1996

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B. Jacobson
SKF Engineering & Research Centre B.V.

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Bo Jacobson

SKF Engineering & Research Centre B.V.
Postbus 2350, 3430 DT Nieuwegein, The Netherlands
e-mail SESKFHE5@IBMMAIL.COM

ABSTRACT

When the refrigerant R134a is mixed with an ester oil and used in compressor bearing lubrication, no EP action is experienced. The refrigerant contains no chlorine and the lubricant additives can not be chemically active like normal EP additives, not to deposit in and destroy the heat exchangers. The lubricant film thickness thus needs to be considerably larger for an R134a compressor bearing than for e.g. an R22 compressor bearing if the same service life should be expected for the two bearings. Typically 50% higher viscosity was needed for the R134a mixture to get a certain wear rate compared to an R22 mixture giving the same wear rate. To avoid the problem with high wear rate at high refrigerant concentrations and to make it possible to run the compressor bearings with less oil an experimental investigation was performed. The experiments investigated the possibility to locally increase the oil concentration at the elastohydrodynamic contacts in the bearings to get better lubrication.

INTRODUCTION

The increasing demands for more environmentally safe refrigeration processes and machines have led to the adoption of a number of new refrigerants and refrigerant mixtures to fulfil the thermodynamic requirements for the refrigeration cycles. When chloro fluoro carbon refrigerants were used in compressors, the chlorine in the molecules functioned as an EP additive. The lubricated surfaces could often not be separated by a continuous lubricant film due to the very low viscosity of the oil-refrigerant mixture, but the chlorine made it possible to get a controlled running in of the surfaces leading to lower surface roughness and better separation of the surfaces. The bearing surfaces reached a stable smooth state after some running time.

Many of the new refrigerants do not easily mix or dissolve in mineral oils and can thus not be used together with the lubricants earlier used for e.g. R11, R12 and R22 refrigeration systems. The solubility and the rate at which refrigerants can be dissolved in the lubricant and also how fast they can be released from the lubricant determine the state of the lubrication at each position in a compressor. As the oil-refrigerant mixture circulates through the different parts of the compressor and the refrigeration circuit, there are different time delays between the local pressure-temperature-state and the equilibrium pressure-temperature-state depending on the lubricant type and the refrigerant type. The old systems with CFC:s and mineral oils have very rapid response with fast dissolution in and a fast release from the oil of refrigerant when the pressure is changed.

With the synthetic esters developed for lubrication of R134a compressors the response time can be chosen more or less freely depending on the requirements. The problem is that the requirements for the compressor and the requirements for the heat exchangers are pointing in quite opposite directions. The compressor lubrication works best when the lubricant contains only a small percentage refrigerant i.e. the solubility of refrigerant in the oil should be low. But this low solubility will result in difficulties to retrieve any of the oil deposited in the heat exchangers. If, on the other hand, the solubility of refrigerant in the lubricant is high, oil in the heat exchangers will be brought back to the compressor by the refrigerant flow but this also leads to problems with separation of the refrigerant from the lubricant at the bearing positions. It will be difficult to reach a high enough oil concentration to get good lubrication of the compressor bearings. There are thus two possible remedies for this problem if the refrigeration heat exchangers are not going to be adversely effected: Locally at the bearings increase the oil concentration so a high enough viscosity is reached or keep the low oil concentration but make the bearings less vulnerable to too low viscosity.

LOCAL OIL CONCENTRATION

The total volume of lubricant at each instance of time being kept in the load carrying oil film in a bearing is extremely small. For a heavily loaded bearing of the type 6309 (deep groove ball bearing with inner diameter 45 mm and outer diameter 100 mm) the total Hertzian contact area between the balls and the rings is about 25 mm^2 and if the mean oil film thickness is 0.4 μm the total load carrying oil volume is 0.01 mm^3. This means that even if the oil concentration in the refrigerant is as low as 1 %, only 1 mm^3 of that mixture needs to be evaporated to leave enough oil on the bearing surfaces for good lubrication.

