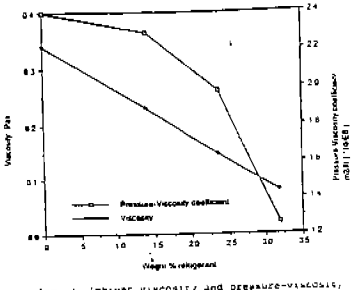

Fig. 4 Ambient viscosity and pressure-viscosity coefficient versus concentration of refrigerant for oil A and R134a.

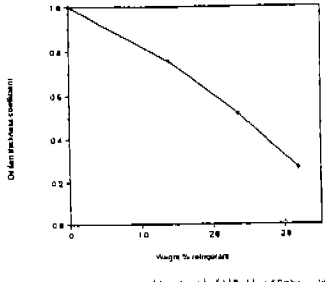

Fig. 5 Ambient oil film thickness coefficient versus concentration of refrigerant for polyolester oil A and R134a

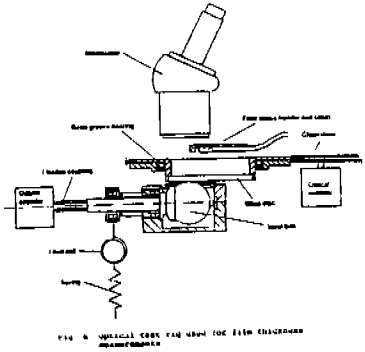

Fig. 6 Optical test rig used for film thickness measurements

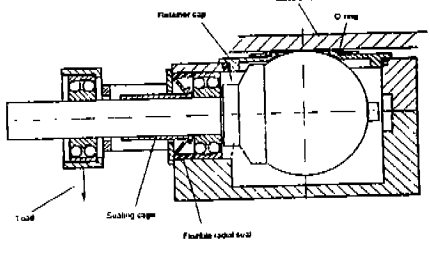

Fig. 7 Design of ball retainer

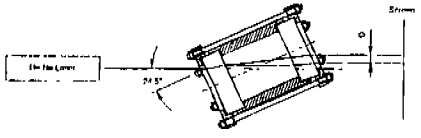

Fig. 8 Cylinder used in pressure viscosity determination of refractive index

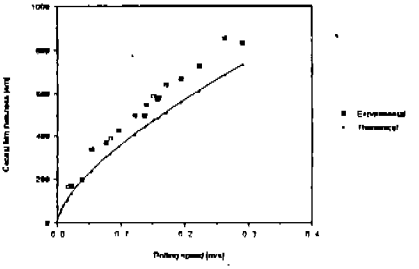

Fig. 9 Central film thickness versus rolling speed for oils I, II and A

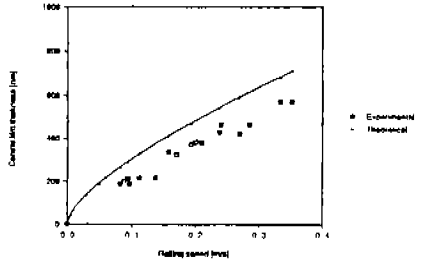

Fig. 10 Central film thickness versus rolling speed for oils I, II and A at 80 °C

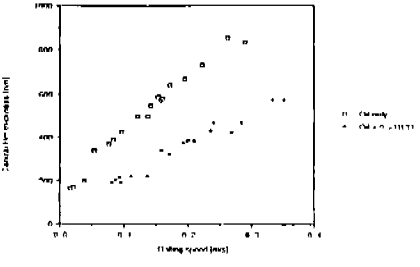

Fig. 11 Central film thickness versus rolling speed for polybutene oil A at 20, 40 and 60 °C

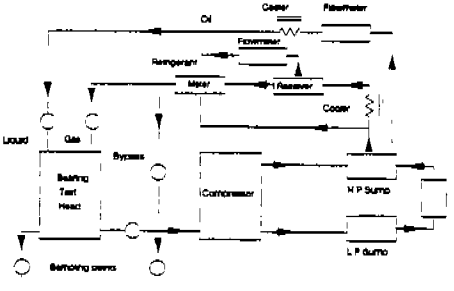

Fig. 12 Bearing test rig arrangement.

