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INFLUENCE OF COMPRESSOR SPEED FLUCTUATION ON BEARINGS DESIGN

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ABSTRACT

Bearings reliability can be affected by multiple factors. Depending on the load and inertia of the rotating parts, the speed of the compressor during one cycle can fluctuate significantly. The main research objective is to investigate how speed fluctuation affects bearings design. The question is whether a fast variation of speed can significantly impact hydrodynamic and asperity power losses, contact pressures or minimum oil film thickness, which potentially influence bearing reliability. The software GT-Suite was used to estimate the speed fluctuation for several compressor application conditions. Then, the software AVL Excite was used to run multi-body dynamics simulation with elastohydrodynamic (EHD) bearings and flexible bodies to identify their critical conditions. A simplified model with one EHD bearing was used to estimate the effect of speed fluctuation on peak asperity contact pressure, showing this methodology´s applicability in minimizing the impact of speed fluctuation on compressor reliability.

Keywords: Bearings, Speed, Fluctuation, Asperity, Reliability

1. INTRODUCTION

Reciprocating compressors are highly dynamic systems and some of them with relatively low cranktrain inertia. High loads in conjunction with low inertias result in large fluctuations of speed within individual cycles, which in turn influence the behavior of the sliding bearings. When trying to increase the reliability a comprehensive bearing design study is needed to minimize peak asperity contact pressure. This paper achieves this by using design of experiments (DoE) methods to determine relevant design and operating parameters.

1.1. Kinematics and Dynamics

Inertia is termed most simply as the resistance of an object to any change in its velocity, be it a translational or angular velocity. Mass (m) represents the translational inertia while rotational inertia (I) is a function of geometry and mass distribution in an object. Using the Euler equation ($\sum M = I \cdot \ddot{\theta}$) the dynamic moments equilibrium around the crankshaft axis (global z-axis) is:

$$a \cdot \left(-F_{BE,X} \cdot \cos\theta + F_{BE,\mathrm{Y}} \cdot \sin\theta\right) + T_{CS} = I_{zz} \cdot \ddot{\theta}$$

Where θ is the periodic crankshaft angle of rotation in reference to the zero angle, a is the crankshaft excentricity, $F_{BE,\mathrm{X}}$ and $F_{BE,Y}$ are the conrod big end bearing forces (see Figure 1), T_{CS} is the crankshaft torque and I_{zz} is the crankshaft inertia around the z-axis (Morillo, Kurka, & Bittencourt, 2018). It can be concluded that a larger crankshaft inertia will make speed fluctuation ($\ddot{\theta}$) less significant.

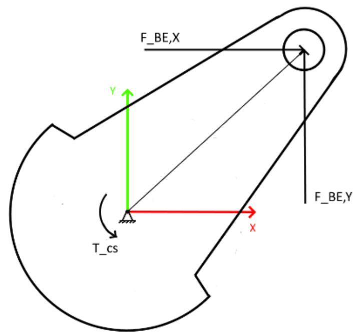

Figure 1: Static force and moments equilibrium of the crankshaft

1.2. Elastohydrodynamic bearings

Hydrodynamic lubrication means that the load-carrying surfaces are separated by a relatively thick (few micrometers) film of lubricant that prevents metal-to-metal contact. Such a film is described fully by the laws of fluid mechanics. Hydrodynamic bearings do not require the introduction of lubricant under pressure, still an adequate supply is required. The pressure necessary to fully separate the surfaces is created by the moving surface itself pulling the lubricant into a wedge-shaped zone at a enough speed. However, due to fluctuation in velocity of the moving surfaces, surface area or lack of lubricant, asperity contact may occur and elastohydrodynamic theory is required to fully describe bearing behavior in this state. In addition to fluid mechanics this mathematical model requires the Hertzian theory of contact stress (Budynas & Nisbett, 2011).

