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Progress and challenges in rolling bearing technology for compressors in industrial heat pumps

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Abstract

The introduction of new process media like refrigerants with low GWP (Global Warming Potential) and the specific operating conditions of heat pumps can create challenges for traditional bearing technology: dilution of the lubricating oil by process media is increasing, lubrication films are getting thinner, and risk of corrosion and chemical compatibility need to be investigated upfront. Furthermore, the use of variable frequency drives (VFD) extends the speed range to both lower and higher values for which the bearings must operate reliably. In certain cases this even requires a change of bearing technology like from hydrodynamic to rolling element bearings. But also rolling bearings can suffer from above mentioned conditions. Therefore this article summarises the extensive research and development to provide state-of-the-art rolling bearing solutions and related engineering advice. This involves tribological testing and lubrication characterization of refrigerants and oil-refrigerant mixtures. The article also touches on the use of hybrid ceramic bearings, stainless materials, coatings, and advanced cage materials. The effect of oil dilution and poor lubrication and its impact on surface distress and rolling bearing life, as well as the effect of contaminating particles have been captured in models to allow engineers to give improved predictions of bearing performance and durability. Finally, different rolling bearing solutions to deal with the challenges related to lubrication and bearing life in heat pump compressors are described, all the way from oil-refrigerant mixture lubrication with moderate high dilution rates, up to a major innovation related to oil-free lubrication, so-called Pure Refrigerant Lubrication (PRL).

Keywords: bearings; ceramic, tribology; lubrication; refrigerants, heat pump; compressor

1. Introduction

In the UN COP26 conference on climate change in 2021 one of the priorities was securing agreements for reducing emissions, with the goal of limiting the increase of global temperatures to 1.5 °C. Investment in renewable energy sources is encouraged while phasing out fossil fuels is identified as crucial to delivering this target [1]. Heat pump technology is seen as a key technology to replace fossil fuel usage and reduce carbon emissions. One major advantage of heat pump technology is that heat can be collected from a variety of renewable or other sources, such as air and water as well as geothermal and waste heat, producing no direct local emissions (as long as heat pump drive energy is also renewable).

Industrial heat pumps and heat pumps for district heating often use waste heat from processes in industries like food & beverage, pulp & paper, cement and building materials industry, chemical and petrochemical industries as well as refineries, and even large computer or data centres that need significant cooling for their operations. Heat pumps can help to cut operating costs and enable the electrification of the heating process.

Different from chillers, compressors in heat pumps must enable higher temperature (and pressure) lifts, often with discharge temperatures larger than 100°C to be applicable e. g. for drying operations and steam

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production. The higher temperatures and pressures, together with the use of (new) refrigerants, lead to bearing lubrication regimes with a mixture of lubricant oil and refrigerant, in some cases with very high rations of low-viscous refrigerant in the oil. At higher temperatures the lubricating films become thinner than in conventional chillers, affecting the life of the bearings. The use of new low-GWP refrigerants, which are typically less chemically stable or more chemically active, or the use of ammonia as a refrigerant might in certain cases (like in the presence of humidity or water) harm the bearing materials by chemical attack, corrosion, ageing or close contact with hydrogen derivatives. The use of variable frequency drives (VFD) broadens the speed range in which the bearing solution must operate reliably. These challenges make rolling bearings one of the most critical machine elements in the evolution of heat pump technology, and reliable solutions need to be developed.

There is good amount of research work dedicated to rolling bearings lubricated with oil-refrigerant mixtures (ORM) and pure refrigerant lubrication (PRL). However, for the purpose of this publication, focus is put on the research that aims to describe and understand the effects of these types of lubrication on bearing performance and bearing life. For refrigeration compressors, the reader is kindly referred to the review paper published earlier by some of the current authors [2].

Early work in the 1990's related to the ORM lubrication of bearings and life effects is described in [3-4]. The effect of refrigerant dilution in lubricated contacts and film thickness is studied experimentally and numerically in [5-9]. Detailed numerical work of the elasto-hydrodynamically lubricated (EHL) contacts with ORM, including thermodynamic effects in dilution is presented in [10]. Recent work related to the modelling of the lubrication quality in rolling bearings with ORM is described in [11-12]. Work related to describing damage and life effects in rolling bearings running under ORM conditions is found in [13]. In relation to study the film thickness and rheological properties of refrigerants in EHL conditions, several experimental and numerical works can be cited [14-18]. Rolling bearings lubricated PRL conditions are studied in [19-22].

The current publication describes the tribological challenges faced by rolling bearings in typical industrial heat pumps, including such that work beyond discharge temperatures of 100°C. It presents information on modelling and experimental aspects which provide explanations for successful technological solutions. It will be impossible to have a complete review of the topic, since there is a very large and diverse literature database, and more research is ongoing while this publication is written. However, the objective of this article is two-fold: (i) to briefly summarize application aspects of rolling bearings above mentioned conditions, including different bearing designs and technologies, and (ii) to more extensively present scientific aspects on lubricant properties of (pure) refrigerants and oil-refrigerant mixtures, including modelling of surface life in rolling bearings. This double objective is difficult to find in existing publications.

