

An Investigation on the Pin Bearings' Optimization of a Hermetic Reciprocating Compressor

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Abstract. A hermetic reciprocating compressor is the most energy consuming component of the refrigerators. Therefore performance improvement studies of the compressor play an important role to reduce overall energy consumption of the refrigerators. Design of bearings is one of the major study areas influencing reciprocating compressor performance.

In this study crank pin and piston pin bearings in hermetic compressor applications are investigated and optimized. The effect of operating conditions, bearing offset between crank pin and piston pin, bearing clearance and bearing location along the shaft on the friction loss were investigated and optimal bearing designs were developed.

Efficiency measurements of the compressor showed that the improved crank pin and piston pin bearing designs provide up to 3.8% increase in the COP as a result of reduction in mechanical loss that is significantly influenced by the length of the bearings, bearing location along the shaft and operating conditions.

Keywords: Reciprocating compressor, Bearing, Mechanical Loss, Friction, Coefficient of performance (COP)

1. Introduction

Positive displacement piston type hermetic compressors play an important role in the energy efficiency of domestic refrigerators. Therefore the tightening energy consumption regulations for domestic refrigerators push the compressor manufacturers to design and produce more and more energy efficient hence environmentally friendly compressors. The most important development objectives for hermetically sealed compressors are increasing the Coefficient of Performance (COP); decreasing manufacturing and operating costs; lengthening operating life, reducing noise and vibration levels. Design of the hermetic compressor bearings is directly related with all these topics.

The performance of a reciprocating compressor is influenced by the mechanical loss. In hermetic reciprocating compressors, mechanical loss is the frictional loss at bearings, which are used to support moving and rotating parts. The main factors affecting the mechanical loss are bearing dimensions, type of bearings, lubrication characteristics, friction coefficient, loading conditions, surface roughness, operating temperature, material, bearing surface coating, relative bearing surface velocity, clearance between journal and bearing housing.

In this study crank pin and piston pin bearings in hermetic compressor applications are investigated and optimized. The effect of operating conditions, bearing offset between crank pin and piston pin, bearing clearance and bearing location along the shaft on the friction losses were investigated and the optimized bearing designs were developed. Detailed parametric numerical simulations were performed using commercial software. According to the simulation results performance measurements were carried out for the selected design parameters. The results of the numerical analysis have shown that



the numerically calculated mechanical loss level is in correlation with the performance results measured in a calorimeter test system.

Bearing design influences the efficiency of reciprocating compressors. Appropriate bearing design, that provides sufficient oil film pressure/thickness while minimizing friction loss, increases COP level of the reciprocating compressor.

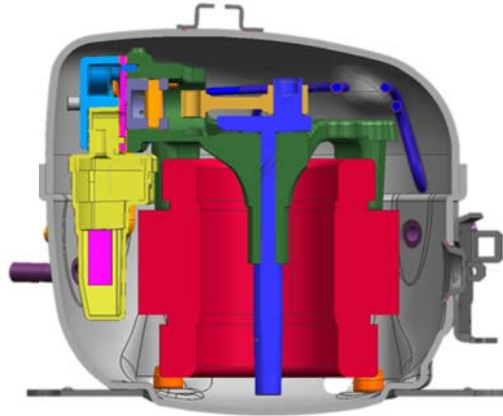


Figure 1. 3D CAD model of the studied reciprocating compressor

Numerical simulations with an engineering software coupled with experimental studies have become an essential way for improvement the performance of reciprocating compressors. A lot of research dealing with the modeling and simulating the compressors and their components are present in the literature. There are also some remarkable researches done in order to develop optimum compressor bearing designs.

M. Duyar [1] numerically solves the elastrohydrodynamic lubrication of a piston pin bearing of a reciprocating compressor. B. Hacioglu [2] presented a model that solves Reynolds' equation for piston cylinder sliding bearing for a compressor piston with oil feed groove. P.Mantri et al [3] modeled the piston-cylinder interaction inside a small hermetic compressor using the Reynolds equation which is solved using finite difference method. A.R. Ozdemir et al [4] investigated and optimized the crank shaft bearings in hermetic compressor applications. In this study the effect of crank shaft geometry, bearing clearance, lubricant viscosity and surface roughness on the friction losses were investigated and the new journal bearing designs were developed.

