Externally Pressurised and Hybrid Bearings Lubricated with R134a for Oil-Free Compressors

A. K. Molyneaux
Federal Polytechnic of Lausanne

R. Zanelli
Federal Polytechnic of Lausanne

EXTERNALLY PRESSURISED AND HYBRID BEARINGS LUBRICATED WITH R134A FOR OIL-FREE COMPRESSORS

Dr Alex K. Molyneaux, Robert Zanelli.

Industrial Energy Laboratory
Federal Polytechnic of Lausanne
1015 Lausanne, Switzerland.

ABSTRACT

This paper discusses the choice and implementation of bearings for oil-free compressors where the system refrigerant is used as the lubricant in externally pressurised and hybrid bearings. The problems of oil contamination in heat pump applications is described followed by methods to produce genuine oil-free hermetic systems. Alternative bearing choices are described for high and low speed applications. Theoretical and experimental results are presented showing the effects of turbulence in the bearing clearance and pockets of externally pressurised thrust and journal bearings for a co-rotating scroll compressor operating with r134a. The paper concludes that design methods are now available to enable the incorporation refrigerant lubricated bearings in hermetic oil-free compressors.

INTRODUCTION

Since the recent realisation that CFC and HCFC significantly contribute to the ozone layer depletion, several forms of action have been undertaken, i.e. to introduce new refrigerants with similar properties, to recover refrigerants from units during maintenance or at the end of their life, to minimise the charge of new units and to use better sealing for larger open type units. This project aims at studying the behaviour of bearings lubricated be new refrigerants instead of oil which would facilitate the design of oil-free hermetic compressors (Favrat et al, 1993). Such a product would avoid refrigerant contamination with oil in refrigerating, air-conditioning and heat pump units. Significant advantages from oil-free operation would be, among others, better heat transfer at high vapour quality in evaporators, improved use of enhanced heat transfer surfaces, easier multistage cycle design, cleaner refrigerant recovery and re-use, and reduced fluid and material compatibility problems.

To make a refrigerant machine, either compressor-motor or expander-generator, truly oil-free requires that not only the main bearings but also any gears or other contacting surfaces be eliminated or redesigned to function with refrigerant. For example, a typical scroll compressor requires bearings to support the electric motor, which operate under conditions for which it is reasonable to envisage the use of refrigerant, however the orbiting mechanisms usually rely on boundary lubrication of the oil. It is difficult to envisage a metal to metal contact lasting sufficient time with only refrigerant as a lubricant.

Hermetic oil free machines are not a new idea, many compressors and pumps were fabricated and operated successfully for nuclear installations, using the working fluid as lubricant, for example Helium gas (Clarke, 1967), invariably though the compressors and pumps were dynamic, i.e. centrifugal, where the elevated speeds facilitated the use of gas bearings. Product lubricated bearings are also found in current pump applications some linked to the use of magnetic couplings and new ceramic materials (Bergmann, 1995) although many of these could generally be classified as low cost, low technology and short lifetime. Successful applications of the use of refrigerants as lubricants in industrial applications are difficult to find. No doubt because one can find bearings off-the-shelf for oil lubrication (rolling element, hydrodynamic sleeves) where there is little need for specialised bearing design knowledge, coupled with the inherent capacity for oil lubricated bearings to survive massive overloads compared to design loads.

BEARING DESIGNS

There are three types of bearing of interest for long lifetime cost sensitive industrial applications, usually referred to as hydrostatic, hydrodynamic and rolling element. Magnetic bearings will clearly be of interest for higher cost specialised machinery but cannot be expected to compete for the general high volume market.

Hydrostatic bearings, or using more recent terminology, Externally Pressurised (EP) bearings, are, as the name suggests, supplied with a source of pressure that supports the load. They are used in many precision machine tools with great success but wider application is limited often by the extra power consumption associated with the pumping power that has to be provided. Analytic and numerical methods are available (Bassani, 1992) to calculate their performance with sufficient precision for liquid lubricated bearings including second order effects such as turbulence and inertial losses. Gas lubricated EP bearings have pneumatic instability difficulties that require much more complex analyses, which are available, but less precise and proven. Normally EP liquid lubricated bearings have a series of pads (pockets) distributed around the available surface which are supplied by pressurised lubricant through a compensation device, these pads will cover typically 60% of the surface. EP bearings are characterised by their ability to support the same loads at any rotational speed up to the design speed but with a load limit which is determined simply by the available supply pressure and the surface area over which it acts: the compensation (flow limiter, i.e. capillary, orifice, etc.) that is used in the supply to the bearing pads merely modifies the stiffness characteristic of the bearing. More exotic compensation devices that produce infinite stiffness over the working range are available but can cause stability problems.

