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ABSTRACT

In addition to the electrical and the thermodynamic losses in hermetic compressors, mechanical losses have a significant influence on the performance of the compressor. In the present paper the friction losses in the journal bearings of a hermetic reciprocating compressor are investigated using numerical methods. A dynamic model is set up to solve the Reynolds equation using a finite volume approach to calculate the pressure field in each journal bearing. The calculation of the fluid film thickness is accomplished with the formulas of the parallel gap. The resulting hydrodynamic forces are equated with forces obtained by a dynamic multibody model of the compressor crank drive to calculate the transient orbit movement of the bearing. Based on the movement of the crankshaft at steady-state conditions, the shear stresses in the gap between crankshaft and housing can be calculated. Thus the cycle averaged friction power loss can be determined. To consider effects such as surface roughness of the bearings or possible contacts between the solids, correlations found in literature are implemented. The present method is used to assess the friction power loss of the journal bearings during the operation with different oil viscosities. The simulated data is verified by simple analytical friction loss calculations based on shear stresses in the Couette flow between bearing and housing.

1. INTRODUCTION

Hermetic compressors either running in ON-OFF or variable speed mode are widely used in domestic refrigeration appliances. The hermetic design of the compressors requests a durability of all the compressor parts of at least 15 years, which is specified by the producer of the cooling appliance. Another difficulty caused by the hermetic design is the complex interaction between electrical, mechanical and thermodynamic losses. The identification of the individual losses is one of the main tasks in the development process of modern hermetic reciprocating compressors. In addition to challenging experimental investigations, simulation models are increasingly used. An example of simulation application in compressor development is the quantification of friction losses in moving parts, especially in journal bearings. Due to the low absolute level of friction power losses (relative level with regard to compressor power is about 15 %) and the hermetic design, classical strip-down methods which are used in engine development are not suitable for the investigation of friction losses in reciprocating compressors for domestic refrigeration appliance. Several studies dealing with the modelling of the journal bearing dynamic behaviour are available. The studies resemble each other concerning the procedure to solve the friction loss problem of journal bearings. The procedure can be summarized as follows: (i) a multi body simulation of the compressor crank drive is carried out to get the dynamic loads on the journal bearings, (ii) the Reynolds equation is solved using numerical schemes to calculate the hydrodynamic forces in the bearings, (iii) a Newton-Raphson algorithm is used to get the movement of the shaft orbit and (iv) based on the shaft movement the friction losses can be determined. A distinction between these studies can be made concerning the modelling of the Reynolds equation or the model accuracy. The use of the short bearing approximation can be

found in e.g. Estupinan and Santos (2009) or Kim *et al.* (2012). A comparison between results of the short bearing approximation and the finite bearing model was carried out by Chieh *et al.* (2007). Duyar and Dursunkaya (2002 and 2006) analysed the dynamic behaviour of compressor journal bearings considering the elastic deformation of the shaft using finite element discretization. An investigation of compressor journal bearings considering mixed lubrication was presented in the work of Matsui *et al.* (2010).
The present paper deals with the investigation of the friction losses in compressor journal bearings for being used as input data for a holistic thermal model of a hermetic compressor. The method is similar to the previous explained procedure using finite bearing model for the solution of the Reynolds equation. Furthermore, mixed lubrication models found in the literature are considered. The gap in the bearing is approximated as parallel. The simulation is used to calculate the influence of the oil viscosity and gap width on the friction losses. The simulated data is compared to simple analytical friction loss calculations based on shear stresses in the Couette flow to give an assessment of the required accuracy of the model to fulfil the requirements of a thermal compressor model.

2. KINEMATICS AND DYNAMICS

The friction analysis of the compressor journal bearings requires the determination of the bearing forces. For this purpose a multibody dynamics model of the piston-conrod-crankshaft system is developed. A detailed description of the friction behaviour of the junctions between piston and conrod respectively conrod and crankshaft is not carried out in the present study. Moreover, the misalignment of the piston is not considered. Figure 1 shows the coordinate systems used for the kinematic description of the crank mechanism parts. The inertial coordinate system is located in the static position of bearing A. The misalignment of the crankshaft due to the bearing eccentricity is considered in the distance and rotating vector of the crankshaft regarding to the initial coordinate system.