This study used the results of A.R. Ozdemir et al [4]'s main bearing designs and focused on the optimization of the other bearings (Crank Pin Bearing and piston pin bearing).

2. Experimental Studies

Calorimeter measurements and pV (pressure volume diagram) test results were the main experimental studies for determining the necessary boundary conditions for numerical calculations. Isobutene (R600a) refrigerant applied during the tests in the compressor calorimeter system. Measurement uncertainties of the calorimeter for different parameters are $< 2\%$. In pV set up piezo-resistive miniature pressure transducers were used to measure the pressure inside the cylinder for bearing force investigations. Reaction forces on crankshaft bearings were created by the gas force on piston. These reaction forces must be carried at the bearings by the oil film pressure generated. An optical encoder was placed on the shaft, for the determination of the cylinder volume and the piston position. pV measurements were conducted at ASHRAE test conditions to examine the pressure characteristics of the investigated compressor. The results of pV measurements of the reciprocating compressor are shown in Figure 2.

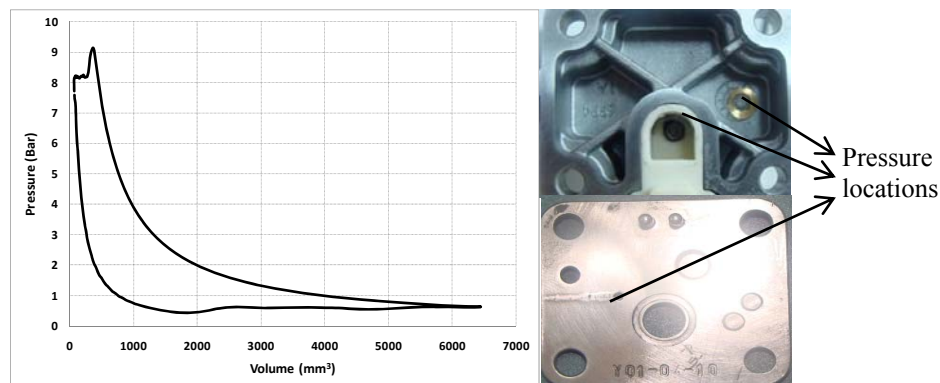


Figure 2. pV-diagram and temperature/pressure measurement locations

3. Numerical Analysis

The commercial bearing design engineering software, which provides different solution techniques for analyzing the dynamics of the bearings, used in this study. Determined conditions have been taken into account in the bearing analysis.

- Cylinder pressure variation with respect to the crank angle obtained from the ASHRAE conditioned pV measurements is used.
- Kinematic viscosity variation with respect to the temperature variation of the lubricant is calculated from Walther-ASTM equation.
- Boundary contact friction coefficient is selected as 0.08

3.1. Computational Model

Selected compressor bearings (crankshaft, piston, connecting rod and piston pin) are rigidly modeled. Calculation domain is divided into a number of cells by meshing process. In the computational model circular and axial mesh nodes were used according to the length (L.) of the bearing. Analyses properties and boundary conditions were given in Table 1. And the model of computational domain simulating the crankshaft and piston pin is shown at Figure 3.

Table 1. Analyses properties and boundary conditions

<u>Upper Main Bearing</u> Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes
1,1	27	73
<u>Lower Main Bearing</u> Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes
0,6	15	73
<u>Crank Pin Bearing</u> Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes
0,9	23	73
<u>Piston Pin Bearing</u> Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes
0,92	31	73
0,83	28	73
0,75	26	73
0,67	23	73