This can be done locally at the bearing by local heating or by the bearing power loss as described by Wardle et al. [1,2]. At the present time, most refrigeration systems utilize two partially miscible fluids; a refrigerant for heat transfer, and a lubricant for lubricating machine elements in the compressor. Compressors typically have a separate sump, pumping means, and distribution system for the lubricating oil, all isolated, but not sealed from the refrigerant. Pressure, temperature and mechanical separation means are employed to maintain a sufficiently oil rich mixture in the sump for reliable lubrication, typically no more than 20% refrigerant by weight. Refrigeration systems are also designed to limit the amount of oil discharged into the heat transfer devices, such as the evaporator, to avoid fouling and associated loss of system efficiency. Solubility characteristics between the fluids are tailored to minimize the amount of dissolved refrigerant in the oil sump, yet provide sufficient solubility in the evaporator, condenser, and interconnecting piping to assure oil leaving the compressor returns via entrainment in the circulating refrigerant.

The need to maintain an oil rich fluid in the sump, and limit the build-up of oil in the evaporator, over the wide range of operating conditions common in refrigeration systems, usually necessitates complex and costly control systems and fluid separation features. Loss of control of the two fluids during extreme system operating conditions is a common cause of compressor failures, particularly compressor bearing failures, due to excessive refrigerant build-up in the sump. For these reasons, efforts have been directed to elimination of the need for oil separation, and to use the refrigerant rich lubricating fluids naturally residing in the evaporator or condenser for lubrication purposes.

The invention, described in [1,2] provides a means of reliably lubricating rolling bearings in refrigerant compressors with refrigerant/oil mixtures which would normally have inadequate viscosity for such purposes. It has been discovered that, generally speaking, mixtures containing less than about 75% oil by weight can not sustain an oil film in rolling element bearings and therefore are unsuitable for lubrication purposes. In conflict with this requirement is the need to restrict oil concentration to 5% or less in the heat exchangers of a refrigeration system. The presence of oil in a heat exchanger negatively affects heat transfer and overall system efficiency. The invention, described in [1,2], provides a means for using a refrigerant rich mixture communicated from either an evaporator or condenser, to lubricate rolling element bearings by a process which concentrates the oil to a level suitable for the lubrication of such bearings. The concentration of oil is achieved by movement of the mixture into the two phase region of its pressure-enthalpy region to release an oil rich fluid for lubrication purposes on the bearing.

Bearing frictional heat plays a significant role in the enrichment process if flow rates are limited to the low levels typical of spot or mist lubrication methods. At such low flow rates, bearing frictional heat, or bearing frictional heat combined with reduction of pressure, is sufficient to evaporate the refrigerant.

The principal advantage of the invention, described in [1,2], is that the use of lubricating fluids naturally residing in either a low pressure evaporator or a high pressure condenser avoids the need for expensive and complicated mechanical oil separators, and avoids the need for a separate oil sump and associated sealing elements. This is particularly true in screw compressors using liquid refrigerant injection instead of oil injection to cool and seal the compression process since an oil separator would only be needed for bearing lubrication. Another advantage is that significantly reduced bearing friction is achieved due to low oil flow which may be as low as that used in spot or mist lubrication.

The invention, described in [1,2], requires that temperature, pressure and flow rate for the mixture be controlled to accomplish the vaporization of refrigerant from the mixture and the deposition of the remaining oil on the bearing. Proper control of these variables will insure good lubrication.

By electrically measuring the oil film build-up in a test bearing, the bearing speed needed for surface separation at different bearing outer ring temperatures and different refrigerant gas pressures was registered, see Figure 1. For each given outer ring temperature a higher pressure gives a higher refrigerant concentration, see Figure 2, but for a constant pressure a higher temperature gives a lower refrigerant concentration in the lubricant. This leads to the surprising result that if the temperature is increased at constant pressure, so much refrigerant boils off from the mixture that the viscosity of the mixture actually increases when the temperature increases. This is clearly seen in Figure 1, where for all gas pressures the speed needed for good lubrication decreases when the temperature increases. It is thus advantageous to have the bearing at a high temperature to boil off as much refrigerant as possible from the lubricant mixture. To boil off as high a percentage of the refrigerant flow as possible with a certain available power, the refrigerant flow should be minimized. Tests were run with 99 percent R134a and 1 percent ester oil at different flow rates through a ball bearing and only at flow rates below 0.01 gallons per minute (0.04 l/minute) could the power loss in the bearing increase the bearing temperature above -4°. At the flow rate 0.005 gallons per minute (0.02 l/minute) the power loss gave a temperature increase of 8° and very good lubrication.

