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International Compressor Engineering Conference
School of Mechanical Engineering

2014

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An Investigation on the Bearing Design and Friction Characteristics of a Hermetic Reciprocating Compressor

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ABSTRACT

The most important design parameters for hermetically sealed compressors used in domestic refrigerators are the Coefficient of Performance (COP); low manufacturing and operating costs; long operating life, low noise and vibration levels and to achieve environmentally friendly constructions. Positive displacement piston type compressors are used in refrigerators today due to their high capacity/volume ratios. Hence, the bearings of the hermetic compressors must also satisfy these design conditions.

Investigation and optimization of crank shaft bearings in hermetic compressor applications are the main objectives of this study. The effect of crank shaft geometry, bearing clearance, lubricant viscosity, surface roughness and bearing location along the shaft on the friction losses were investigated and the new journal bearing designs were developed. Detailed parametric numerical simulations were performed using commercial software. According to the results of the simulations number of compressors were assembled with the selected design parameters and performance measurements were carried out. The results of the numerical analysis have shown that the numerically calculated mechanical loss level is similar to the performance results measured in a calorimeter test system. Results of the simulations and experiments were evaluated with six sigma (6σ) techniques.

Measurement of the efficiency of the compressor with the improved bearing design showed between 3.2% and 4.5 % increase in the coefficient of performance (COP) with respect to the compressor with previous bearing design. This study shows that the mechanical loss characteristics are significantly influenced by the length of the bearings, clearance between the crankshaft journal and its bearing, kinematic viscosity and operating conditions. Bearing analysis results help to characterize the optimum journal bearing parameters which lead to improved mechanical efficiency of the compressor.

INTRODUCTION

A hermetic reciprocating compressor is the most effective component (90%) with respect to the energy consumption of the refrigerators. Therefore performance improvement studies of the compressor play an important role to reduce overall energy consumption of the refrigerators. Electromotor, thermodynamic and mechanical losses are the main three factors which effect the compressor performance.

The performance of a reciprocating compressor is influenced by the mechanical losses. 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 losses are bearing dimensions, types of bearing, lubrication characateristics, friction coefficient, load, surface roughness, temperature, material, coating, relative velocity, clearance between journals and bearing housings. Investigation and optimization of crank shaft bearings in hermetic compressor applications are the main objectives of this study . The effect of crank shaft geometry, bearing clearance, lubricant viscosity, surface roughness and bearing location along the shaft on the friction losses were investigated and the new journal bearing designs were developed.

The design of the optimum bearing influences the efficiency of reciprocating compressors. In order to increase the COP of the compressor an efficient bearing design which provides optimum oil film pressure/thickness and minimum friction loss must be used. Furthermore the optimum bearing design helps to reduce the vibration produced by dynamic behavior of the compressor.

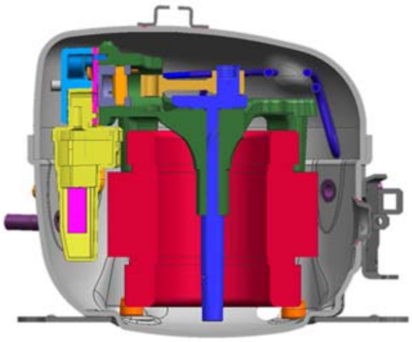

Figure 1: 3D CAD model of the studied reciprocating compressor

In addition to the theoretical analysis, numerical simulations with engineering software coupled with experimental studies have become an essential development tools in the improvement of performance of reciprocating compressors. Positive efforts have also been made for the complexity of the physical phenomenons inside the compressor ambient. A lot of research work dealing with the use of software coupled with experimental studies as a design tool for various compressor components can be found in the literature. There is also some remarkable research in the development of optimum bearing designs.

M. Duyar [1] et al developed an elastohydrodynamic lubrication (EHL) model and used it for the design improvement of reciprocating compressor bearing. The effects of lubricant viscosity, nominal clearance, pin material, bearing diameter and bearing length were studied and optimized. In the following study [2] M. Duyar et al used the developed EHL model [1] to improve the bearing design to avoid wear. J.Kim et al [3] studied lubrication characteristics between the crankshaft and the journal bearing. The importance of clearance between the crankshaft and the journal bearing, the length of the bearing and the diameter of the crankshaft on lubrication was verified with theoretical analysis. M. Matsui et al [4] developed an analytical model for mixed lubrication in bearings of reciprocating compressors for refrigerators. H.J. Wisbecket al [5] presented a dynamic model for the lubrication process of bearing systems including solid friction and wear. Journal bearings in hermetic reciprocating compressors were analyzed experimentally and numerically by B.J. Kim, et al [6]. In this study the critical Sommerfeld number was proposed as a design criterion.

