Comprehensive 3D-elastohydrodynamic simulation of hermetic compressor crank drive

S Posch1, J Hopfgartner1, E Berger1, B Zuber1, R Almbauer1 and P Schöllauf2

1 Graz University of Technology, 8010 Graz, Austria
2 Secop Austria GmbH, 8280 Fürstenfeld, Austria

posch@ivt.tugraz.at

Abstract: Mechanical, electrical and thermodynamic losses form the major loss mechanisms of hermetic compressors for refrigeration application. The present work deals with the investigation of the mechanical losses of a hermetic compressor crank drive. Focus is on 3d-elastohydrodynamic (EHD) modelling of the journal bearings, piston-liner contact and piston secondary motion in combination with multi-body and structural dynamics of the crank drive elements. A detailed description of the model development within the commercial software AVL EXCITE Power Unit is given in the work. The model is used to create a comprehensive analysis of the mechanical losses of a hermetic compressor. Further on, a parametric study concerning oil viscosity and compressor speed is carried out which shows the possibilities of the usage of the model in the development process of hermetic compressors for refrigeration application. Additionally, the usage of the results in an overall thermal network for the determination of the thermal compressor behaviour is discussed.

Nomenclature

| Roman symbols | Greek symbols |
|---------------|---------------|
| h             | \( \mu \)     | fluid film thickness | dynamic viscosity |
| p             | \( \phi_x, \phi_y, \phi_S \) | pressure | flow factors |
| t             | \( \rho \)     | time | density |
| u             | \( \sigma_d \) | velocity | standard deviation of the combined roughness |
| x, y          |               | coordinate |                |

1. Introduction

The compressor is the core element of the cooling circuit of a refrigeration appliance. In addition to the improvement of the entire refrigeration cycle in terms of energy efficiency to meet the energy standards, the further development of the single components like the compressor is driven forward. The determination of the loss mechanisms namely electrical, mechanical and thermodynamic is indispensable in the development of innovative efficient hermetic reciprocating compressors. Due to the hermetic design of the compressors, the different loss mechanisms interact with each other and thus a holistic consideration of the overall thermal compressor behaviour is necessary. Several studies presented thermal compressor models trying to use as much physical information as possible. In [1] and [2], the authors combined 3d heat conduction formulation for solid parts and lumped formulation of the gas path with experimental data or correlations from the literature to model the heat transfer coef-
coefficients. A more detailed method is used in [3]. Additionally to the 3d heat conduction formulation of the solid parts the refrigerant gas path was simulated 3d using commercial CFD software. Mechanical and electrical losses were considered via heat sources in the particular part or in the lubrication oil. The determination of the mechanical losses in the compressor, mainly the journal bearing losses and the piston losses, has been published in several studies. A majority of the studies uses the similar general simulation strategy embedded in self coded simulation codes. The simulation strategy can be summarized in the following steps: (i) determination of the dynamic loads by applying multi body simulation, (ii) calculation of the hydrodynamic forces solving the Reynolds equation with numerical methods, (iii) calculation of displacements using a Newton-Raphson algorithm and (iv) calculation of the friction losses based on the displacements. A short overview of some studies and their statements concerning the modelling of the Reynolds equation and the model accuracy should be given at this point: [4] applied three different approaches to simulate the motion of the crank shaft and furthermore the journal bearing forces, whereby one approach takes into account lateral and tilting oscillations of the crank shaft. The modelling of the journal bearing forces was carried out applying the so-called short bearing approximation. The authors stated that no significant difference in the estimation of the journal bearing forces and the fluid film pressure between the different cases was found. A comparison between short and finite bearing approach can be found in [5]. As the authors concluded, the short bearing approach can be used as a first approximation to predict the bearing performance. In order to predict good results for piston cylinder contact, the finite bearing approach should be used. A more detailed calculation of the journal bearings of a hermetic compressor by using the averaged Reynolds equation introduced in [6] and [7] was presented in [8]. By applying the finite bearing approximation, the authors determined the friction power losses for the two journal bearings and investigated the influence of the oil viscosity and the gap width in the bearing. The method was verified by simple analytical friction loss calculations based on Couette shear stress.

The present study deals with the detailed calculation of the friction losses in a hermetic reciprocating compressor using the commercial software package AVL EXCITE Power Unit. An example of an application of the software can be found in [9]. The study analyses the piston secondary movement and its impact on the structure-borne noise behaviour of a direct injection Diesel engine. Although the software is usually used in engine development, the present study should show the ability of the software for the usage in reciprocating compressor investigations. The software is able to simulate the friction forces taking into account the elastic deformation of the crank drive parts. A general workflow for the use of AVL EXCITE Power Unit in the simulation of the mechanical losses of a hermetic reciprocating compressor is given and the model is utilised to investigate the influence of compressor speed and oil viscosity.

The compressor used in the present study works with R600a (isobutane) and contains a displacement of 5.5 cm³. Its COP at ASHRAE test conditions (-23.3 °C/55 °C) is approximately 1.8.