Description of Compressor Designs used in Heat Pumps

In heat pumps utilising the vapour compression cycle, the compressor is the main mechanical component. There are two important compressor types used in heat pumps of industrial size with typical drive motor power above 50 kW [23] where rolling bearings play a key role for compressor reliability and efficiency: screw and centrifugal compressors (Figure 1). For detailed description of these compressor designs and typical bearing arrangements, the reader is referred to [2] and [24].

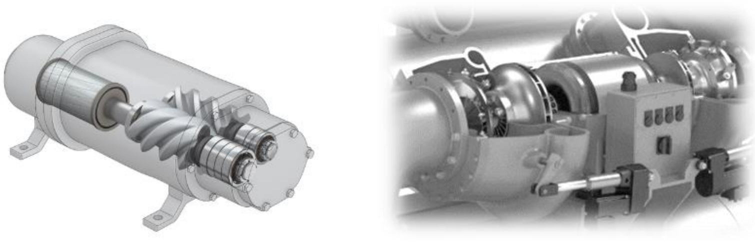

Fig. 1. Left: screw compressor. Right: centrifugal refrigerant compressor.

The bearing arrangements are usually the same or very similar for refrigerant compressors used in air-conditioning (i.e. for chillers) or in heating applications. Nevertheless, the operating conditions are heavily influenced by the needed temperatures and used refrigerants. Every refrigerant has its own characteristics which is commonly visualised in the pressure-enthalpy diagram (Figure 2). Within the "vapour dome" (the two-phase region) pressure and temperature are linked with each other. Therefore, if the source and the sink temperature (evaporator and condenser temperatures) of a specific application are known, the characteristics of the used refrigerant will link it to the pressure levels – which set the suction and discharge pressure of the compressor. A heat pump operating at higher temperature will create higher gas pressures leading to higher bearing forces. At the same time the change in pressure and temperature affects the dilution of the lubricant and its viscosity.

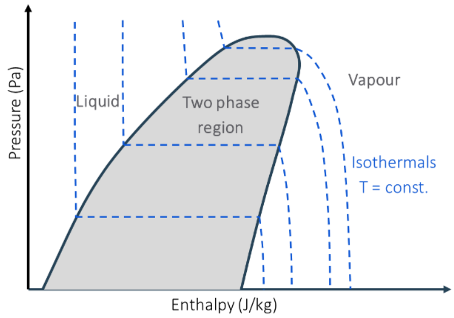

Fig. 2. Characteristics of a refrigerant in the pressure-enthalpy diagram.

1.1. Description of Oils and Refrigerants

In the past, commonly used refrigerants for compressors were the low-pressure chlorofluorocarbon CFC-11, and medium-pressure refrigerants like CFC-12 and hydrochlorofluorocarbon HCFC-22. In 1989, a global ban of ozone depleting refrigerants – indicated by their Ozone Depletion Potential (ODP) - was agreed upon in the Montreal Protocol. New HCFC and hydrofluorocarbon (HFC) refrigerants were developed having low or no ODP. Low pressure refrigerant HCFC-123 replaced CFC-11 and medium pressure refrigerant HFC-134a became a substitute for CFC-12 and HCFC-22.

When global warming started to become a bigger concern in the 1990s, another unintended characteristic of refrigerants got more attention, namely their high Global Warming Potential (GWP). For example, HFC-134a has a GWP of 1300 times that of CO2. In 1997 the so called Kyoto Protocol agreement on reduction of global greenhouse gases was proposed - but not ratified by major countries and never fully implemented. In 2016 it was decided to use the more successful format of the Montreal Protocol to control phase down of HFCs, referred to as the Kigali Amendment of the Montreal Protocol.

The newer refrigerants were not necessarily simple substitutes for CFCs. HFC-134a refrigerant, for example, is incompatible (due to different molecular polarity) with common mineral oils normally used with CFCs. An important property of any refrigerant and oil combination is the ability to dissolve with one another. It is therefore necessary to use synthetic lubricants like Polyol-Ester (POE) or polyalkylene glycol (PAG). Refrigerant HFC-134a is used in screw and centrifugal compressors at higher speeds and its phase out is governed by the Kigali Amendment. Refrigerant HCFC-123 is still used mainly in centrifugal compressors and is being phased out in accordance with the Montreal Protocol.