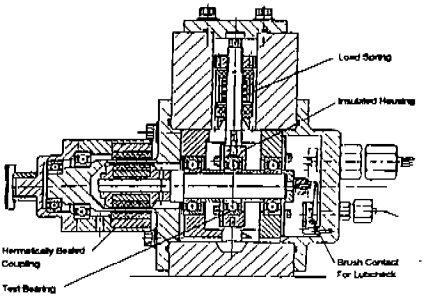

Fig. 13 Bearing test head arrangement

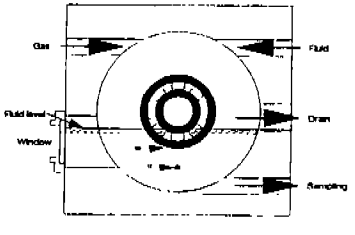

Fig. 14 Test head lubrication arrangement.

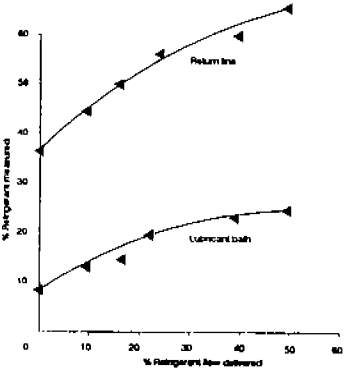

Fig. 15 Refrigerant concentration in test head.

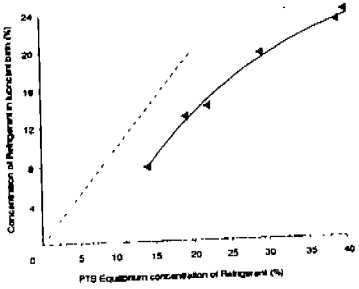

Fig. 16 Refrigerant concentration relative to PTS equilibrium.

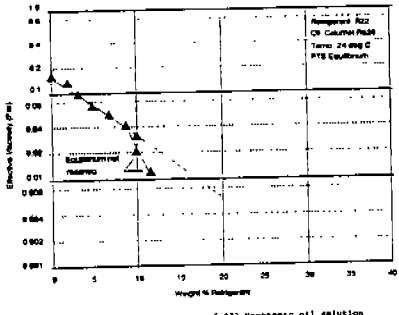

Fig. 17 Effective viscosity of R22-Napthenic oil solution determined for PTS equilibrium.

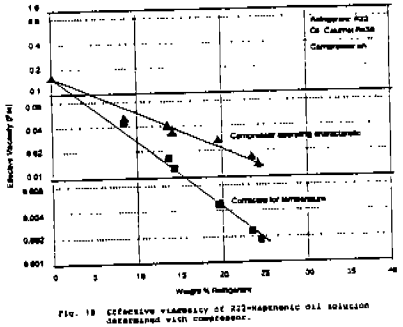

Fig. 18 Effective viscosity of R22-Napthenic oil solution determined with compressor.

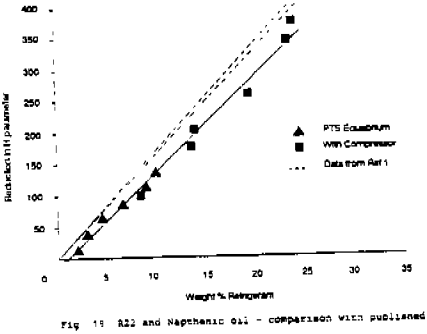

Fig 19 R22 and Napthenic Oil - comparison with published data.

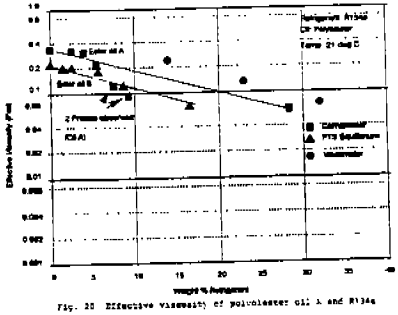

Fig. 20 Effective viscosity of Polyolester Oil A and R134a