The modelling of hydrodynamic forces in hydrodynamic bearings is carried out by solving the Reynolds equation which links the hydrodynamic pressure and the fluid film thickness. However, this basic form of the Reynolds equation considers only the macro geometry of the bearing and neglects effects due to surface roughness. To model the asperity contact between surfaces, the present study utilizes the model developed by Greenwod & Tripp (1970).

2. NUMERICAL MODELS

2.1. GT-Suite 1D code

GT-Suite is a multiphysics platform, enabling simultaneous fluid flow and rigid multi-body dynamic analyses of reciprocating compressor cranktrains. In combination with a virtual PID-controller for the motor torque, the speed fluctuation minimum, maximum and amplitude values are obtained. EHD bearings are substituted with revolute joints for the benefit of faster solve times. Two models with different levels of accuracy are available: a “simplified” or a “full fluid model”.

In the “full fluid model” the complete fluid dynamics of the compressor are modelled, including valve behavior and suction and discharge mufflers. Different operating conditions can be specified, the main parameters being the condensing and evaporating temperatures, compressor speed and type of refrigerant. The output of this model used in the present study is the crankshaft angle resolved cylinder pressure ($p_{Cyl} = f(\theta)$). Consequently, the big end bearing forces $F_{BE,\mathrm{X}}$ and $F_{BE,Y}$ are defined as $F_{BE,\mathrm{X}} = f(\theta)$ and $F_{BE,Y} = f(\theta)$. The aim of the simplified fluid model is to provide a fast-solving model that could evaluate full operating condition envelopes within a reasonable amount of time. The main benefit of fast solve times comes with a restriction to conservative operating conditions, therefore the full fluid model must be used for evaluating the highest condensing temperature conditions (see 3.1).

Operating conditions in the present study are defined by three parameters: speed, evaporating temperature and condensing temperature. An envelope containing all possible combinations of usual values was evaluated to visualize parameter correlation and give a general understanding of the order of magnitude of speed fluctuation amplitudes.

Condensing Temperature [°C]								
65	0.4	0.36	0.48	0.52	0.61	0.71	0.83	1
55	0.24	0.29	0.33	0.39	0.45	0.52	0.61	0.71
45	0.2	0.24	0.28	0.33	0.39	0.45	0.52	0.6
35	0.16	0.2	0.24	0.28	0.33	0.39	0.45	0.51
25	0.13	0.16	0.2	0.24	0.28	0.33	0.38	0.43
	-30	-25	-20	-15	-10	-5	0	5

Evaporating Temperature [°C]

Figure 2: Speed fluctuation on the operating conditions envelope (normalized)

2.2. AVL Excite full model

AVL Excite is an industry-leading rigid and flexible multi-body dynamics simulation software. Utilizing the Finite Element Method (FEM) it offers strong numerical coupling between structure dynamics and elastohydrodynamic (EHD) bearings on multiple levels of detail. The existing compressor geometry is meshed using Ansys and preprocessed for use in Excite using mesh condensation methods (AVL, 2020). Features that define joints, representing the interaction between bodies, are the advantages of using this software for this analysis. The results that can be extracted include friction power losses and pressure distribution. Such numerical simulations are already used in the industry to calculate frictional losses, among other things, and show same parameter dependence as analytical investigations (Posch, et al., 2016) (Ozdemir, et al., 2014).

A reciprocating compressor used in this numerical model normally consists of six flexible bodies: crankshaft, crankcase, conrod, piston, piston pin and rotor. All bearings are modelled as full EHD bearings, meaning the model contains four radial bearings: lower bearing, upper bearing, conrod big end bearing and conrod small end bearing. Additional bearings in the model are the piston liner bearing (also EHD) and axial bearing, which is defined as a non-linear spring connection.