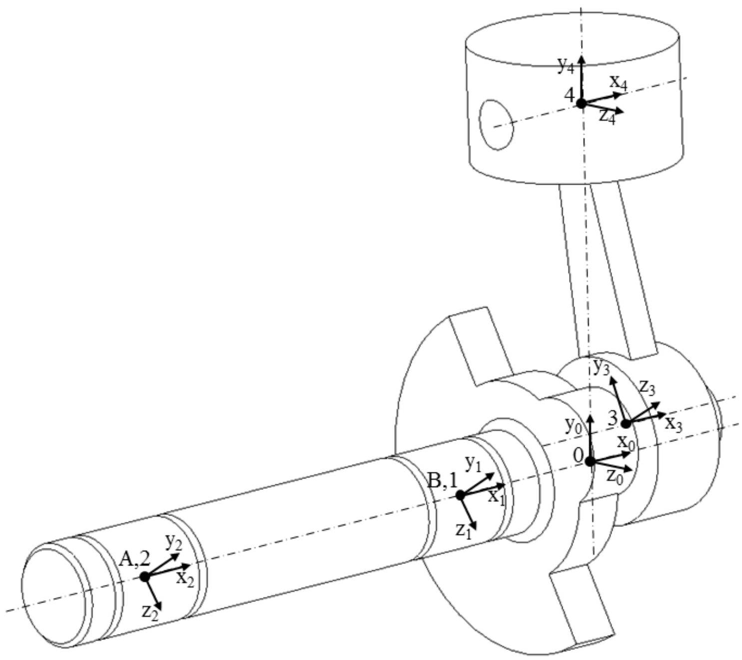


Figure 1: Coordinate systems

Using the Newton-Euler method the equations of motion can be expressed for each part.

Piston:

$$m_{Pi} \cdot \ {}_{I}\boldsymbol{a}_{Pi} = \ {}_{I}\boldsymbol{F}_{p} - \ {}_{I}\boldsymbol{F}_{4} + \ {}_{I}\boldsymbol{F}_{Cyl} \tag{1}$$

Conrod:

$$m_{Cr} \cdot \ {}_{I}\boldsymbol{a}_{Cr} = \ {}_{I}\boldsymbol{F}_{4} - \ {}_{I}\boldsymbol{F}_{3} \tag{2}$$

$${}_{I}\Theta_{Cr} \cdot \dot{\omega}_{Cr} + m_{Cr} \cdot \ {}_{I}\boldsymbol{r}_{3-Cr} \times \ {}_{I}\boldsymbol{a}_{Cr} = \ {}_{I}\boldsymbol{r}_{3-4} \times \ {}_{I}\boldsymbol{F}_{4} \tag{3}$$

Crankshaft:

$$m_{Cs} \cdot \ {}_{I}\boldsymbol{a}_{Cs} = \ {}_{I}\boldsymbol{F}_{A} + \ {}_{I}\boldsymbol{F}_{B} + \ {}_{I}\boldsymbol{F}_{3} \tag{4}$$

$${}_{I}\Theta_{Cs} \cdot \dot{\omega}_{Cs} + \omega_{Cs} \times \left({}_{I}\Theta_{Cs} \cdot \omega_{Cs} \right) + m_{Cs} \cdot \ {}_{I}\boldsymbol{r}_{1-Cs} \times \ {}_{I}\boldsymbol{a}_{Cs} = \ {}_{I}\boldsymbol{r}_{1-2} \times \ {}_{I}\boldsymbol{F}_{A} + \ {}_{I}\boldsymbol{r}_{1-3} \times \ {}_{I}\boldsymbol{F}_{3} + \boldsymbol{T} \tag{5}$$

The set of equations gives the time dependent functions of the reaction forces in the main journal bearings according to the cylinder pressure. The cylinder pressure can be determined by experiments or computational fluid dynamics (CFD), respectively. In the present study the cylinder pressure is given by CFD simulation of the compressor gas line.