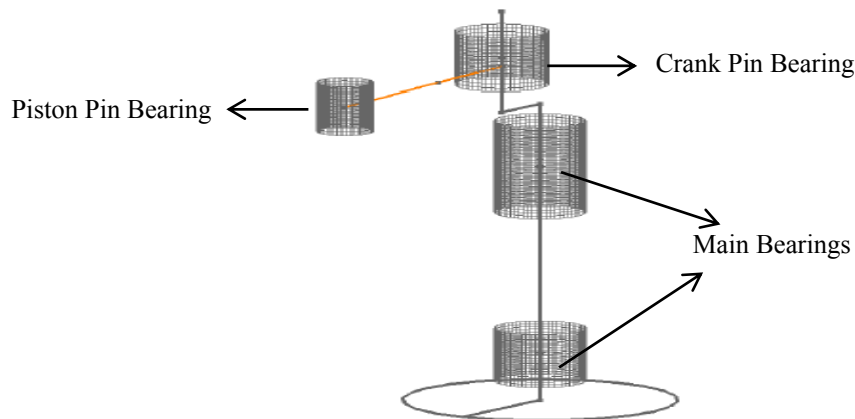


Figure 3. Model of bearings and computational domain

3.2. Parameters and Operating Conditions

Operating conditions and parameters for numerical analysis of bearings and the dynamic behavior analysis of the reciprocating compressor are given in Table 2. Finite difference discretization method which uses the Half Sommerfeld boundary condition to predict oil pressure for the given boundary conditions was used during the analysis. The ASHRAE conditions were used as the compressor test conditions (54.4 °C condensing temperature, -23.3 °C evaporating temperature).

Table 2. Properties and operating conditions

Parameters and Operating Conditions	
Piston Pin Bearing/Shaft Diameter Ratio	0.8
Length of conrod/Shaft Diameter Ratio	3.2
Cylinder bore diameter/Shaft Diameter Ratio	1.8
Mass of conrod (gr)	19
Mass of piston pin (gr)	5.8
Mass of piston (gr)	18
Oil temperature (°C)	80
Kinematic Viscosity (at 40°C)	5 cSt
Nominal oil pressure (bar)	0.624
Working condition	ASHRAE
Maximum rated speed (rev/min)	2930

3.3. Solution Method

Cylinder pressure variation for one cycle of the compressor was used for kinematic analysis.

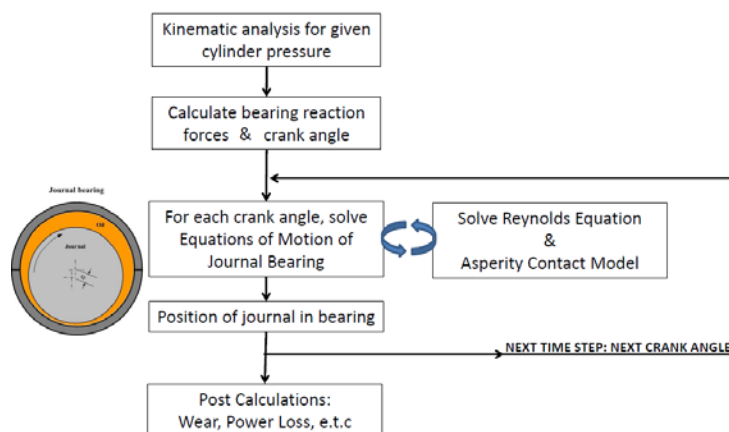


Figure 4. Flow chart of solution method

Solution methodology starts with the calculated bearing reaction forces for every crank angle. Equations of motion of journal bearings were solved according to the governing equations in which oil film pressure was calculated using Reynolds equation and asperity contact model for every crank angle in the second step of the program. Finally maximum film pressure, minimum film thickness, hydrodynamic power loss and boundary lubrication power loss were calculated.

4. Results and Discussions

Numerical calculations were realized with 4 cycle of computation for obtaining the cyclic convergence for the all cases. Results of numerical and experimental studies are summarized in this section.

4.1. Numerical Results:

144 different combinations of the parameters given in Table 3 were used for hydrodynamic bearing analysis in the numerical study. The load acting on the bearings was calculated under the ASHRAE operating conditions. The bearing analyses were realized for certain assumptions and results were compared with the baseline case of the model.