Hydrodynamic bearings, often called Self-Acting (SA) these days, generate the pressures necessary to support the bearings loads internally, due to the pumping effect of the trapped viscous fluid between the rotating and non-rotating components. They are found in all engines and similar machinery, and are characterised by their ability to support load only while rotating, but have an upper load limit that is dependant on the precision of the surfaces. Normally the surfaces in contact are as smooth as possible with the exception of lubricant supply channels, but for journal bearings a variety of stability enhancing features are found (herringbone grooves, lobes, pads, pockets etc.), especially for high speed lightly loaded bearings.

Hybrid bearings are EP bearings that rely on the SA effects to support the applied loads as bearing speed increases. Hence loads can be greater than those for pure EP bearings since bearing pressures are self generated.

Rolling Element Bearings (whether they are ball, roller or tapered elements) function due to the formation of an elasto-hydrodynamic (EHD) lubricant film to separate the moving surfaces, primarily due to the excellent pressure-viscosity characteristics of normal lubricating oils. Recent advances in material purity has resulted in a range of REBs that will function when oil lubricated without fatigue wear for limited loads, assuming good lubricant conditions, i.e. < 5 micron filtration. There is very little research about the use of REBs lubricated with refrigerants and until the pressure-viscosity characteristics of them are known it is difficult to estimate any EHD effect. New ceramic materials for the components of REBs offer new possibilities that need to be studied.

MATERIAL ASPECTS

The objective in developing refrigerant lubricated bearings implies no contact between the rotating and non-rotating components, but this will be difficult to achieve under all operating conditions. During the initial start of the machine there is unlikely to be pressure available for EP bearings to function as EP bearings, and SA bearings require the rotational speed to generate lift. Hence materials and/or surface treatments will be essential that permit short duration contact without significant wear nor seizure.

THEORY

The analysis of normal laminar flow incompressible EP or SA bearings is more or less fully understood and there are no technical difficulties in their application, assuming second order effects due to cavitation and temperature are taken into account, even for dynamic performance, usually based on finite element analyses of Reynolds equation: to the extent that design guides and standard bearing components can now be found. These provide satisfactory designs while oil lubrication is the objective since the overload capabilities of oil bearings is large due to their design and even under excessive loads they survive due to the pressure-viscosity characteristics and boundary lubrication: these cannot be relied on for refrigerant lubrication and the design needs to be more precise: added to this is the problem of turbulence due to the lower relative viscosityís of these fluids.

For example, typical lubricating oil has a viscosity of the order of 10 e-3 Ns/m^2 while liquid r134a has a value of the order of 0.2e-3 Ns/m^2. An oil bearing of typical dimensions (40 mm diameter at 3000 rpm) would have a Reynolds number of the order of 30 but the same bearing with r134a would have a Reynolds number of the order of 2000 with approximately 50 times less load (and power loss) To support the same load an r134a lubricated bearing would require a smaller clearance and/or larger diameter and will have highly turbulent flow. This is not a problem in itself but complicates the analyses and results in a bearing where the load and power become more dependant on speed, for EP and SA bearings. Two principle methods are commonly found to consider theoretically the effects of turbulence: the turbulent flow correlationís developed by Elrod , Ng, and Constantinescu (Taniguchi, 1987) and the friction factor model (San Andres 1992). The former of these can be viewed as equivalent viscosity corrections similar to the slip flow corrections found in gas bearing slip flow.

Figure 1 shows the effective increase in viscosity for a range of Reynolds numbers likely to occur in the bearing geometryís and conditions of interest. This viscosity increase factor is for Couette flow in the direction of rotation, a similar correlation exists for the transverse direction and for the Poiseuille flow. An iterative scheme is proposed assuming the largest of the Couette and Poiseuille values to arrive at the final effective viscosity. Kosasih (1993) confirms the values of turbulent factors (Gx, Gz) suggested by Tanaguchi (1987) showing that errors arise principally for bearings operating in the transition regime from laminar to turbulent flow. San Andres (1992) discusses the effect of pressure on the performance of bearings lubricated with cryogenic fluids, principally liquid oxygen and hydrogen, but refrigerant r134a does not have such a large change of density with pressure (at the conditions envisaged) as these latter fluids and therefore the effect of pressure on the operation of r134a lubricated bearings should be small.