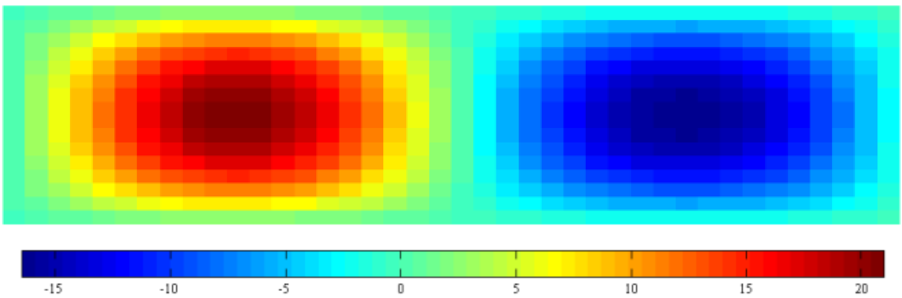
$${}_{I}\boldsymbol{F}_{p} = \begin{pmatrix} 0 \\ p_{CFD} \cdot A \\ 0 \end{pmatrix} \tag{6}$$

3. HYDRODYNAMIC FORCES

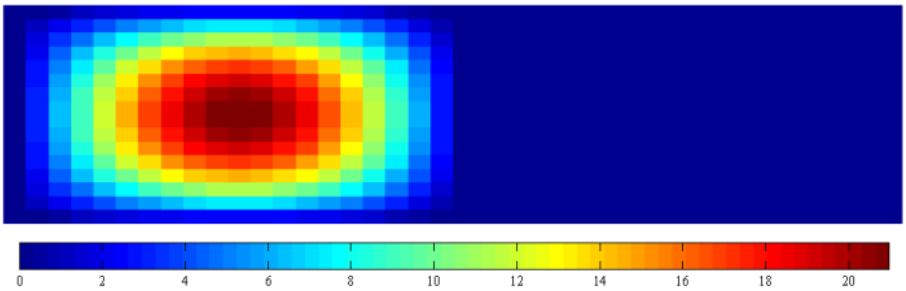
The modelling of the hydrodynamic forces in the journal bearings is carried out by solving the Reynolds equation which links the hydrodynamic pressure and the fluid film thickness.

$$\frac{\partial}{\partial x}\left(\frac{\rho\, h^3}{12\eta}\frac{\partial p_h}{\partial x}\right) + \frac{\partial}{\partial y}\left(\frac{\rho\, h^3}{12\eta}\frac{\partial p_h}{\partial y}\right) = \frac{(u_I + u_{II})}{2}\frac{\partial(\rho h)}{\partial x} + \frac{\partial(\rho h)}{\partial t} \tag{7}$$

The terms on the left-hand side of the Reynolds equation represent the Poiseuille flow, the terms on the right-hand side represent the Couette flow and the displacement flow, respectively. To solve this kind of partial, inhomogeneous elliptic differential equation, numerical methods have to be applied. In the present study the Reynolds equation is solved using the finite volume approach. The fluid film area in each bearing is divided in a certain number of cells and the terms of the Reynolds equation can be discretised. To avoid negative fluid film pressures, the Gümbel boundary condition is used which sets negative pressure values to zero. Figure 2 shows the hydrodynamic pressure distribution in bearing A without (a) and with (b) Gümbel boundary condition.



(a)



(b)

Figure 2: Hydrodynamic pressure in bearing A without (a) and with (b) Gümbel boundary condition [MPa]

To create a link between the eccentricity of the bearing and the fluid film thickness, the gap in the bearing is assumed to be parallel. According to the geometric relations shown in Figure 3 the fluid film thickness can be expressed as follows (Woschke, 2013):