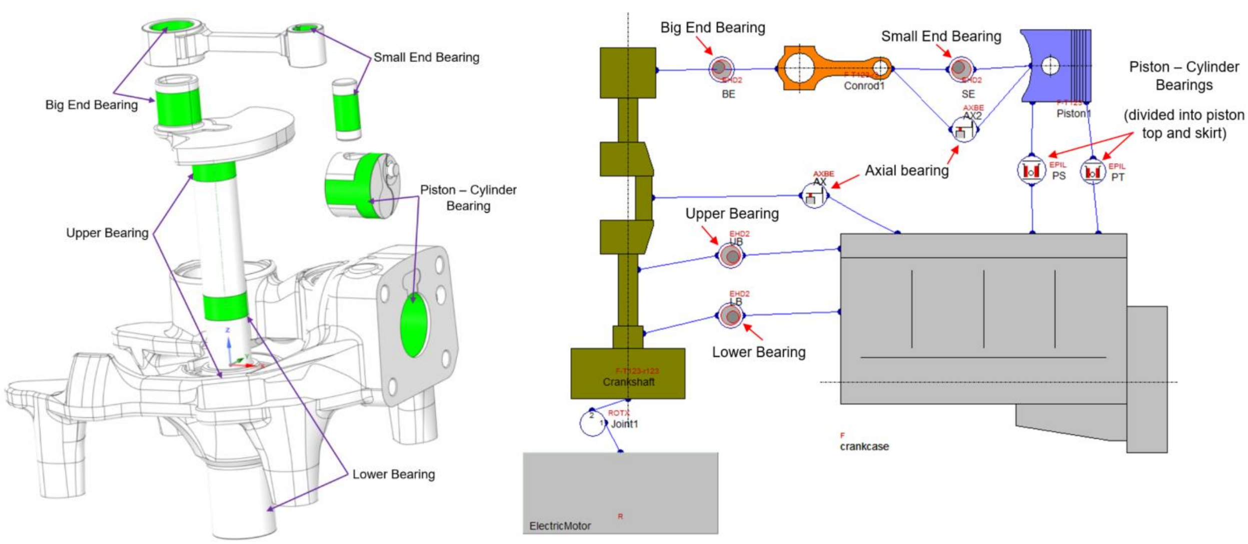

Figure 3: Bearing locations

2.3. AVL Excite reduced model

Evaluating full model simulations is very time consuming and computationally intensive. To reduce solve times and make the model usable for studies with numerous design points a reduced model with only one EHD bearing is used (Table 1). The lower bearing is substituted with an idealized revolute joint, the piston-cylinder contact with an axial constraint. The decision which bearings or contacts can be idealized and to what degree is based on preliminary full model results (see 3.1).

Table 1. Full model vs. reduced model comparison

Contact	Full model	Reduced model
Lower bearing	EHD	Revolute joint
Upper bearing	EHD	EHD
Conrod big end	EHD	-
Conrod small end	EHD	-
Piston liner	EHD/EPIL	-
Axial bearing	NONL	NONL

2.4. Sensitivity Analysis

A sensitivity analysis in conjunction with a screening study is performed to determine relevant factors and their correlation with the peak asperity pressure and minimum oil film thickness. Each parameter is varied within a specified range and its effect on the defined output parameters is noted. To correctly determine relevance and ensure equal weighing, each parameter is varied by +20%/-20% in relation to its base value (see Table 2). The individual design points are created using a 2-level full-factorial design of experiments (DoE) configuration. The chosen parameters already showed significant influences on bearing design in other similar studies (Meier, et al., 2018) (Ozdemir, et al., 2014).

Table 2. Parameter lower and upper bounds

Parameter	Lower bound	Upper bound
Oil temperature	-20%	+20%
Bearing clearance	-20%	+20%
Bearing width	-20%	+20%
Compressor speed	2300 rpm	3450 rpm

Additionally, rotor inertia was varied separately due to the restrictions of the model. Note, that in Excite the crankshaft and rotor are represented by one body, therefore changing the rotor inertia by 20% will not change the entire crankshaft assembly inertia by 20%.

Table 3: Crankshaft inertia design points

Crankshaft inertia	original	+20% rotor inertia	+40% rotor inertia

The following output parameters were defined: minimum oil film thickness and peak asperity pressure.