1.2. Later Generation Refrigerants and Natural Refrigerants

1.2.1. Hydrofluoroolefins (HFO) and Hydrochlorofluoroolefins (HCFO)

New refrigerants with low GWP and zero ODP are now being developed and phased in by the global refrigeration and air conditioning industries. Two promising new refrigerants are low pressure refrigerant HCFO-1233zd(E) and medium pressure refrigerant HFO-1234ze(E) and various blends containing these refrigerants. The GWP of the new refrigerants is less than five and the ODP is zero. These refrigerants have a very short atmospheric lifetime, thus they are chemically more active, which can lead to corrosion or material incompatibility, especially in the presence of moisture, adding an extra element in the equation. Besides all this, the thermodynamics of the new refrigerants make the oil-refrigerant mixtures more prone to having high concentrations of refrigerants (30 % or higher) during important times of the operation cycle.

1.2.2. Natural refrigerants

Natural refrigerants such as ammonia (NH3, R717) and carbon dioxide (CO2, R744) are also used increasingly, although ammonia has limitations due to some toxicity and flammability, and carbon dioxide is used at high pressures, which explains why mainly reciprocating compressors are applied for CO2 cycles.

Ammonia (NH3, R717) is toxic for humans, but its strong smell makes leaks easily detectable. There are different grades of ammonia based on application with different requirements on the water content. Industrial-grade anhydrous ammonia, commonly called metallurgical or refrigeration grade, has very little water contamination. Refrigeration-grade ammonia has a maximum of about 150 ppm water content. Ammonia is flammable and has a lower explosive limit (LEL) of 15 percent and an upper explosive limit (UEL) of 28 percent. When the ammonia vapor is mixed with a miscible oil, the LEL can be as low as 8 percent.

Considering only its thermodynamic properties CO2 (R744) is not the ideal refrigerant candidate. However, this is compensated by its very good heat transfer coefficient, very low viscosity and its relatively insensitiveness to pressure losses. From the environmental point of view, CO2 is very attractive, with OPD equal to zero and GWP equal to 1, and CO2 is non-corrosive and basically non-toxic. Recently, mixtures of CO2 and ammonia are considered to increase efficiency. NH3/CO2 mixture is extremely efficient for low and very low temperature applications (below -40 °C). And CO2 is also used in supercritical conditions (sCO2) for higher temperature regimes.

2. Lubrication with Refrigerants

In most refrigerant compressors (except e.g. such using magnetic bearings) the refrigerant plays a role in the lubrication of the bearings, either as a component in the oil-refrigerant mixture or as the sole bearing lubricant in pure refrigerant lubrication (PRL) applications. In any case, the knowledge of the lubricating properties of the refrigerants is needed. And of course, lubrication and tribological aspects are important parameters in the estimation of bearing life and other aspects like frictional losses.

2.1. Model for Oil-Refrigerant Mixtures

Studies have been carried out in the past to model lubrication properties of oil-refrigerant mixtures. In reference [7] the Eyring theory is used to derive equations for the piezo-viscosity coefficient and viscosity:

$$\boldsymbol{\alpha_{mix}} = \frac{\boldsymbol{m s_{ref}(\alpha_{ref} - \alpha_{lub})}}{\boldsymbol{s_{ref}(m-1)+1}} + \boldsymbol{\alpha_{lub}} \tag{1}$$

$$\boldsymbol{ln(\eta_{mix})} = \left(\boldsymbol{ln\,\eta_{ref} - ln\,\eta_{lub}}\right)\left(\frac{\boldsymbol{m s_{ref}}}{\boldsymbol{(m-1)s_{ref}+1}}\right) + \boldsymbol{ln\,\eta_{lub}} \tag{2}$$

where $\boldsymbol{m = M_{lub}/M}$, being $\boldsymbol{M}$ the molecular mass of the component.

Bair [16] offers an alternative mixing law to equation (2). Reference [11] offers an adaptation to these equations in consideration of any refrigerant and lubricant, see equations (3) and (4). Equations (1) and (2) are modified with constants (k_{al}, k_{et}) multiplying the molecular mass ratio, as follows:

$$\boldsymbol{\alpha_{mix}} = \frac{\boldsymbol{k_{al} m s_{ref}(\alpha_{ref} - \alpha_{lub})}}{\boldsymbol{s_{ref}(k_{al} m-1)+1}} + \boldsymbol{\alpha_{lub}} \tag{3}$$

$$\boldsymbol{ln(\eta_{mix})} = \left(\boldsymbol{ln\,\eta_{ref} - ln\,\eta_{lub}}\right)\left(\frac{\boldsymbol{k_{et} m s_{ref}}}{\boldsymbol{(k_{et} m-1)s_{ref}+1}}\right) + \boldsymbol{ln\,\eta_{lub}} \tag{4}$$

These constants are calibrated from measurements of viscosity of different oil-refrigerant mixtures, so calibration functions that depend on oil viscosity and working temperature can replace the originally constant parameters $(\boldsymbol{k_{al}}, \boldsymbol{k_{et}})$, thus,

$$\boldsymbol{k_{al} = f(\eta_{0.lub}, T)} \quad \text{and} \quad \boldsymbol{k_{et} = f(\eta_{0.lub}, T)} \tag{5a, 5b}$$

These equations have been proven to be more accurate than the original ones for several mixtures of refrigerants and oils when compared with measured values [19].