$$h(\varphi) = (r_I - r_{II}) - ex \cdot \cos(\varphi - \xi) \tag{8}$$

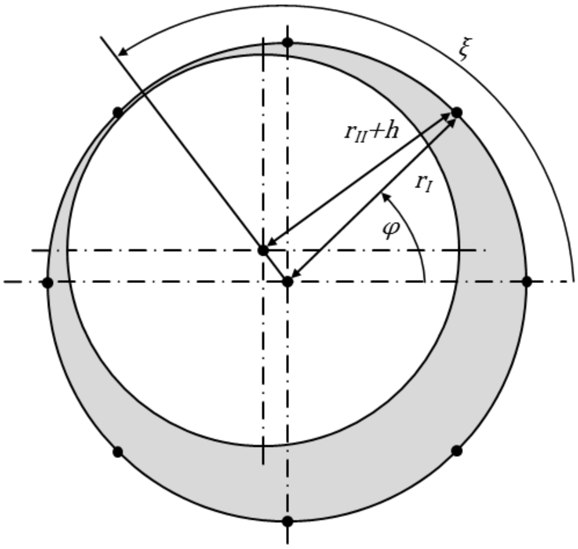


Figure 3: Determination of the fluid film thickness

The basic form of the Reynolds equation (7) considers only the macro geometry of the bearing parts. Patir and Cheng (1978 and 1979) introduced an extended form of the Reynolds equation to consider also micro effects due to surface roughness. The authors used flow factors which depend on the roughness of the bearing surface and called the new form Average Reynolds Equation.

$$\frac{\partial}{\partial x}\left(\phi_x \frac{\rho\, h^3}{12\eta} \frac{\partial \overline{p_h}}{\partial x}\right) + \frac{\partial}{\partial y}\left(\phi_y \frac{\rho\, h^3}{12\eta} \frac{\partial \overline{p_h}}{\partial y}\right) = \frac{(u_I + u_{II})}{2}\left(\frac{\partial\left(\rho \overline{h_T}\right)}{\partial x} + \sigma_\delta \frac{\partial \phi_S}{\partial x}\right) + \frac{\partial\left(\rho \overline{h_T}\right)}{\partial t} \tag{9}$$

For a more detailed description of the flow factor determination the interested reader is referred to the primary works by Patir and Cheng (1978 and 1979). To calculate the possible solid contact between the surfaces the model of Greenwood and Tripp (1970) is used in the present study.

4. NUMERICAL ALGORITHM

The solving procedure starts with the input of the geometric data of the compressor crank mechanism parts. Initial values of eccentricity *ex* and minimum fluid film thickness angle ξ are set to zero as well as their time derivatives. Time resolution is fixed for the whole procedure and corresponds to a crank angle of two degrees. Starting at a certain position of the crankshaft (here crankshaft angle is zero) the cylinder pressure of the CFD calculation is read into the bearing simulation program. Solving the multibody dynamic system, the bearing forces can be calculated for the current crank angle. The calculation of the hydrodynamic bearing forces starts with the determination of the fluid film thickness according to (8) with the defined start values of the new time step. The start values of the new time step are the calculated eccentricity *ex* and minimum fluid film thickness angle ξ of the previous time step or the initial values if it is the first time step, respectively. To get the orbit of the crankshaft for the current time step, a two-dimensional Newton algorithm depending on *ex* and ξ is used (10). In each iteration step the Averaged Reynold Equation (9) is solved numerically for both bearings and the hydrodynamic bearing forces are compared with the calculated bearing forces of the multibody system regarding absolute value and force direction. If the deviation between hydrodynamic and multibody forces in both bearings is higher than the convergence criteria β_F the iteration procedure is repeated, otherwise the algorithm proceeds to the next time step. The numerical algorithm is carried out until the deviation of the time-dependent shaft orbit between the current and the previous cycle is lower than the convergence criteria β_C and steady-state conditions are reached.

$$\begin{gathered} \boldsymbol{x} = \binom{ex}{\xi} \\ \boldsymbol{f}(\boldsymbol{x}) = \boldsymbol{F}_{MBS} - \boldsymbol{F}_{Rey} \\ \boldsymbol{x}_{n+1} = \boldsymbol{x}_n - \left(\boldsymbol{J}_f(\boldsymbol{x}_n)\right)^{-1} \boldsymbol{f}(\boldsymbol{x}_n) \end{gathered} \tag{10}$$

5. RESULTS

The present numerical bearing model is used to simulate hydrodynamic bearings of a hermetic reciprocating compressor running at 3000 rpm. The pressure in the cylinder is determined by CFD simulation of the compressor gas line considering fluid-structure interaction of the compressor valves. Operating conditions are set to -23 °C for evaporating temperature and 45 °C for condensing temperature (R600a), respectively. The baseline value for oil viscosity is 8 cSt and bearing clearance is 5 µm. Bearing width of bearing A is one quarter less than bearing B, whereas the shaft diameter is equal for both bearings.