3. RESULTS

3.1. Effect of speed fluctuation

Full film lubrication requires enough speed to generate the required pressure that fully separate two surfaces. Due to speed fluctuation within an individual compressor cycle this speed is sometimes not reached and asperity contact occurs. Preliminary studies for this generic compressor have determined that this occurs more often in the upper bearing and has the largest effect on total bearing friction power losses (see Figure 4). Additionally, the "reliability" operating condition at 2300 rpm is chosen as the focus point of the present study because it is the most critical in terms of asperity contact. It is also the condition at which reliability tests are carried out. The following figure shows these differences in different operating states. Furthermore, Figure 4 shows that asperity contact pressure changes inversely with compressor speed as no asperity contact occurs at 4500 rpm or even 3000 rpm.

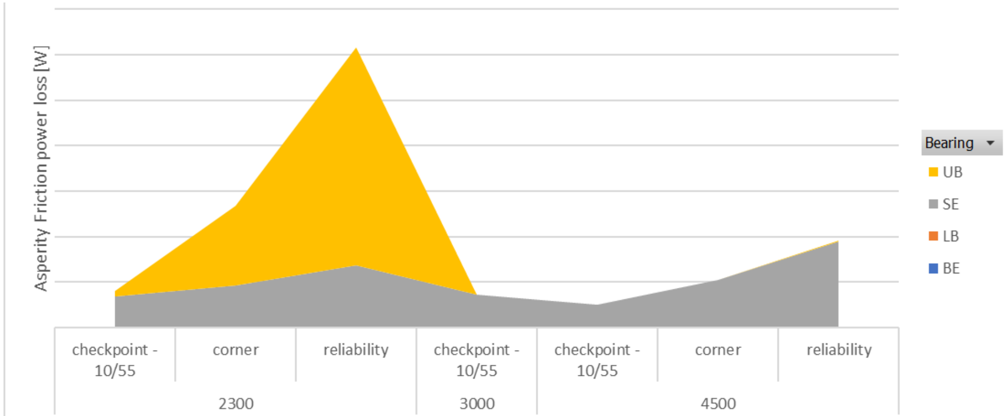

Figure 4: Asperity friction power loss at different operating conditions

Speed fluctuation can be further reduced by increasing the rotor inertia (see Figure 5), where a flattening of the curve can be observed by increasing the inertia as per Table 3. Average curves were added to validate the fact that the average compressor speed does not change.