Now, in rolling bearing life calculation the lubrication quality parameter kappa (κ) is used as a measure of the lubrication condition in the bearing. This parameter is defined as the ratio between the actual viscosity used in the bearing at the working temperature and the required viscosity recommended by the manufacturer and it is properly defined and explained in ISO 281:2007 [34]. The required viscosity is a parameter provided by the bearing manufacturer. Thus, the only parameter that remains to be estimated for calculation of κ in oil-refrigerant mixture conditions is the actual lubricant viscosity and equation (2) can be used for this purpose.

For bearing life calculations, reference [4] recommends increasing the required kinematic viscosity of the bearing (ν_1) calculated as oil by multiplying it with a factor $f = 3$, for HFC-134a and POE oil. This is to account for the reduction in piezo-viscosity plus other unaccounted effects (e.g.: lubricity reduction and chemical aggressiveness) in the actual viscosity estimated with equation (2). For very compressible refrigerants an extra increase factor should be derived for correction in the final calculated κ, but here this will be disregarded. Bearing life can be calculated from this final adjusted value of κ called $\kappa_{oil/ref}$. Therefore, according to [11] it is:

$$\boldsymbol{\nu_{adj} = \nu f_{adj}} \tag{6}$$

where ν is the actual kinematic viscosity calculated as oil but with the value obtained from the mixture. Thus,

$$\boldsymbol{\kappa_{oil/ref} = \kappa_{oil}(f_{adj})} \tag{7}$$

with, κ_{oil} being the lubrication quality factor calculated as oil but with the viscosity of the mixture. And, by neglecting the difference in compressibility between oil and refrigerant,

$$\boldsymbol{f_{adj} = \left[\left(\frac{\alpha_{mix}}{\alpha_{oil}}\right)^{0.7}\frac{1}{f_s}\right]} \tag{8}$$

Where

$$\boldsymbol{f_s = 1 + (f - 1)tanh\left(\frac{4s_{ref}}{s_{ref.c}}\right)} \tag{9}$$

and $\boldsymbol{f}$ being the safety factor as described by Meyers [4], but because it was initially introduced by Jacobson it will be called the "Jacobson" safety factor here, defined at a reference refrigerant dilution fraction $\boldsymbol{s_{ref.c}}$. Notice that f_s will become 1 when the dilution of refrigerant s_{ref} becomes zero i.e. when there is no refrigerant mixed into the oil.

The above model has been substantially validated with several HFO, HCFO and HFC refrigerants, but it is likely that some adaptation will be required for Ammonia and CO2. Unfortunately, not much experimental data are available in the literature to either verify this model or to adapt it for those important natural refrigerants. For example, when it comes to ammonia only limited literature is available [25], mainly on the development of Daniel plots for mixtures of ammonia and various oils with dilution rates up to 6% ammonia in saturation conditions. And on the piezo-viscosity behaviour of ammonia no information has been found, (except for relatively low pressures [26], though it is likely that this simple molecule - similar to CO2 - will be an iso-viscous liquids, similar to water. So this could be a "safe" approach for design purposes until more data becomes available. Equation (1) is still valid for $\alpha_{ref} = 0$.

3. Rolling Bearing Life and Other Bearing Failure Modes

The typical life-critical issues in refrigerant compressor bearings, where usual maximum Hertzian pressures are below 2 GPa, are surface-related problems, caused by e. g. poor lubrication and contamination, which leads to wear and/or surface distress, also known as micropitting. Other surface failure modes are also possible e.g. smearing (or adhesive wear), however, those are less often observed in these applications since in general no large rotational accelerations are experienced. Thus, most of the effort invested to increase the life of the bearings goes to preventing or delaying surface distress from poor lubrication conditions. This is why the authors in an earlier publication [2] have included a special section to this topic. Details about performed reliability studies can be found in the following publications: on the accuracy of high reliability values in bearing life calculation [43] and on the accuracy of comparing life models with endurance testing [44].

In [2], the authors also included a modelling example for indications of surface distress occurrence and ways of mitigation. Surface distress is thus strongly influenced by high oil dilution rates, low viscosities of the oil-refrigerant mixture, low piezo-viscosity and higher compressibility of the refrigerants, plus sometimes some corrosive attack due to the chemical activity of the refrigerants, especially in combination with moisture. Note that in [27] a new bearing life model is described that separates the surface distress from subsurface fatigue. This model – which is briefly described in the next section - should be better suitable to handle life calculations in this type of diluted applications, not only because of its better link to physics but also due to its flexibility to accommodate different surface related failure modes.