EXPERIMENTAL STUDIES

In order to conduct the mechanical analysis and determine the necessary boundary conditions for numerical calculation, a pV set-up was built and calorimeter measurements were performed. Piezo-resistive miniature pressure transducers flush mounted in the valve plate were used to measure the pressure inside the cylinder for bearing force investigations. Compression work was done on gas by piston during the compression phase of the compressor. The gas force on piston creates reaction forces on crankshaft bearings. These reaction forces must be carried in the bearings by the oil film pressure. For the determination of the cylinder volume an optical encoder was placed on the shaft, from which the position of the shaft can be determined. From the position of the shaft the piston position was calculated and also the momentary cylinder volume. pV measurements were conducted at ASHRAE test conditions to examine the pressure characteristicsof the investigated compressor. The results of pV measurements of the reciprocating compressor are shown in Figure 2. Furthermore detailed temperature measurements were also done at various points inside the compressor.

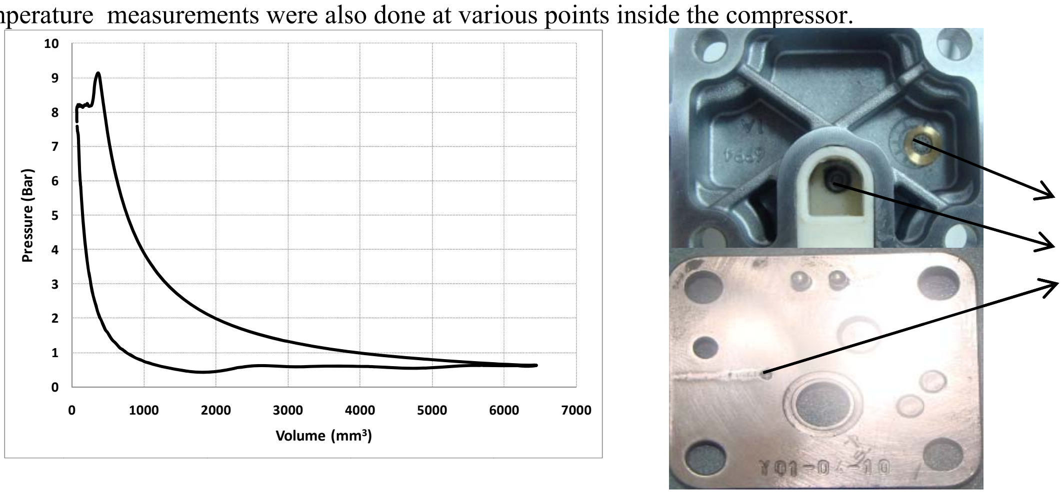

Figure 2: pV-diagram and temperature/pressure measurement locations

NUMERICAL ANALYSIS

Commercial enginering software for analysing the dynamics of the crankshaft and its interacation with the cylinder block was used. Software provides a number of different solution techniques for predicting the crankshaft mechanism dynamics using models of varying degreses of sophistication. Following conditions has been taken into account for carrying out the bearing analysis.

- Body crank hole lenght is fixed
- Boundary contact friction coefficient is 0.08
- Kinematic viscosity variation with respect to the temperature of the lubricant calculated from Walther-ASTM equation.
- Cylinder pressure variation with respect to the crank angle is used from the ASHRAE condition pV results.

Computational Model

Compressor crankshaft, piston, connecting rod and piston pin were rigidly modelled as shown in Figure 3. Meshing is a process where the calculation domain is divided into a number of cells. Mesh quality plays significant role in the accuracy and the stability of the numerical solution. Circular and axial mesh nodes were used in this model with respect to the length (L.) of the bearing. Analyses properties and boundary conditions were given in Table 1.