Figure 4 shows the shaft orbits of both bearings as a function of the oil viscosity. The illustrated curves represent one crankshaft revolution after steady-state conditions are reached which is the case after 7-8 cycles depending on the considered configuration. According to the higher forces due to the compression process at bearing B the eccentricity ratio of bearing B is higher than bearing A. An increase of the oil viscosity results in a decrease of the shaft deflection. Furthermore, higher oil viscosities damp the relative motion between the bearing parts and thus the crankshaft orbit is focused on a smaller region. The influence of the shaft orbit on the bearing clearance is shown in Figure 5. The curves show an increase of the eccentricity ratio of both bearings with an increased bearing clearance. This behaviour is a result of the nonlinear relation between the bearing pressure and the local oil film thickness. Small bearing clearance already yields in higher oil pressure at small shaft movement, so the deflection of the crankshaft is kept at low values. This effect can be seen in the shape of the crank orbit which is smoother at smaller bearing clearance values.

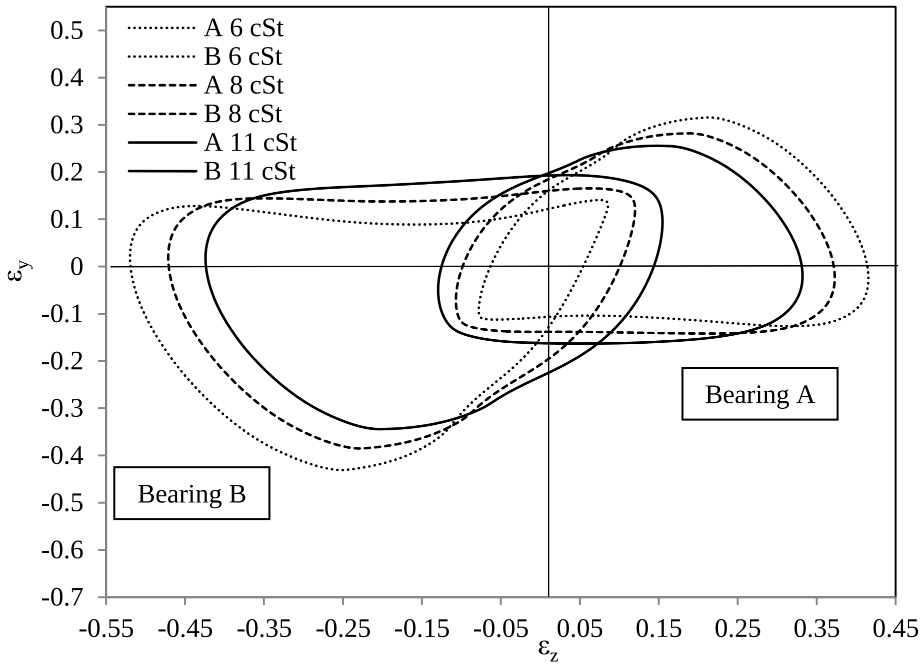


Figure 4: Shaft orbits with different oil viscosity

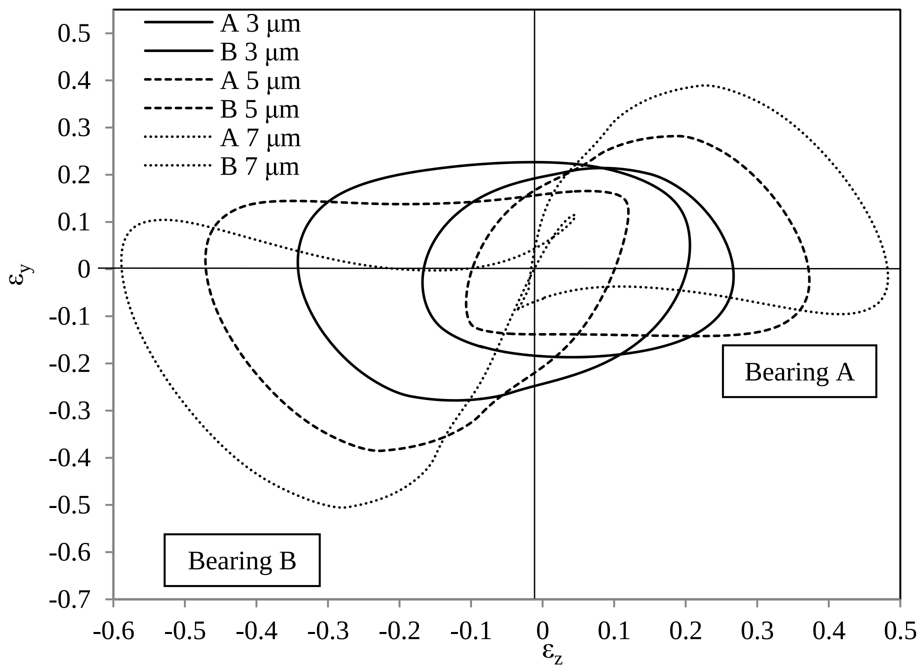
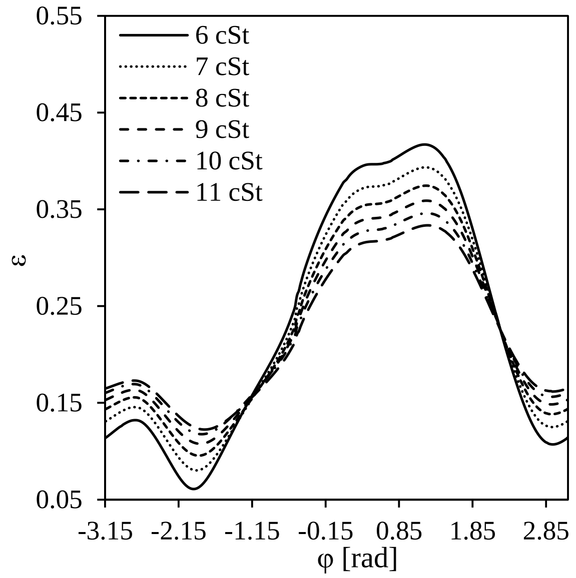
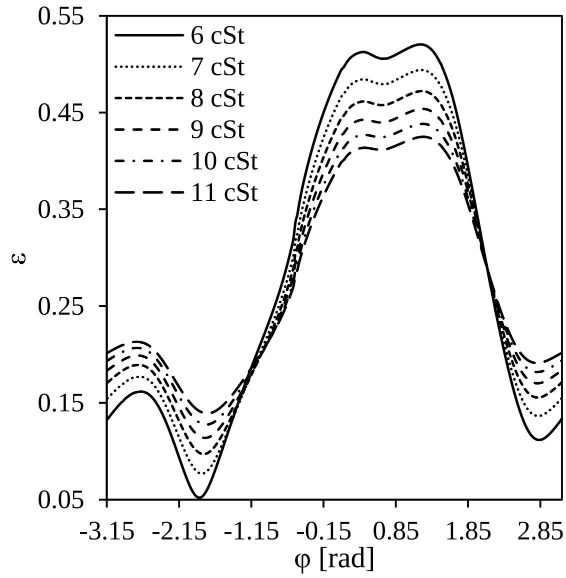


Figure 5: Shaft orbits with different bearing clearances

An addition illustration of the simulated shaft movement in terms of the absolute eccentricity ratio ε over the crank angle φ can be seen in Figure 6 and 7. According to the shaft orbit curves the eccentricity ratio of bearing B is higher than of bearing A. The figures show the peak in the eccentricity curve in the area of the top-dead centre of the compressor piston and the resulting high reaction forces in the piston-conrod-crankshaft system. The nonlinear dependence between bearing pressure and local oil film thickness (and its time derivatives) can also be seen in the eccentricity ratio curves. Curves of different viscosity or gap width can intersect especially in regions of low reaction forces and small eccentricity ratios.



(a)



(b)

Figure 6: Eccentricity ratio over one crank shaft revolution of bearing A (a) and bearing B (b)