3.1. The Generalized Bearing Life Model (GBLM)

Following [27] the bearing life equation associated to a reliability (S) can be written as:

$$\boldsymbol{L} = \frac{\left[ln\left(\frac{1}{S}\right)\right]^{1/e}}{u}\left[\bar{A}\int_{V_v}\frac{\langle\sigma_v-\sigma_u\rangle^c}{z^h}dV_v + \bar{B}\int_A\langle\sigma_s-\sigma_u\rangle^c dA\right]^{-1/e} \tag{10}$$

For nomenclature see the mentioned references. In rolling bearings the chosen reliability is usually 0.9, thus the life L will be known as the L_{10} life of the bearing. The first integral in equation (10) is the usual subsurface term [28], while the second integral is a new integral accounting for the fatigue damage at the surface, which can be assessed with a numerical model for e.g. surface distress [29] or indentations [30], etc.

In order to understand the different tribological processes happening at the surface of a rolling bearing, one can take the example of hybrid ceramic bearings, i.e. bearings that use rolling elements made out of ceramic materials instead of steel. It is well known that hybrid bearings are more resilient against poor lubrication conditions and indentations than steel-steel bearings [31, 32]. Therefore, the assessment of the surface integral in equation (10) for this type of bearings is customized [27, 33]. If the new GBLM model is applied [33] much longer lives for hybrid than steel-steel bearings are predicted under difficult conditions, i.e. poor lubrication and contamination. Of course, in case of good and clean lubrication conditions but heavy loads, subsurface fatigue will dominate and hybrid bearing life will be penalised by GBLM due to the higher elastic modulus of ceramics creating somewhat higher contact stresses – which is consistent with test results.

3.2. Bearing Damage due to Particles and Indentations

Particle entrapment in all-steel and hybrid bearings is discussed in [35] and [36]. In [35], the indentation process is described in addition to particle entrapment, presenting some modelling tools for this. The entrapment results where summarised with the use of an arbitrary indentation severity parameter defined as:

$$\boldsymbol{IS} = \frac{h_p s_p}{\phi a} \tag{11}$$

Equation (11) refers to the schematics of an indentation profile along the rolling direction as depicted in Figure 3 left. Where *a* refers to the Hertzian semi-width along the rolling direction of the contact.

The reference discusses a simple entrapment model based on the friction coefficient between the particle and the bearing surfaces μ_p and also based on the rolling element diameter D_w and negligible lubricating central film thickness h_c. Giving the maximum size of a spherical particle to be entrapped as:

$$\boldsymbol{d_{p.max}} = \frac{D_w}{cos(tan^{-1}\mu_p)} - \boldsymbol{D_w} \tag{12}$$

The fact that a particle is entrapped in a contact does not necessarily mean that it will indent the rolling surface and generate damage. It all depends on size, material and hardness of this particle in relation to the counter face. Very small particles will move through the lubricant film without causing any damage to the mating surfaces. Larger and very hard brittle particles may shatter in the contact producing many smaller particles. Figure 3 (right) depicts some predicted maximum entrapment particle diameters for varying rolling element diameter and particle-surface friction coefficient. The minimum particle diameter according to equation [37] that might still indent the bearing surfaces is also depicted in the plot (as a reference) using the horizontal black dotted line, notice that the minimum steel particle diameter causing an indentation would be of the order of the central film thickness h_c - for a typical heat pump bearing $h_c \approx 0.1\,\mu m$. Notice that experiment estimated values for μ_p steel-steel are reported in [35] of about $\mu_p = 0.15$, but it will be lower for the case of ceramic rolling elements.

Of course, most contamination particles are not spherical, however the model is valid for particle diameters estimated from the shortest particle length, e.g. for a cylindrically shaped particle the parameter d_p would refer to the cylinder diameter and not to the length. The damage produced by indentations in the context of GBLM and EHL contacts is described in detail in [30].

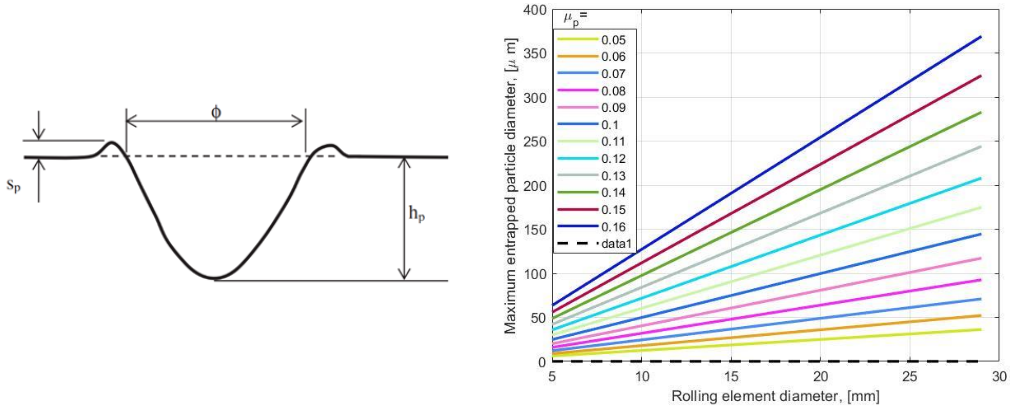

Fig. 3. Left: Schematics of an indentation profile along the rolling direction of the contact.
Right: Maximum entrapment diameter particle as calculated from equation (12). The horizontal black dotted line gives an indication of the minimum steel particle diameter that could produce an indentation on the bearing surfaces according to equation [37].