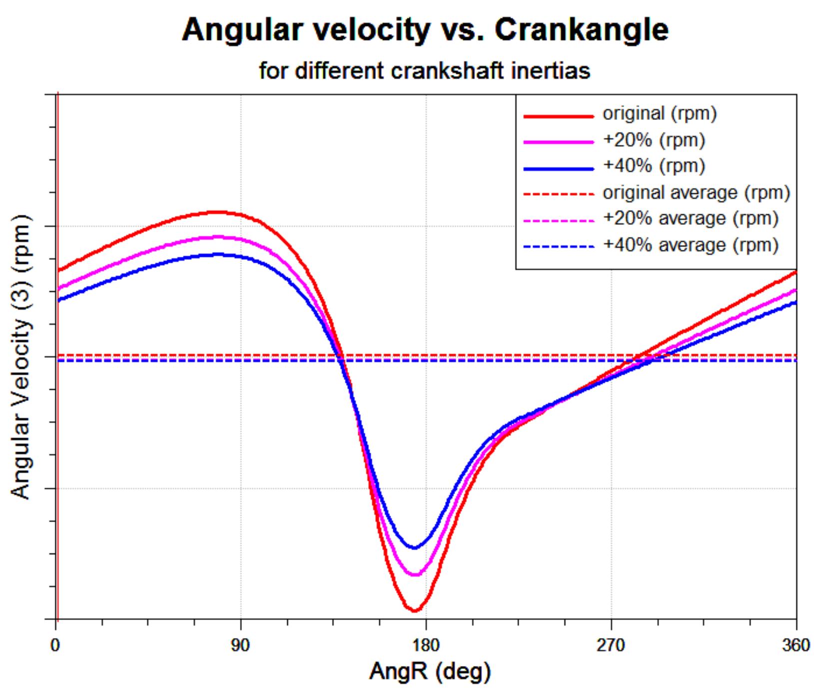

Figure 5: Speed fluctuation within one cycle

Figure 6 represents the effect of fluctuating and constant speed on pressure in the bearing. Both cases show similar shapes and similar location of peak asperity contact pressure (see Figure 7). In both cases the peak appears at 182° crankshaft angle. On the other side, results show that when the bearing was affected by the fluctuating speed, the asperity contact pressure increased, causing the total pressure to increase with it. 2D result plots of Figure 7 show the exact locations where asperity contact occurs (on top edge of the upper bearing shell).