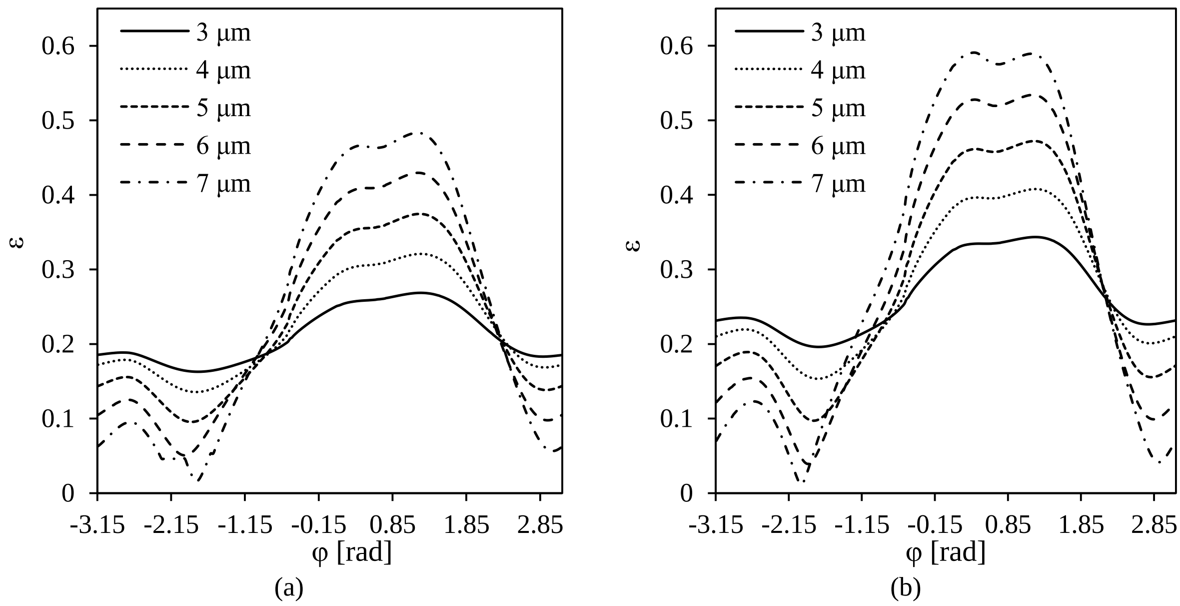


Figure 7: Eccentricity ratio over one crank shaft revolution of bearing A (a) and bearing B (b)

Results of the determination of the friction power loss in the bearings are shown in Figure 8. Friction power loss values of bearing B are higher than of bearing A. This is because of the higher reaction forces in addition with the larger bearing width of bearing B which results in higher friction surface. Viscosity variations show a linear dependency of the friction power loss increasing with higher viscosities. The numerical simulation of the friction power loss with different gap widths shows a stronger power loss increase at smaller gap widths. A comparison between numerical results and simple analytical Couette flow calculation show constant curve offset for a viscosity variation. For gap width investigation, the offset between numerical and analytical results increases for higher gap widths. Generally, the deviation between numerical and analytical results is significantly higher for higher loaded bearings like bearing B in the present study.