3.3. Surface Distress Phenomenon

Micropitting is a term widely used in the gearbox industry to describe micro surface spalls and cracks, which sometimes appear on the surface of rolling–sliding contacts. ISO 15243 [38] refers to this failure mode as surface distress or surface initiated fatigue, i.e. the failure of the rolling contact metal surface asperities under poor lubrication conditions and with certain amount of sliding motion causing the formation of (1) burnished areas (glazed; grey stained), (2) asperity microcracks, and (3) asperity micro-spalls, see Figure 4 (left). This failure mode will be described here using the term of surface distress. It is well known that surface distress is the result of the competition between surface fatigue and mild wear [29]. Mild wear (wear at asperity level) modifies the initial topography reducing the asperity heights and also removes fatigued surface layers - refreshing the exposed material and thus delaying surface fatigue. The modelling of this phenomenon is possible, and one of the current authors has developed a numerical model for surface distress [29] that can be applied for the present case of oil-refrigerant mixtures [2]. The model requires 3D digitised roughness data from several positions of the two contacting surfaces (inner - or outer - ring raceway and roller - or ball). A calculation example simulating the fatigue weakening by corrosion from refrigerants is given in [2].

3.4. Integrating Surface Failure Modes in GBLM

The model described in [29] can be used to generate tendency curves by simulating surface distress from using roughness samples of many bearings (all types and sizes) under different loads and lubrication conditions (In the bearing industry the parameter κ is used [34] rather than the film thickness/roughness ratio Λ used in other engineering fields to describe the quality of lubrication). In [29] a curve-fitted equation to model the surface integral representing surface distress is obtained from the simulations. An example of the graphical representation of this equation is depicted in Figure 4 (right), where the normalized value of the surface integral ($\tilde{I}_s$) is plotted for different normalised equivalent loads (P/P_u) under different lubrication qualities (κ).

The surface integral is defined as

$$\boldsymbol{\tilde{I}_s = I_s u^e \left(\frac{C}{P}\right)^p / \left(K\, ln\left(\frac{1}{0.9}\right)\right)} \tag{13}$$

with K being a scaling factor and,

$$\boldsymbol{I_s = \bar{B} \int_A \langle \sigma_s - \sigma_u \rangle^c dA} \tag{14}$$

As can be seen in Figure 4 (right) when lubrication conditions deteriorate (low values of κ) the surface integral increases in value, this is an indication of higher probability of surface distress. When the load increases in general the integral increases, except in the cases of very poor lubrication where the integral is slightly reduced due to the increase of mild wear. The scaling and behaviour of these curves can be adapted for the case of material weakness, e. g. due to corrosion from refrigerants and moisture. Other adaptations are possible depending on the balance wear and fatigue, or with the use of hybrid bearings [27], etc.

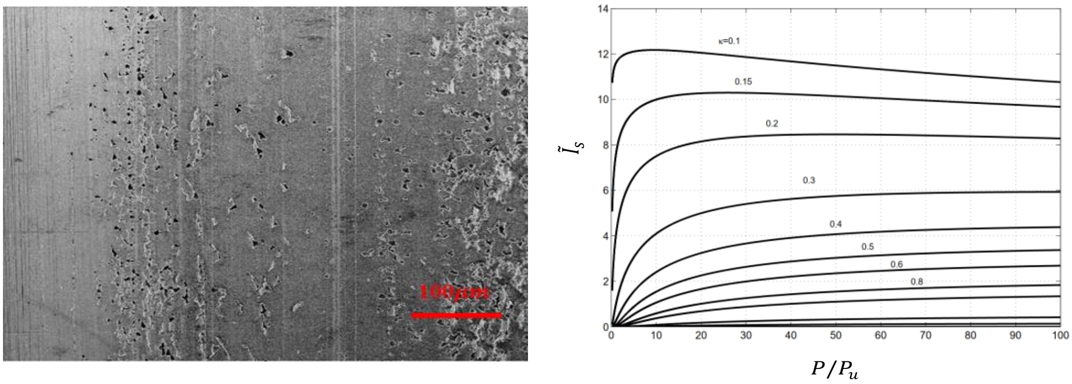

Fig. 4. Left: Surface distress in a raceway bearing surface; Right: Example of normalised surface integral for the GBLM model as a function of normalised equivalent load and lubrication quality in the bearing, as given in [29].