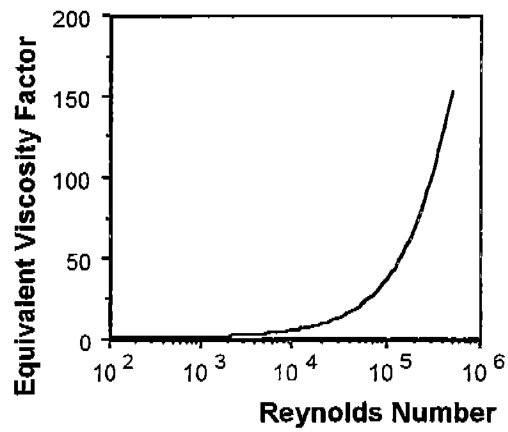
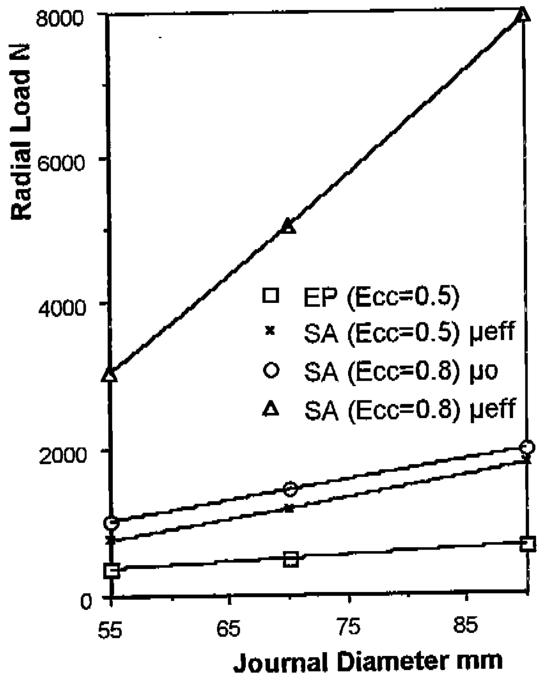


Figure 1: Increase of Effective Viscosity Due to Turbulence (Taniguchi et al, 1987)

Wasson (1993) mentions the use of ceramic materials for hydrostatic bearings but confuses the situation by concluding that water lubrication reduces power consumption by approximately 50%, which appears to be true only for the conditions and geometry chosen. Kumar and Rao (1992) discuss the stability of turbulent bearings, showing that they have lower stability margins than laminar bearings; but the use of Somerfield number confuses the issue since comparison is not based on the same point of reference (Somerfield number is non-dimensional load, hence operating clearance and eccentricity are lower for the turbulent flow bearing compared to the laminar flow bearing, lower eccentricity is a major factor in reducing stability). Russo and Russo show that stability is improved for turbulent flow, while San Andres (1991) shows that inertial effects (pressure drop at the change of film height at pocket edges) coupled with the induced cavitation, reduces stability.

TEST CASE SCROLL

One of the major objectives of this work was to provide r134a bearings for a co-rotating scroll compressor, envisaged to be for a heat pump with up to 18 kW electric input at 9000 rpm. To achieve this hybrid bearings are considered suitable, at low speeds the external pressurisation effect will support the loads and at higher speeds the self acting effect will increase operating clearance allowing reduced power consumption. An additional advantage in heat pump applications is the possibility to use liquid from the cycle at suitable pressures and temperatures without having to add a separate pump with its associated losses. The design of the bearings would be achieved using the data in Bassani together with turbulent correlationís as suggested by Taniguchi, with additional bearing and rotordynamic analyses using commercial computer programs (Molyneaux , 1994). This would provide a rapid means of designing refrigerant lubricated bearings without recourse to detailed finite element analyses of individual bearing configurations. Figures 2 and 3 shows the theoretical performance of journal EP and SA bearings of various diameters as well as the power consumption. The turbulence in the bearing clearance is such that these powers are 2 to 5 times those that would be expected for laminar flow, with Reynolds numbers up to 20000, while in the pockets the increase in effective viscosity is of the order of 100 times and Reynolds numbers up to 450000. The loads for the EP bearings are not increased due to the turbulence since it is assumed that the pocket pressure ratio is constant. Optimisation of pocket geometry to minimise power consumption gives a small improvement. Momentum power is not included in these results for power consumption.