Table 3. Numerical Analysis Cases

Parameters	
Crank Pin B. Clearance/Shaft Diameter Ratio	1.9E-03/1.6E-03/ 1.3E-03
Piston Pin B. Clearance /Shaft Diameter Ratio	1.2E-03/4.2E-04/6.7E-04/9.2E-04
Piston Pin Bearing/Shaft Diameter Ratio	0.92/0.83/0.75/0.67
Bearing offset/Shaft Diameter Ratio	1.4E-01/ 8.3E-02/0
Upper Main Bearing/Shaft Diameter Ratio	1.1
Lower Main Bearing/Shaft Diameter Ratio	0.6
Crank Pin Bearing/Shaft Diameter Ratio	0.9
Kinematic Viscosity (at 40°C)	5 cSt
Working condition	ASHRAE

Maximum film pressure, minimum film thickness and total power loss which include the sum of hydrodynamic power loss and boundary lubrication power loss were calculated for every case. Total power loss ratio variation with respect to the defined cases is given in Figure 4. Maximum film pressure, minimum film thickness and power loss ratio results for crank pin and piston pin bearings are given in Figure 5. and Figure 6. respectively.

According to the evaluation of the numerical results in defined analysis cases range; decreasing the length of the bearings reduces the total power loss, on the other hand minimizing the clearance between the crankshaft journal and its bearing increases the total power loss. The optimization of the bearing dimensions (diameter/length/clearance) is critical to reduce hydrodynamic, mixed lubrication and boundary contact power losses and wear in the compressor.

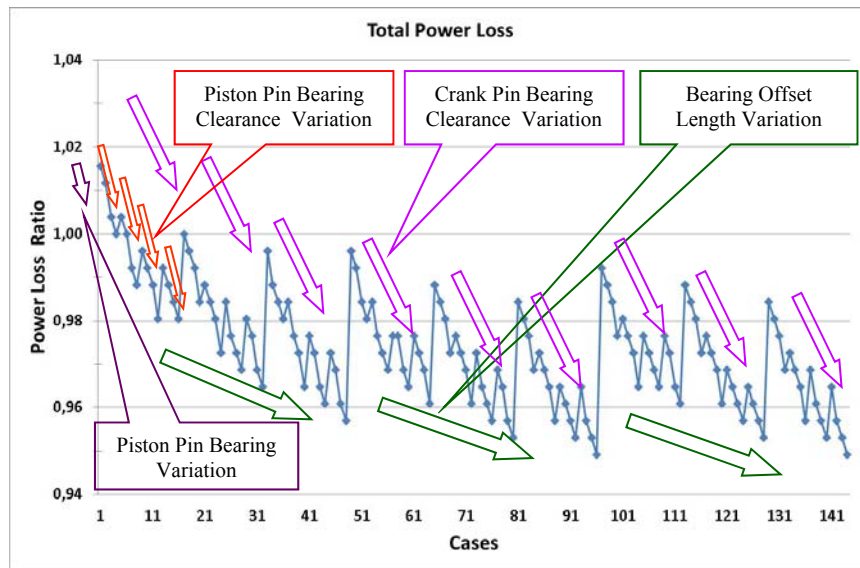


Figure 5. Total power loss ratio results of bearings

There is critical maximum film pressure limit for every application based on the experience. For this reason evaluation of the numerical analysis with experimental results is very crucial for the determination of the limits. On the other hand, critical lower limit of the minimum film thickness changes with the surface roughness characteristic of the bearing and journal.