2. Simulation

The software AVL EXCITE Power Unit is an advanced tool for the simulation of rigid and flexible multi-body dynamics for the application in combustion engine and drivetrain development. Due to the similar structure of reciprocating compressors and combustion engines, the usage of the software for the calculation of mechanical losses in refrigeration compressors is useful. An overview of the workflow for setting up the simulation of the compressor crank drive with AVL EXCITE is given in this chapter. Furthermore, the pre-processing steps concerning condensation of the compressor parts and the theoretical background of the hydrodynamic force modelling are given.

2.1 Layout

An illustration of the simulation layout within the used software can be seen in Figure 1. The model is built up with subsystems representing several compressor parts connected with relevant types of joints. In the present study, a mechanical loss calculation of the overall compressor crank drive is carried out. Therefore, all crank drive parts and the crankcase are considered. Additionally, detailed modelling of
the friction behaviour in the journal bearings of the crank shaft, connecting rod big end and the piston-liner with elastohydrodynamic (EHD) contacts is carried out. Joints with little influence on the overall mechanical losses like the piston-piston pin and the connecting rod-piston pin connection are modelled with simple revolution joints by setting linear or non-linear spring/damper functions [10].

Geometric input data of the crank drive enable the calculation of the compressor dynamics via multi-body simulation.

2.2 Pre-processing
EHD simulation requires detailed information about the elastic behaviour of the considered parts. In the pre-processing phase of the simulation, the body input data for the use in AVL EXCITE Power Unit is generated. Starting from the 3d-CAD model, a finite element model with defined degrees of freedom is generated in the commercial software SIMULIA Abaqus. A predefined interface between the two software packages ensures the implementation of the input data in correct syntax. Figure 2 shows exemplarily the condensation process of the connecting rod. At the big end of the connecting rod seven node points with defined degrees of freedom are created which have to be aligned with the appropriate nodes on the crank shaft. Due to the simple modelling of the connecting rod small end, the part is condensed to only one node with defined degrees of freedom.

2.3 Simulation handling
After the several compressor parts are loaded into the software and are linked by appropriate contacts, the compressor load must be defined. In the present study, a pressure profile gained by CFD simulation is used. The CFD simulation was carried out in the commercial software ANSYS Fluent at the previous mentioned ASHRAE test conditions modelling the valve motions by the use of ideal moving
flat plates. Assuming equal pressure profile for every crankshaft revolution, the pressure is set to be periodic. Depending on the surface contact model, the roughness values of the several bearing partners must be specified and the material properties are necessary. Since the software AVL EXCITE Power Unit is mainly used for the simulation of combustion engines, the oil database does not contain the used compressor oil, therefore, the oil properties must be specified by the user. Gravitational forces are neglected to avoid the implementation of axial bearings.

2.4 Lubrication theory
The calculation of the hydrodynamic forces is carried out by solving the average Reynolds equation introduced in [6] and [7] which links the hydrodynamic pressure to the fluid film thickness.

\[
\frac{\partial}{\partial x} \left( \phi_x \frac{\rho h^3}{12\mu} \frac{\partial p_h}{\partial x} \right) + \frac{\partial}{\partial y} \left( \phi_y \frac{\rho h^3}{12\mu} \frac{\partial p_h}{\partial y} \right) = \frac{(u_I + u_II)}{2} \left( \frac{\partial (\rho h_T)}{\partial x} + \frac{\partial p_h}{\partial x} \right) \frac{\partial \phi_s}{\partial x} + \frac{\partial (\rho h_T)}{\partial t} \tag{1}
\]

The terms in the average Reynolds equation can be interpreted as follows: the term on the left-hand side represents the Poiseuille flow, the terms on the right-hand side represent the Couette flow and the time dependent displacement flow, respectively. Compared to the original Reynolds equation, several flow factors which depend on the roughness properties of the surfaces are introduced. Additionally to the hydrodynamic contact, an elastic-plastic asperity contact can occur which is considered with the model presented in [11]. The oil supply of the regarded contacts is not modelled specially, therefore, the contacts are assumed to be filled with oil. To fulfil the continuity balance, a cavitation approach similar to the model in [12] is used in the software.

3. Results
Before presenting the results of the comparison in terms of friction power loss, results of the predefined standard case are shown. The standard case is defined as the previous described compressor working in steady-state mode for R600a at a suction pressure of 0.629 bar and a discharge pressure of 6.04 bar corresponding to -23.3 °C evaporating temperature and 45 °C condensing temperature, respectively. Compressor speed is set to 3000 rpm and the oil viscosity is assumed to be constant at 8 cSt. The usage of the commercial software AVL EXCITE Power Unit enables easy post-processing. A visualisation of the hydraulic pressure in the regarded EHD contacts is shown in Figure 3. It can be seen that the region in the piston-liner contact shows two peaks in the hydrodynamic pressure field, indicating increased wear in this region.
Figure 3. Visualisation of the hydraulic pressure in the EHD contacts.