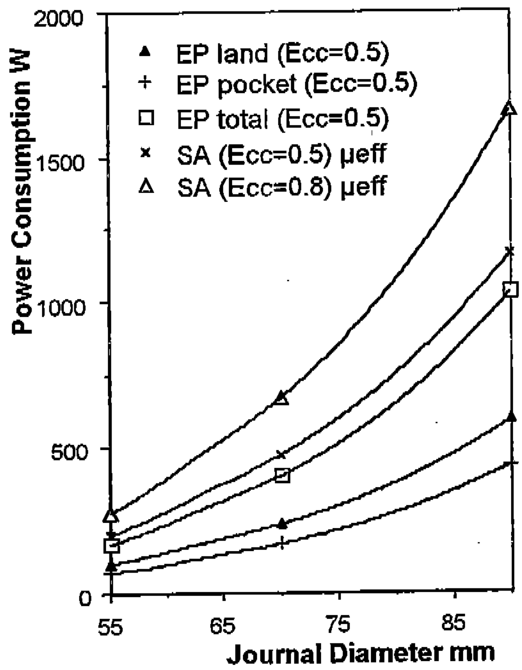


Figure 2: Theoretical Load Capacity **Figure 3: Theoretical Friction Power**

Journal bearing theoretical results in figures 2 and 3 assume bearing length = 30 mm, r134a viscosity = 270 μNs/m^2, EP supply pressure = 9 bar, ambient pressure = 1 bar, pocket pressure ratio = 0.4, EP land width ratio = 0.3, 10000 rpm, radial clearance = 30 μm, pocket depth = 2 mm. All these results are equally applicable to laminar flow restrictors or orifice flow restrictors, the difference between these two being the resulting stiffness which in this application is not of primary concern.

EXPERIMENTS ON EP BEARINGS

Confirmation of the above theoretical results has been achieved by performing experiments using a simple test apparatus, with a rotor supported on two EP journals and one EP thrust bearing driven by a variable speed electric motor through a torque meter. Displacement measurements were made at three locations to measure rotor motion, and there is an axial loading mechanism to enable load capacity to be measured. Flow was measured using tube flowmeters especially calibrated for r134a. Temperatures and pressures were measured and all recorded automatically by computer. A pump was used to vary bearing supply bearing pressure up to 6 bar above the system (condenser) pressure of approximately 4 bar. A small volume of liquid was evaporated and the resulting gas used to supply the axial loading mechanism to minimise parasitic losses. A magnetic coupling was used to transmit the torque from the electric motor and torque meter in ambient air, to the rotor inside the pressurised chamber. All the three EP bearings had 4 pockets, each with supply feed and pressure measurement connections. Nominal journal bearing radial clearance was 30 micro-m, diameter 80 mm and length of 10 mm, the pockets were 4 mm wide and 2 mm deep. The thrust bearing had an outer diameter of 100 mm, inner diameter of 80 mm. Bearing materials were stainless steel and bronze as this gives negligible change of bearing clearance with temperature due to their close thermal expansion coefficients. One difficulty with turbulent bearings is the choice of flow restrictor, to mimic the usual laminar flow practise one would like a turbulent flow restrictor that allows a constant pocket pressure ratio with speed. Orifice flow restrictors can be optimised for one operating condition due to the considerable change of effective viscosity, at zero speed a diameter of 1.0 mm is optimal (pocket pressure 50% of supply at no load) but at 10000 rpm a diameter of 0.5

mm is optimal. A value of 0.8 mm was used as a compromise for initial tests. Recent work by Kim and O'Neal (1995) provides a means to improve on this compromise.

RESULTS

Figure 4 shows the axial load against axial clearance when not rotating, as can be seen the agreement is poor at the higher loads probably due to the geometry of the bearings. The pocket pressure ratio varied between 0.02 and 0.2 during these experiments, which is low, as would be expected with the smaller than optimal orifices.