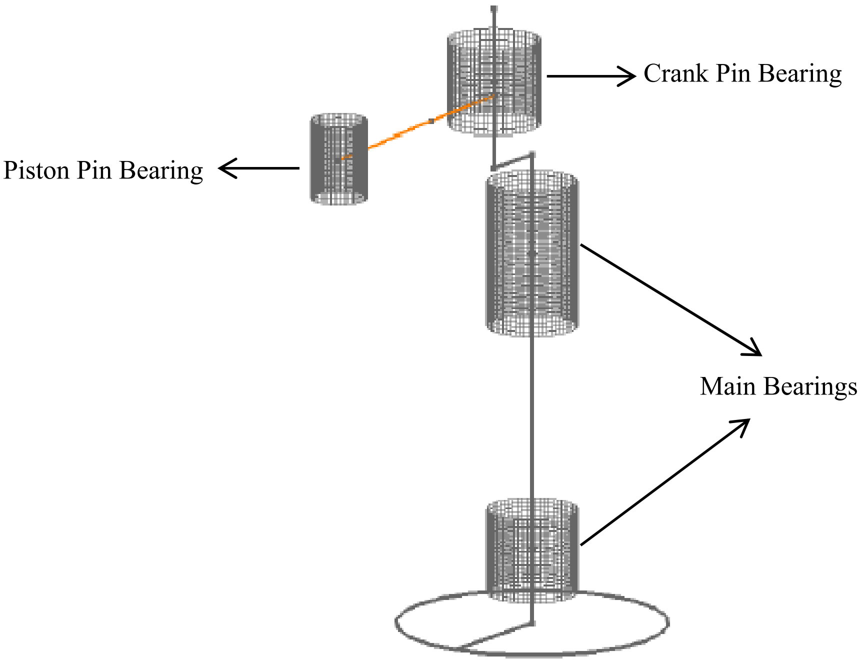

Figure 3: Model of bearings and computational domain

Table 1: Analyses properties and boundary conditions

Upper Main Bearing L. / Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes	Lower Main Bearing L. / Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes
1.6	39	73	0.9	23	73
1.3	33	73	0.8	19	73
1.1	27	73	0.6	15	73
0.8	21	73	0.4	11	73
Big End Bearing L. / Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes	Small End Bearing L. / Shaft Diameter	# of axial mesh nodes	# of circular mesh nodes
0.9	23	73	0.8	28	61

Parameters and Operating Conditions

To analyze the bearings inside compressor appropriate parameters and operating conditions must be applied. Table 2 shows the parameters and operating conditions for numerical analysis of a journal bearing and the dynamic behavior analysis of the reciprocating compressor. Finite difference discretization method and the Half Sommerfeld boundary condition to predict oil pressure at the boundary condition 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	
Clearance/Shaft Diameter Ratio	1.0E-03 – 6.7E-04
Upper Main Bearing/Shaft Diameter Ratio	1.6
Lower Main Bearing/Shaft Diameter Ratio	0.8
Crank Pin Bearing/Shaft Diameter Ratio	0.9
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 (^{o}C)	80
Kinematic Viscosity (at 40^{o}C)	5 cSt
Nominal oil pressure (bar)	0.624
Working condition	ASHRAE
Maximum rated speed (rev/min)	2930

Solution Method

Solution methodology was given in Figure 4. Cylinder pressure variation with respect to the crank angle for one cycle of the compressor was taken from the pV results. Cylinder pressure diagram was used for kinematic analysis and bearing reaction forces were calculated for every crank angle. At each time step 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 the entire compressor cycle. Maximum film pressure, minimum film thickness, hydrodynamic power loss and boundary lubrication power loss were calculated in the last step of the solution.