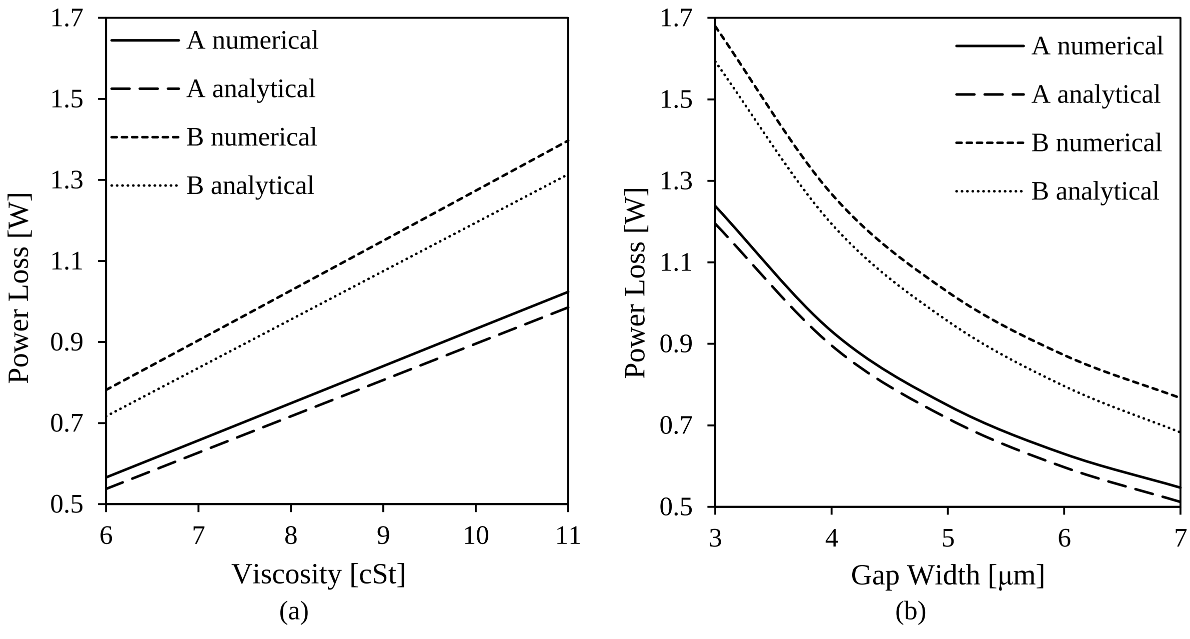


Figure 8: Power loss as function of viscosity (a) and gap width (b)

A comparison between the simulations with and without the usage of Patir and Cheng's (1978 and 1979) Average Reynolds Equation yields in friction power loss deviations below 1 %. It should be mentioned, that the maximum determined eccentricity ratios of both bearings in the present study are significant smaller than 1 so the influence of the surface roughness on the friction power loss is also small. In simulations of compressor operating points with higher cylinder pressure which results in higher reaction forces in the bearings the eccentricity ratio will increase and thus the surface roughness influence on the friction power loss will also increase. Similar considerations can be made regarding the influence of the shaft movement in the multibody dynamics. Stronger shaft movement influences the dynamics of the piston-conrod-crankshaft system and consequently the hydrodynamics of the bearings. In the present study the influence of the shaft movement consideration on the friction power loss is below 1 %.

6. CONCLUSION

A numerical investigation of the hydrodynamic bearing system of a hermetic reciprocating compressor is shown in this study. The model is based on the solution of the Average Reynolds Equation of Patir and Cheng (1978 and 1979) to compare hydrodynamic bearing forces with reaction forces gained by multibody dynamics calculation which results in a two-dimensional Newton algorithm problem for the fluid film properties ex and ξ. The presented algorithm is used to calculate the dynamic behaviour of the journal bearings of the compressor operating at -23 °C evaporating temperature and 45 °C condensing temperature (R600a). A parameter study for several oil viscosities and gap widths are carried out to show the abilities of the method for the compressor development. Additionally, the present algorithm is used to calculate the friction power losses of the journal bearings. In terms of the friction power loss modelling, the following conclusions can be emphasized:

- The comparison between numerical and analytical (simple Couette flow) simulations show a similar dependence of the parameters, but analytical simulation underestimates friction power losses especially at higher loaded bearings.
- The use of the complex Averaged Reynolds Equation of Patir and Cheng (1978 and 1979) can be avoided at lower loaded bearings with small shaft movements. In this case, the classical Reynolds equation can be used.
- A similar conclusion can be made for the consideration of the shaft movement in the multibody dynamics. Small shaft movements can be neglected here.

NOMENCLATURE

$\boldsymbol{a}$	acceleration vector	(m/s²)
A	area	(m²)
ex	eccentricity	(m)
$\boldsymbol{F}$	force vector	(N)
h	fluid film thickness	(m)
m	mass	(kg)
p	pressure	(N/m²)
$\boldsymbol{r}, \boldsymbol{x}$	distance vector	(m)
t	time	(s)
$\boldsymbol{T}$	torque moment vector	(N m)
u	velocity	(m/s)
x, y, z	distance components	(m)
Subscripts		
0, 1, 2, 3, 4	coordinate system index	
A, B	bearing index	
Cr	conrod	
Cs	crankshaft	
Cyl	cylinder	
I	inertial system	
I, II	shaft, bearing	
MBS	multi body system	
Pi	piston	
Rey	Reynolds forces	
Greek symbols		
β	convergence criteria	(-)
ε	relative eccentricity	(-)
η	dynamic viscosity	(N s/m²)
Θ	inertia tensor	(kg m²)
ξ	minimum fluid film thickness angle	(°)
ϕ_x, ϕ_y, ϕ_s	flow factors	(-)
φ	attitude angle	(°)
ω	rotational speed	(s^{-1})
ρ	density	(kg/m³)
σ_δ	standard deviation of the combined roughness	(m)

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