By decreasing the refrigerant-lubricant flow rate through the bearing, until the bearing temperature gets so high that the lubricant at that temperature and pressure can contain only small amounts of refrigerant, the lubrication will be optimized giving a thick oil film and a low power loss.

LOW WEAR RATE BEARINGS

If the concentration of refrigerant in the lubricant can not be decreased below 25 percent, the oil film thickness will not be large enough to give acceptably low wear rates for the bearing contact surfaces in R134a compressors running at speeds below 6000 rpm. As already stated in the 1994 International Compressor Engineering Conference at Purdue [3] the wear rate was only becoming acceptably low at 6000 rpm and 20 percent refrigerant both for R134a and R22 in tests with angular contact ball bearings. The viscosity of the R134a mixture was then about 50 percent higher than the viscosity of the R22 mixture showing that the EP action of R22 makes it possible to run at a thinner oil film with the same amount of wear.

When the test results were analysed more in detail, the lubricant film thickness needed for acceptably low wear rate was much higher than expected for normally lubricated bearings in an air environment. For the R22-mineral oil mixture at the bearing position the viscosity needed was about 3.5 times higher than what should be needed for a bearing surrounded by air. The R22 atmosphere was not as efficient as air in guiding the running-in and smoothening the bearing surfaces.

The R134a-ester oil mixture was even worse. More than 5 times higher viscosity than normally predicted for bearings in air atmosphere was needed to get the wear rate down to acceptably low levels. The bearings did not seem to run in at all. This leads to two conclusions:
1) the bearing type has to have extremely smooth surfaces already from production. It will not run-in and become smooth.
2) the whole lubricant-refrigerant system has to be kept clean, so that contaminant particles can not be overrolled and destroying the smooth bearing surfaces. If the bearing surfaces are roughened by rolling on particles they will later not be possible to separate with an oil film and the bearing wear will be rapid.

CONCLUSION

To get good bearing lubrication also at high concentration of refrigerant in the lubricant is necessary to dramatically decrease the flow of oil-refrigerant mixture to the bearings. Only then it is possible for the bearing power loss to boil off a high enough percentage of the refrigerant to get high enough viscosity for good lubrication. This makes it possible to operate a compressor without an oil separator for bearing lubrication.

As the lubricant film thickness to surface roughness ratio needed for long bearing life in R134a compressors is higher than in an air environment, bearings with very smooth surfaces should be chosen and contamination particles in the lubricant should not be allowed to destroy the surfaces.

REFERENCES

[1] US patent 08/184861

[2] European patent application 95200132.9

[3] Jacobson, Bo, "Lubrication of Screw Compressor Bearings in the Presence of Refrigerants", Proc. Vol. 1, 1994 International Compressor Engineering Conference at Purdue, pp. 115-120