3.5. Relative Surface Fatigue Index in GBLM

One of the advantages of separating the subsurface and surface damage integrals is that they can be compared and in this way an indication of the most fatigued area can be determined. Consider the parameter called “relative surface fatigue index” denoted by the symbol S_R:

$$\boldsymbol{S_R = \frac{I_s}{I_s + I_{ss}}} \tag{15}$$

where I_s is given by equation (14) and I_{ss} by

$$\boldsymbol{I_{ss} = \bar{A} \int_{V_v} \frac{\langle \sigma_v - \sigma_u \rangle^c}{z^h} dV_v} \tag{16}$$

Notice that when $S_R \to 1$ the surface integral is dominant which means that the higher damage is expected at the surface. When $S_R \to 0$ fatigue is taken mainly by the subsurface. This parameter is important because when operating conditions are changed, changes in this parameter will indicate how the surface or the subsurface is being affected. The parameter S_R can also be calculated locally, for example along the raceway profile in a roller bearing, reflecting the effect of the local stresses. Reference [30] shows good examples of the use of S_R for the case of indentations and the interaction with the lubrication quality parameter.

4. Technical Bearing Solutions for Heat Pump Compressors

In the following, some technical solutions to address the challenges in heat pump compressors bearings as indicated above are presented.

4.1. Surface Treatments in Conventional Rolling Bearings

For oil-refrigerant mixtures under poor lubrication and/or high contamination, surface (heat) treatments that have a large potential for rolling bearings are nitriding and carbonitriding. The latter one is an added surface heat treatment with the addition of ammonia to the standard gas carburisation process. This introduces N and C atoms diffusing into the surface steel, improving the fatigue strength and reducing wear. Carbonitrided (HN code) bearings have been tested and compared with normal heat treated bearings in harsh contaminated conditions in [39]. Wear rate was reduced to about 45 % of the normal bearings and bearing life was roughly doubled for those particular test conditions. In addition, the surface distress model described before was used to simulate the behaviour and provide understanding.

4.2. Semi-Hybrid Rolling Bearings

It has been verified by testing [40] that steel rolling bearings which contain at least one rolling element made of ceramic (Si3N4) material can substantially improve performance under poor lubrication and high contamination conditions. Therefore, the next “line of defence” after nitriding or carbonitriding for lubrication with oil-refrigerant mixtures should be this type of rolling bearings, sometimes also called “semi-hybrid” or “self-healing” bearings. The working mechanism is similar to hybrid bearings as explained in [35, 37].

4.3. Raceway Coatings

If a bearing raceway coating has to be chosen in harsh oil-refrigerant lubrication conditions, it is recommended to select a sacrificial coating that will mainly help to run-in the surfaces improving the tribology and smearing itself for the steel surfaces. In this way, it will also protect the surfaces from mild chemical attacks (mild corrosion). This is the case for special bearing-grade black oxide coating [41], which could be a rolling bearing performance improver in mild severity conditions of poor lubrication with oil-refrigerant mixture. An alternative coating solution against corrosion is zinc-coated bearings.

For lower refrigerant dilution rates other coatings may be used that can provide the bearings with some protection. This is the case for special DLC (NoWear™) with and without porosity seal, made to endure in poor lubrication conditions and to fight adhesive wear.

4.4. Hybrid ceramic bearings

For extreme conditions such as dilution rates much higher than say 20%, for very low operating viscosity, and/or very high or low speeds, hybrid bearings comprising ceramic rolling elements (usually made out of Si3N4/silicon nitride) and steel rings are a favourable solution. The reasons for the superior performance of hybrid bearings under such conditions was already explained in section 3.1. In case the chemical conditions in the compressor also bear the risk of corrosive attack, or if the bearings should be lubricated only by the refrigerant in oil-free compressors, the rings need to be made out of a bearing-grade corrosion resistant material with high toughness, see next section (Figure 5).

4.5. High-nitrogen stainless steel hybrid bearings

When the bearing rings cannot be protected by oil or grease against corrosive media in the machine, rings made out of stainless steel are often the most reliable solution. Various stainless material options exist, which have to be selected based on their corrosion resistance as well as their mechanical strength in rolling bearing application. Inner and outer rings made of through-hardened martensitic stainless steel with high nitrogen content show high corrosion-protection capability and fatigue toughness. It is of high importance to tune the steel making and heat treatment process parameters to obtain a fine microstructure that leads to high toughness and so making the steel an excellent material for the demanding conditions in highly loaded rolling contacts.

With thinner lubrication films or in media lubricated applications (like for PRL) also the raceway surface finishing quality tends to play a more important role for the bearing performance. This is why more stringent raceway honing specifications are applied related to surface roughness and imperfections, as well as to manufacturing process cleanliness applied for media lubricated applications.