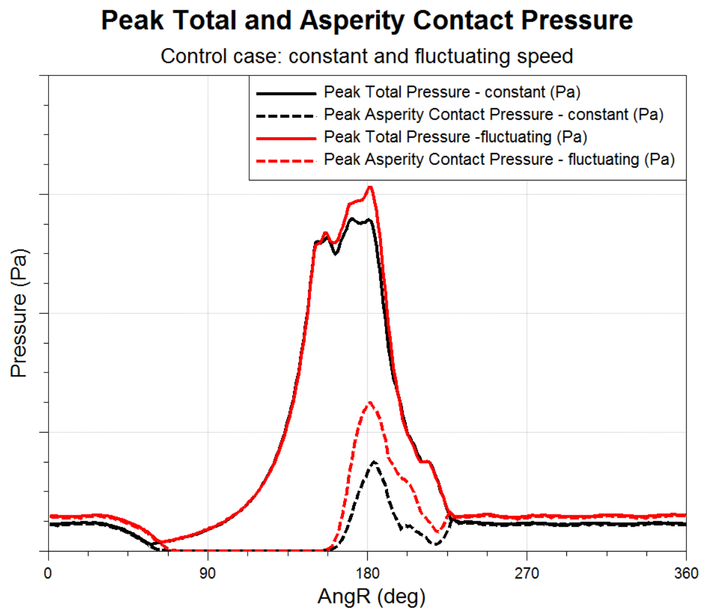

Figure 6: Peak total pressure vs crank angle for reference case

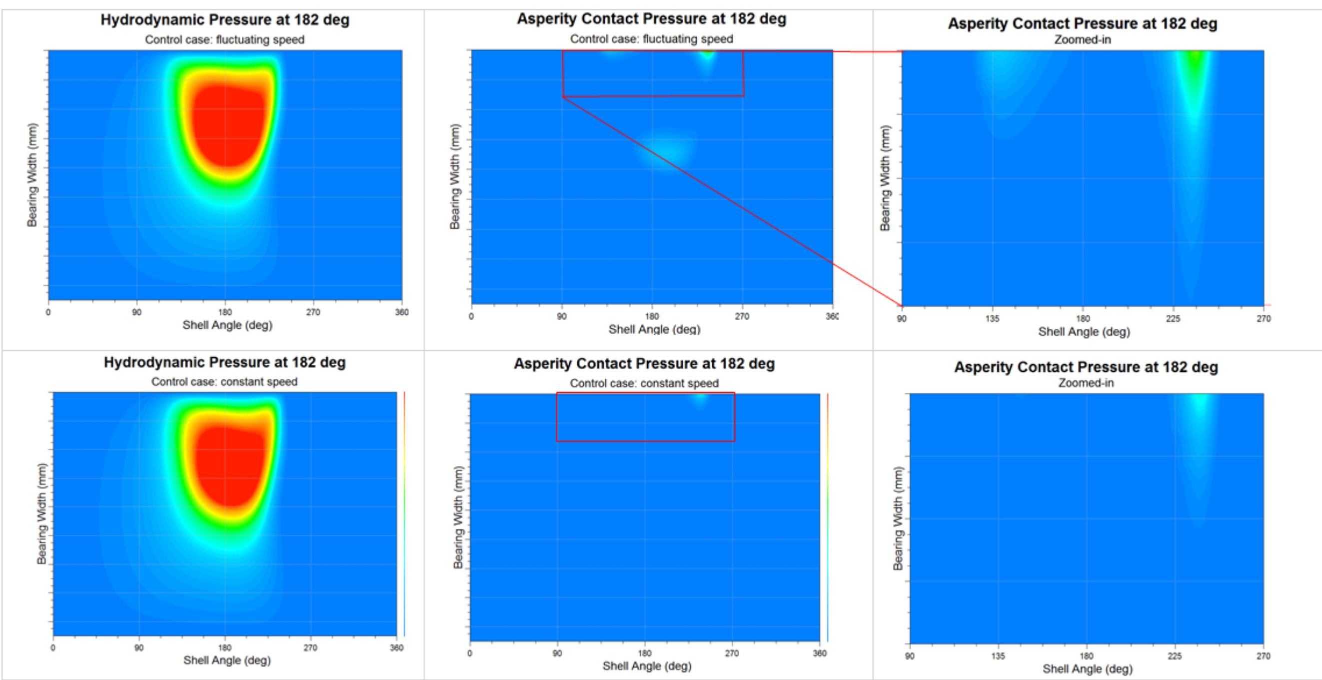

Figure 7: Hydrodynamic pressure (left), Asperity contact pressure (centre) and zoomed-in to upper edge asperity contact pressure (right) distributions on the upper bearing shell

3.2. Sensitivity analysis results

Results were gathered from a 2-level full factorial DoE study, where each parameter was changed by +20%/-20% in relation to its base value. The parameters of interest are oil temperature, clearance and width of the upper bearing and compressor speed. The following figure shows collected results for the asperity contact pressure. Similar tendencies are observed regarding to minimum oil film thickness.

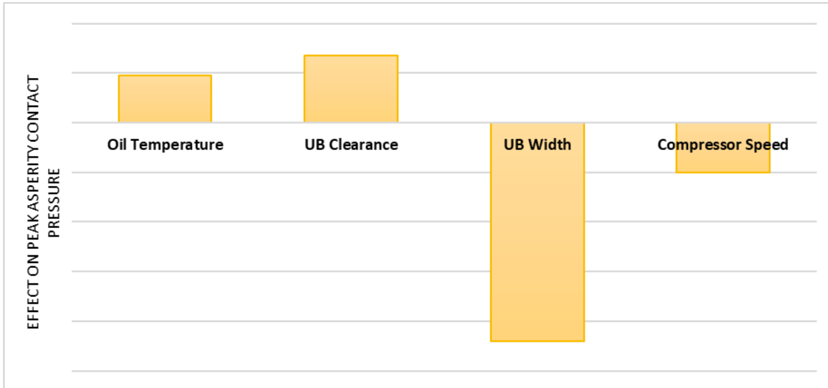

Figure 8: Sensitivity analysis factor correlation

The results (visible on Figures 8, 9 and 10) show that oil temperature and compressor speed have the lowest influence on asperity contact pressure and oil film thickness. They also show that oil temperature has a direct and compressor speed an inverse effect on asperity contact pressure. Slightly higher is the influence of clearance in the EHD bearing (direct relationship), which was also identified as an influencing factor by another publication. In it, Meier et al. (2018) showed that by changing the clearances (decreasing them) it is possible to minimize friction power loss from asperity contact and on the other side transfer these losses to viscous (hydrodynamic) contact. The DoE results in this paper showed that the highest effect (inverse relationship) comes from the EHD bearing width. Resulting influences are seen in the effect that a decrease of bearing width (length) leads to reduction of hydrodynamic friction power loss and an increase of asperity friction power loss, also shown by Meier et al. (2018).