FIGURES

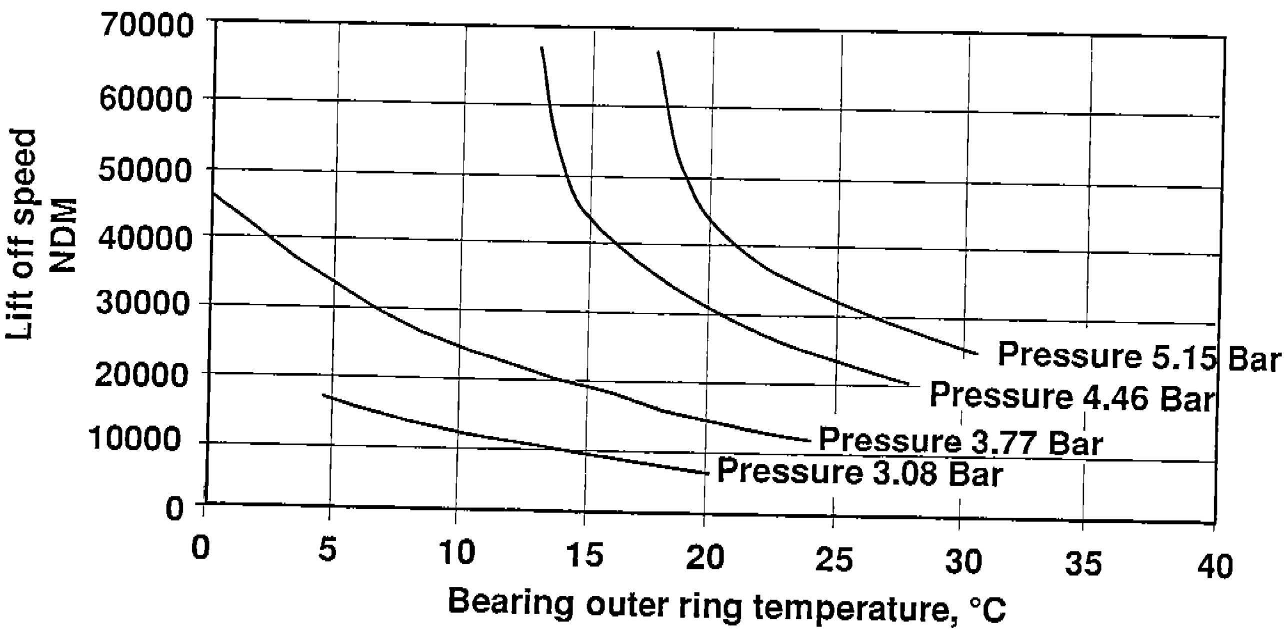

Fig. 1 Bearing lift-off speed (rpm x bearing mean diameter in mm) as a function of refrigerant gas pressure and bearing outer ring temperature.

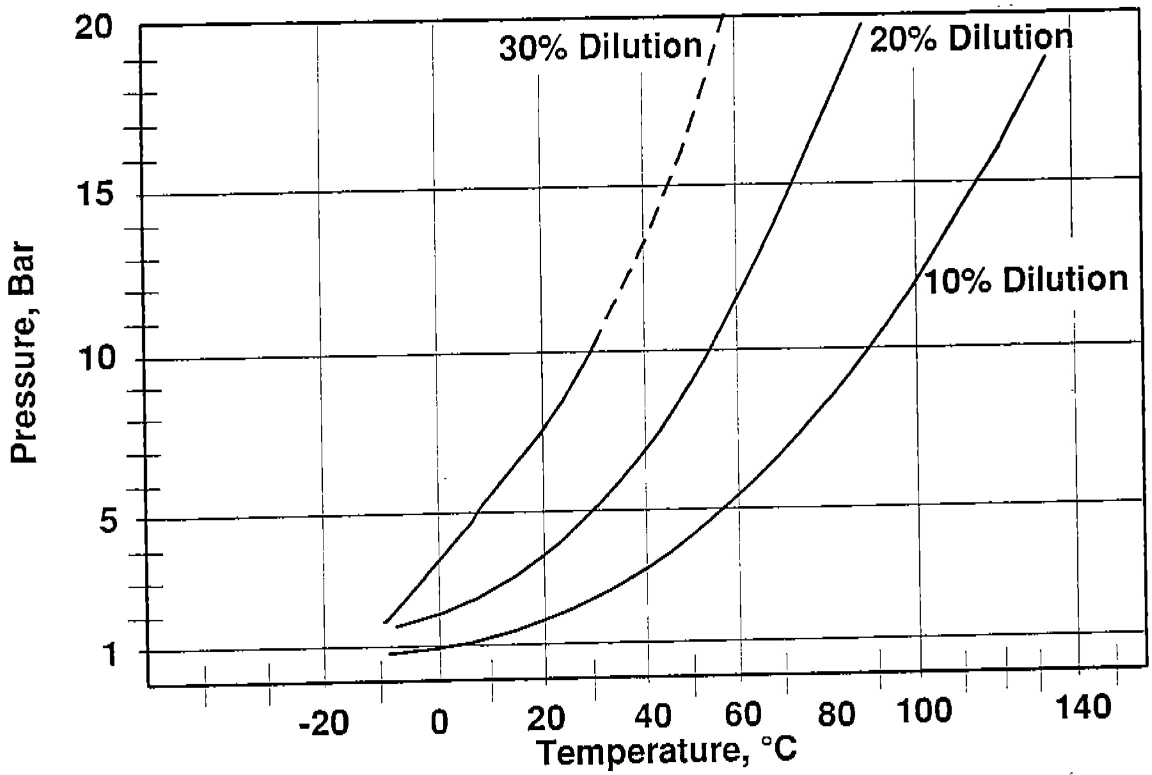

Fig. 2 The refrigerant gas pressure-percentage dilution of refrigerant in the lubricant-temperature relationship.