The simulation of the standard case is carried out until steady-state conditions between two consecutive time steps are reached. For the standard case five crankshaft revolutions are needed to reach steady-state conditions yielding in a computational time of 20 hours on a quad core 3.50 MHz processor.

Figure 4 and Figure 5 illustrate results in terms of the reaction forces in the particular EHD contacts for one crankshaft revolution at steady-state conditions. The force curves reflect the crank mechanism behaviour and show the high forces at the connecting rod big end joint. Furthermore, the reaction forces at the piston liner contact and at the connecting rod big end joint have approximately sinusoidal character. In contrast, the amplitude reaction forces at the journal bearings of the crankshaft are smaller compared to the connecting rod big end and the curves do not show sinusoidal character.

Figure 4. Reaction forces in the crankshaft journal bearings.
Figure 5. Reaction forces in the connecting rod big end and the piston liner contact.

Simulation results of the body kinematics can be seen in Figure 6 and Figure 7. The combination of piston pin offset and partly one-sided piston side load yields in tilting angle curve according Figure 6. As the orbital paths of the crankshaft journal bearings show, the displacement of the crankshaft at journal bearing 2 is smaller compared to bearing 1 due to the lower force amplitude acting on it. Furthermore, the higher loads according to Figure 4 at bearing 1 lead to a not perfectly cyclical orbital path. The lower force amplitude also yields in an almost circular displacement curve of journal bearing 2 and furthermore, to a faster achievement of the final shape of the orbital path compared to bearing 1.

Figure 6. Piston tilting angle.

Figure 7. Orbital path of the crankshaft journal bearings.

The investigation of the friction loss behaviour of the considered joints is carried out by varying the oil viscosity and the rotational speed of the crankshaft. Figure 8 and Figure 9 illustrate the calculated friction power loss according to their occurrence. It is assumed, that the cylinder pressure gained by CFD results can be obtained for every variant. The viscosity variation (at constant speed of 3000 rpm) shows almost linear dependency of the friction power loss in the several joints and thus the overall friction loss.
Similar results are obtained for the variation of the compressor speed (at constant viscosity of 8 cSt). The comparison in Figure 9 shows an almost linear dependence between compressor speed and friction power loss. Another aspect which can be derived from the simulation results is the big influence of the reaction forces on the friction loss compared to the relative speed in the regarded bearing. By comparing the results of the connecting rod big end and the crankshaft journal bearings, the significant influence of the reaction forces can be seen. The share of the power loss in the connecting rod big end joint on the overall power loss is higher compared to the individual crankshaft journal bearings although the relative speed between the bearing gap walls in the connecting rod big end is smaller compared to the relative speed in the crankshaft journal bearings.

Additionally to the investigation of the compressor crank mechanism dynamics and kinematics, the obtained results can be used in thermal modelling of the compressor. The results can serve as input data for modelling approaches like presented in [12]. By taking into account the produced heat due to
friction in the individual joint and considering the heating of the oil, the overall accuracy of such models can be increased.

4. Conclusion
The presented paper deals with the investigation of the friction behaviour of a reciprocating hermetic compressor for refrigeration application. For this purpose a 3d-elastohydrodynamic simulation using the commercial software AVL EXCITE Power Unit was carried out. Focus was laid on the simulation of the connections where the highest friction losses occur which are the two crankshaft journal bearings, the connecting rod big end and the piston liner contact. The simulation model was used to determine the dynamics and kinematic behaviour of the compressor crank mechanism. Furthermore, the influence of the oil viscosity and the compressor speed on the friction power loss was investigated. Based on a predefined standard configuration (3000 rpm, 5 cSt) the friction power loss according to their occurrence and the overall friction power loss of the several configurations were compared. The results show an approximately linear dependency of rotational speed and oil viscosity on the friction loss. It can be pointed out that the connecting rod big end joint produces higher friction losses compared to the other connections due to the high reaction forces although the relative speed between the journal walls is quite small.

The gained results can be used to estimate the usage possibility of the regarded crank mechanism design in a variable speed compressor. According to the presented results, the friction power loss at lower compressor speeds will decrease but this effect can be cancelled out due to the lower temperature in the compressor at lower speeds yielding in higher oil viscosities and therefore in increasing friction losses. This circumstance shows a trade-off in the design of variable speed compressors. The choice of the lubrication oil usually depends on the compressor modes in which the overall compressor temperature is high like in the refrigerator pull-down mode. By linking simulation models of the friction behaviour and thermal compressor models, a holistic approach of modelling the oil influence on the compressor performance can be achieved.

Acknowledgements
This work has been carried out within the framework of ECO-COOL, a research project initiated and funded by the FFG (Austrian Research Promotion Agency). Furthermore the authors particularly acknowledge the technical support by Secop Austria GmbH, formerly ACC Austria GmbH and Liebherr-Hausgeräte Lienz GmbH.

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