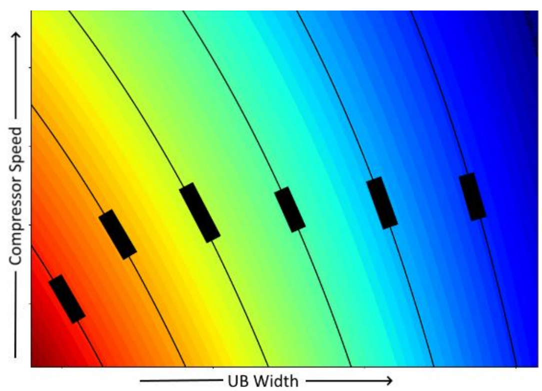

Figure 9: Peak asperity contact pressure

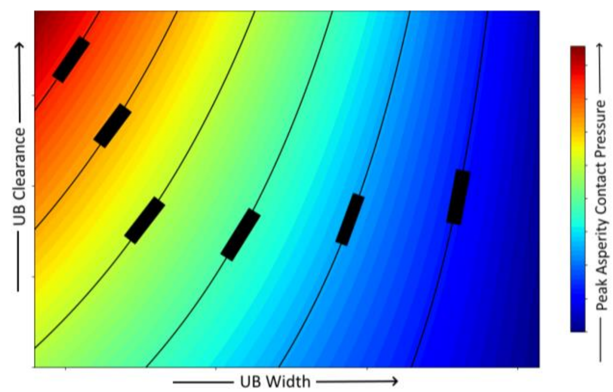

Figure 10: Peak asperity contact pressure

4. CONCLUSIONS

This paper presents an investigation about speed fluctuation on bearings design in a compressor and looks to provide an answer whether and how speed fluctuation affects hydrodynamic and asperity contact pressures in sliding bearings. To investigate the worst-case scenario, first the most affected bearing by the asperity pressure in the compressor was identified (Upper Bearing). Alongside, the most critical operating condition was used for the numerical model (Reliability Condition at 2300 rpm), which also showed the highest friction losses due to asperity contact. Second, the scope of sensitivity analysis and its variables were defined (lower and upper bounds). Third, the parameters were incorporated into previously mentioned software and results show the following conclusions:

1. Increasing the inertia of the rotor in the compressor can lead to reduction of speed fluctuation.

2. A bearing with fluctuating angular velocity produces higher asperity pressure if compared with constant speed.

3. The width of the bearing contributes mostly to the asperity contact pressure and minimum oil film thickness (in an inverse relationship). While other parameters (oil temperature, clearance and compressor speed) contribute approximately equally, effect of bearing width had a four times bigger influence.

Future research on this topic could also cover other parameters and investigate their effects on total and asperity pressure and minimum oil film thickness. Such parameters could be roughness of surfaces of journal and shell in EHD bearings, shape of the bearing geometry or material properties (eg. Young´s modulus). The high sensitivity of bearing friction losses to surface roughness was already discussed in another publication (Meier, et al., 2018). Interesting results could also be gained from performing the same study on bearings in the piston-liner connection of the compressor. Additional option to improve upon this paper is the sensitivity analysis configuration. The factors were all varied by the same amount (+20%/-20%), in future research, this variation could be adapted for each factor individually to better reflect reality.

This paper adds to the already existing knowledge pool on the topic of EHD bearings simulation with respect to the negative but realistic effect of speed fluctuation on bearing reliability (through asperity contact pressure). The conclusions of the analysis serve as help when designing better, more reliable bearings in different compressors.

NOMENCLATURE

p	Pressure (kPa)	M	Moment (Nm)
F	Force (N)	T	Torque (Nm)
a	Crankshaft throw (mm)	θ	Crankshaft angular position (deg)

Abbreviations (in text or subscript)

UB	Upper bearing	*BE*	Big end
SE	Small end	*Cyl*	Cylinder
LB	Lower bearing	*EPIL*	Piston-liner contact joint
NONL	Non-linear	*DoE*	Design of experiments
EHD	Elastohydrodinamic	*FEM*	Finite element method
rpm	Revolutions per minute		

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