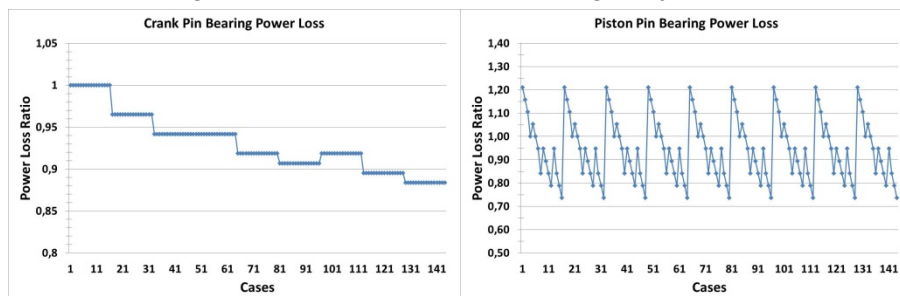


Figure 6. Power loss ratio results of crank pin/piston pin bearing

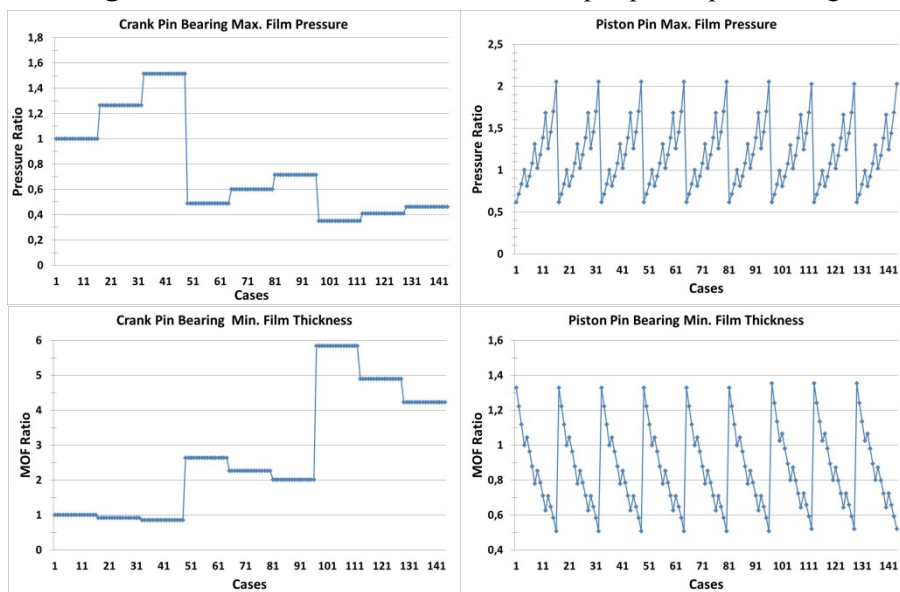


Figure 7. Max. film pressure and min. film thickness ratio results of crank pin/piston pin bearing

All the results of the numerical analysis were investigated in detail with statistical techniques. Some of the cases given in Table 4. were selected for summarizing the interaction between the parameters. Case4 represents the base design. Total power loss difference was evaluated with regard to base design. Variation of the maximum oil film pressure and minimum clearance with respect to the crank and bearing angle were also investigated in this study at different cases.

Table 4. Summary of the numerical analyses results

ASHRAE 5 cSt	Crank Pin Bearing (CPB)			Piston Pin Bearing (PPB)			TOTAL Power loss Ratio	Dimensions			
	Max. film pres. Ratio	Min. film thickness Ratio	Power loss Ratio	Max. film pres. Ratio	Min. film thickness Ratio	Power loss Ratio		CPB Clearance Shaft Diameter	PFB Clearance Shaft Diameter	Piston Pin Bearing Shaft Diameter	Bearing offset Shaft Diameter
Case1	1,00	1,00	1,00	0,62	1,33	1,21	1,02	1,3E-03	4,2E-04	9,2E-01	1,4E-01
Case5	1,00	1,00	1,00	0,81	1,04	1,05	1,01	1,3E-03	6,7E-04	9,2E-01	1,4E-01
Case4	1,00	1,00	1,00	1,00	1,00	1,00	1,00	1,3E-03	4,2E-04	6,7E-01	1,4E-01
Case16	1,00	1,00	1,00	2,05	0,51	0,74	0,98	1,3E-03	1,2E-03	6,7E-01	1,4E-01
Case36	1,51	0,86	0,94	1,00	1,00	1,00	0,98	1,9E-03	4,2E-04	6,7E-01	1,4E-01
Case52	0,49	2,64	0,81	0,94	1,00	1,00	0,98	1,3E-03	4,2E-04	6,7E-01	8,3E-02
Case48	1,51	0,86	0,81	2,05	0,51	0,74	0,96	1,9E-03	1,2E-03	6,7E-01	1,4E-01
Case144	0,46	4,23	0,76	2,03	0,52	0,74	0,95	1,9E-03	1,2E-03	6,7E-01	0

4.2. Experimental Results:

Parameters for the experimental studies given in Table 5. were defined according to the interaction between the numerical results of the cases. Components of the compressor were investigated with respect to the surface roughness, material, coating, form, clearance between journals and bearing housings before the assembling process of the compressor for eliminating the undefined influences. Same class components were selected for the assembling process.