For extreme conditions, a specific high-nitrogen steel with a dedicated heat treatment for high corrosion resistance, fatigue toughness and also higher temperature stability was developed. It was proven to be a reliable solution for applications like sour gas compressors, cryogenic pumps, and pure refrigerant lubricated compressors. Some examples for media- and mixed-media lubricated applications are presented in [42].

Figure 5: Special hybrid ceramic bearings with super-tough high-nitrogen stainless steel rings

4.6. Cage Material Selection

There is a large variety of cage designs and materials for any bearing type. In demanding application, the cage design choice is often either a massive metal cage or a lightweight polymer cage. Common materials for metal cages are brass, steel, or sometimes also aluminium alloys. Massive metal cages offer high mechanical strength and for very critical applications special pocket designs or surface coatings can be applied. Massive (stress-free) brass cages have also a proven track recorded in ammonia compressors.

Besides standard polymer cages, different high-performance polymers have been developed for bearing applications. Fibre reinforced PEEK (polyether-ether-ketone) is a very robust solution in demanding applications, especially where an extended temperature range, chemical stability, and a lightweight design are required. The combination of ceramic rolling elements and a polymer cage allows for a very low weight of the rotating components in the bearing which are not directly mounted on the shaft or in the housing, reducing their inertia which is especially beneficial in high acceleration or very low load conditions. PEEK cages have also been proven to work in ammonia compressors up to 120°C operating temperature

5. Discussion

The present publication summarises the current status of rolling bearing technology for refrigerant compressor applications, with emphasis on the special conditions found in heat pumps, where the lubrication with oil-refrigerant mixtures is common practice and the bearing operating conditions are harsher than for most other compressor applications. It is suggested that standard bearing technologies for conventional refrigerants and requirements with oil-dilution rates usually lower than 20% will be increasingly challenged by new and more chemically active low-GWP refrigerants (according to Kyoto protocol requirements). Dilution rates up to 60% or even higher in some sections of the thermodynamic duty cycle, and also much higher process temperatures and pressures leads to challenging conditions for the rolling contacts in the bearings.

As a graphical summary of the application of the different rolling bearing technologies, the authors propose the schematics of Figure 6 to describe the possible usage ranges and technology sophistication level of the different rolling bearing technologies discussed in the current paper, as a function of the tribological and chemistry complexity and risks.

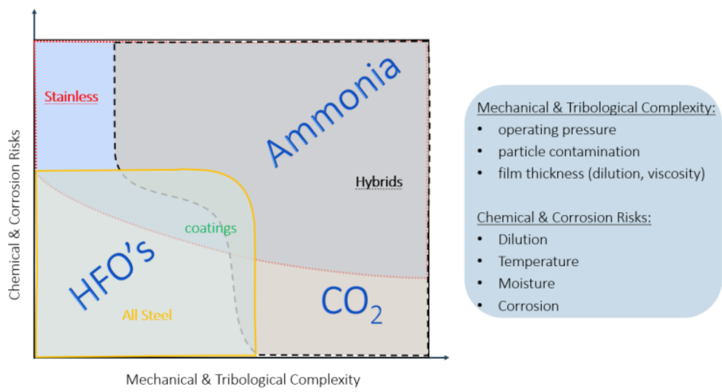

Fig. 6. Schematics of potential solutions for rolling bearings depending on the application and tribological and corrosion risks.

Notice that the ordinate in Figure 6 (mechanical & tribological complexity) represents a qualitative scale where no values are given, but it should illustrate the experience of the authors. The abscissa of the figure represents the risks related to chemical aspects. Note that tribological risks (i.e. due to very low viscosity or speeds) can be mitigated with the use of hybrid bearings. High chemical risks (high corrosion potential) can be mitigated with the use of special bearing-grade stainless steels. In the transition area between all-steel bearings to hybrid bearing and stainless steel rings, the use of special heat treatments, coatings, or semi-hybrid bearings can be an appropriate fit-for purpose solution – in economic and as well as technical terms.

6. Conclusions

The following conclusions can be drawn based on the above presented research and also practical experience:

1. Standard all-steel rolling bearing technology is currently challenged by the arrival of new, more environmentally friendly refrigerants in the industrial compressor sector, which bring higher refrigerant dilution rates, more chemically active compounds and – for heat pumps or also data centre chillers - much higher temperatures.
2. As a response to this challenge, different rolling bearing technologies are proposed, like the use of special heat treatments as well as hybrid ceramic rolling bearings.
3. A better understanding still needs to be obtained for specific lubrication mechanisms and failure initiation mechanisms in heat pumps with low-GWP refrigerants. This implies work to be done on tribological refrigerant characterisation, tribology contact modelling, tribo-testing, and bearing testing under representative compressor conditions.

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