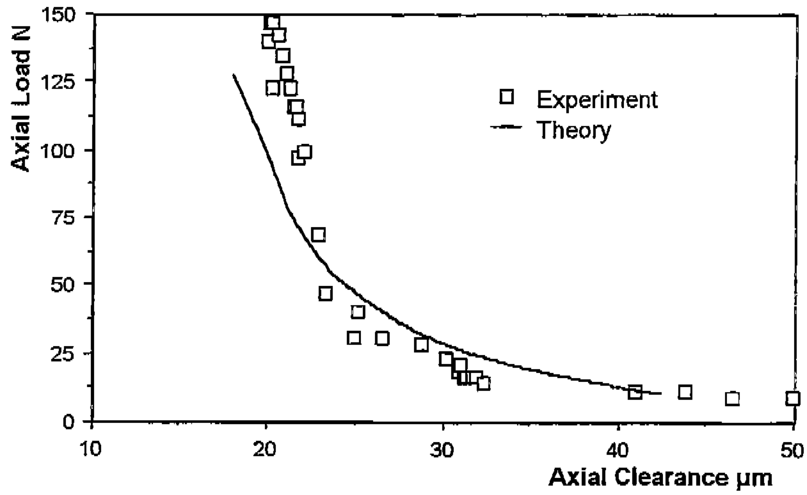


Figure 4: Externally Pressurised Thrust Bearing Axial Load

2.5 bar supply pressure, r134a lubricant, Zero speed.

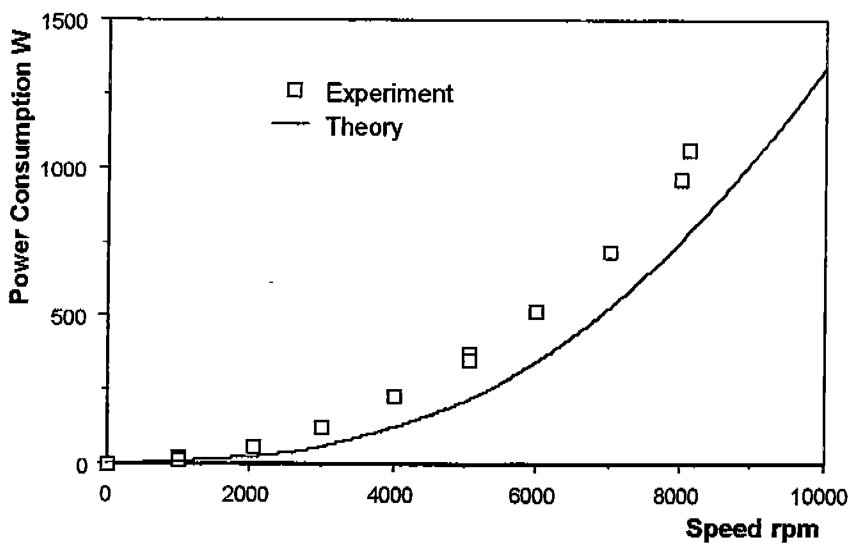


Figure 5: Externally Pressurised Bearing Power Consumption

2.5 bar supply pressure, r134a lubricant, 2 journals, 1 thrust

One of the major interest in performing these experiments was to measure the power consumption, this is shown in figure 5. Higher speeds were not possible at this stage due to axial vibration arising from the inability to load sufficiently the axial bearing. No windage has been taken into account on the rotor but losses in the thrust bearing and journal bearing exit zone were. Parasitic losses from the magnetic coupling and its support ball bearings were measured with values up to 140 W at 10000 rpm; these were subtracted from the values shown. Calculations following the SKF manual suggest that the two sealed ball bearings consumed 56 W at 10000 rpm i.e. the remaining 94 W being the magnetic coupling losses. Inertial effects were calculated to be negligible following the method of Bassani (1992).

Further experimentation is planned to test a hybrid bearing under similar conditions and to measure the rotor dynamic response.

CONCLUSIONS

The theoretical and experimental results reported in this paper show how refrigerants like r134a can be used in the liquid phase to lubricate externally pressurised, self acting or hybrid bearings as long as care is taken to take into account turbulent losses in the land and pocket regions.

The equivalent viscosity method used to modify laminar flow calculations have shown that in the cases tested the results are sufficiently close to design for satisfactory steady state operation. Further work is required to examine the dynamic behaviour and especially to optimise the restrictor type and size.

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