Table 5. Defined experimental cases

Parameters	
Crank Pin B. Clearance/Shaft Diameter Ratio	1.9E-03 / 1.3E-03
Piston Pin B. Clearance /Shaft Diameter Ratio	1.2E-03 / 4.2E-04
Bearing offset/Shaft Diameter Ratio	1.4E-01 / 0
Piston Pin Bearing/Shaft Diameter Ratio	0.67
Upper Main Bearing/Shaft Diameter Ratio	1,1
Lower Main Bearing/Shaft Diameter Ratio	0,6
Crank Pin Bearing/Shaft Diameter Ratio	0.9
Piston Pin Bearing/Shaft Diameter Ratio	0.8
Kinematic Viscosity (at 40 ⁰ C)	5 cSt

Calorimeter measurements were conducted at ASHRAE test conditions to examine the performance characteristics of the investigated compressor samples. Repeatability and reproducibility of the measurement system evaluated during the tests. Minimum five samples with minimum three repetitions were tested in each configuration. The results of the performance measurements of the reciprocating compressor samples are shown in Table 6. According to the experimental results decreasing the bearing offset and increasing the clearance between the crank pin /piston pin journals and their bearings reduce the total power loss. Performance measurements of the compressor with the improved bearing design (Case 1) showed up to 3.8 % increase in the coefficient of performance (COP) with respect to the compressor with previous bearing design (Case 5).

Numerical and experimental studies for various operating conditions and oil viscosities will be useful for the overall bearing design of the compressor.

Table 6. Experimental test results of the samples

ASHRAE			Dimensions		
	Average	Comparison of	<u>C.P.B. Clearance</u>	<u>P.P.B. Clearance</u>	<u>Bearing offset</u>
5 cSt	Δ COP %	Num.&Exp. Results Δ COP %	Shaft D. Ratio	Shaft D. Ratio	Shaft D. Ratio
Case 1	3.8	2.8	1.9E-03	1.2E-03	0
Case 2	2.5	1.7	1.9E-03	4.2E-04	0
Case 3	2.2	1.6	1.3E-03	1.2E-03	0
Case 4	0.8	0.4	1.3E-03	4.2E-04	0
Case 5	0.0	0.0	1.3E-03	4.2E-04	1.4E-01

5. Conclusions

In this paper factors affecting the crank pin bearing and piston pin bearing optimizations are investigated:

- Decreasing the length of the bearing reduces the total power loss.
- Increasing the clearance between the crankshaft journal and its bearing decrease the total power loss.
- Optimization of the bearing dimensions (diameter/length/clearance) is critical to reduce hydrodynamic, mixed lubrication and boundary contact power losses and wear rate in the compressor.
- Statistical evaluation of the numerical analysis results show that the numerically calculated total power loss is closely related with efficiency gain measured at the experimental tests.
- Efficiency measurement of the compressor with the improved crank pin and piston pin bearing design showed up to 3.8 % increase in the coefficient of performance (COP) with respect to the compressor with base bearing design.
- This study shows that the mechanical loss characteristics are significantly influenced by the pin bearing offset and the pin bearing clearance.

Results of this study helps to characterize the optimum crank pin and piston pin bearing parameters which lead to improved overall mechanical efficiency of the compressor. Based on the results, this study will be extended by numerical analysis and experimental studies with respect to the parametric variation of operating conditions for various oil viscosities.

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Acknowledgement

The authors would like to express their sincere appreciation to Arçelik A.Ş.