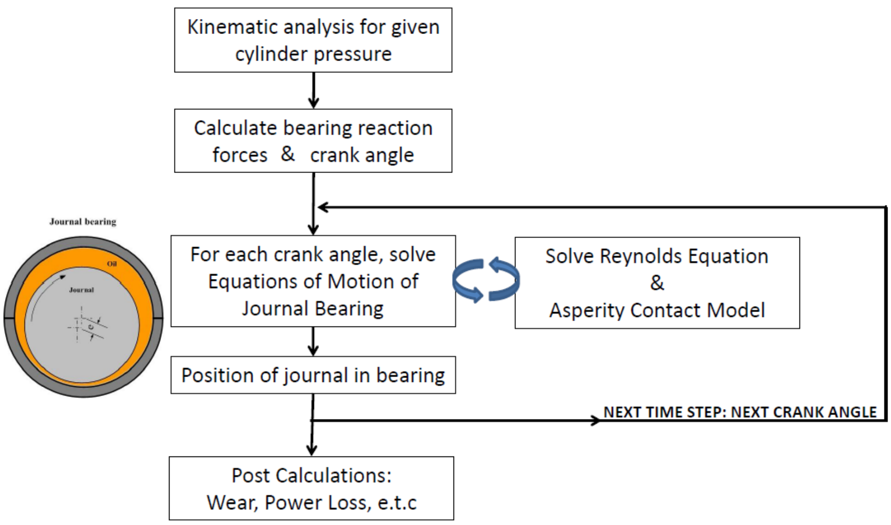

Figure 4: Flow chart of solution method

RESULTS AND DISCUSSIONS

The results of numerical computation in the bearings are examined in the 4th cycle of computation for obtaining the cyclic convergence for the all cases. We were able to confirm that the computational results were in good agreement with the experimental values.

Numerical Results:

In this numerical study 64 different combinations of the parameters given in Table 3 were used for hydrodynamic bearing analysis. The load acting on the shaft 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	
Clearance/Shaft Diameter Ratio	1.7E-03/ 1.3E-03/ 1.0E-03/ 6.7E-04
Upper Main Bearing/Shaft Diameter Ratio	1.6 / 1.3 / 1.1 / 0.8
Lower Main Bearing/Shaft Diameter Ratio	0.9 / 0.8 / 0.6 / 0.4
Crank Pin Bearing/Shaft Diameter Ratio	0.9
Piston Pin Bearing/Shaft Diameter Ratio	0.8
Kinematic Viscosity (at 40°C)	5 cSt
Working condition	ASHRAE

Maximum film pressure, minimum film thickness and total power loss which includes the sum of hydrodynamic power loss and boundary lubrication power loss were calculated for every case. The variations of normalized total power loss at the main bearings with respect to the defined cases were given in Figure 5. Maximum film pressure and minimum film thickness results for upper/lower main bearings were given in Figure 6, Figure 7.

According to the evaluation of the numerical results; decreasing the length of the bearings reduces the total power loss, 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 and boundary contact losses and wear in the compressor bearings.

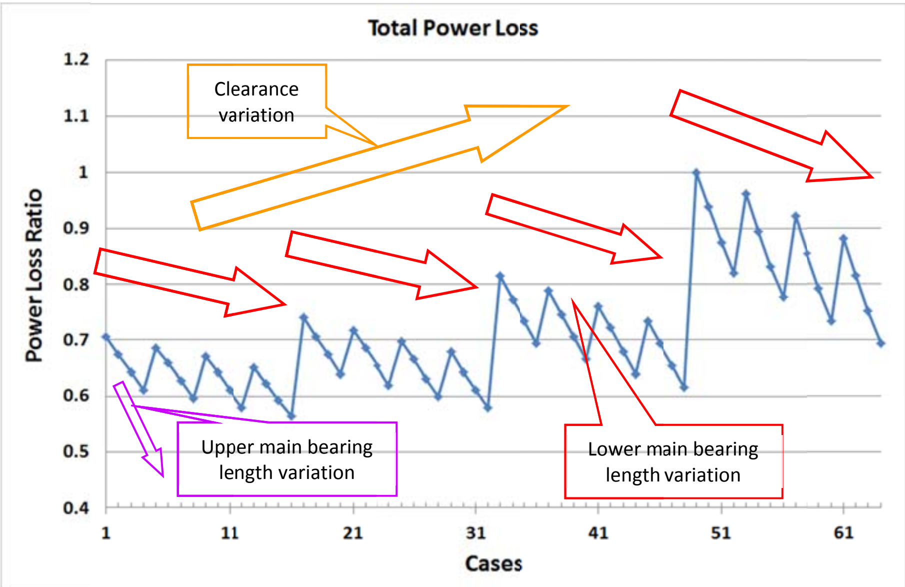

Figure 5: Total power loss ratio results of main bearings

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 roughness of the bearing and journal.

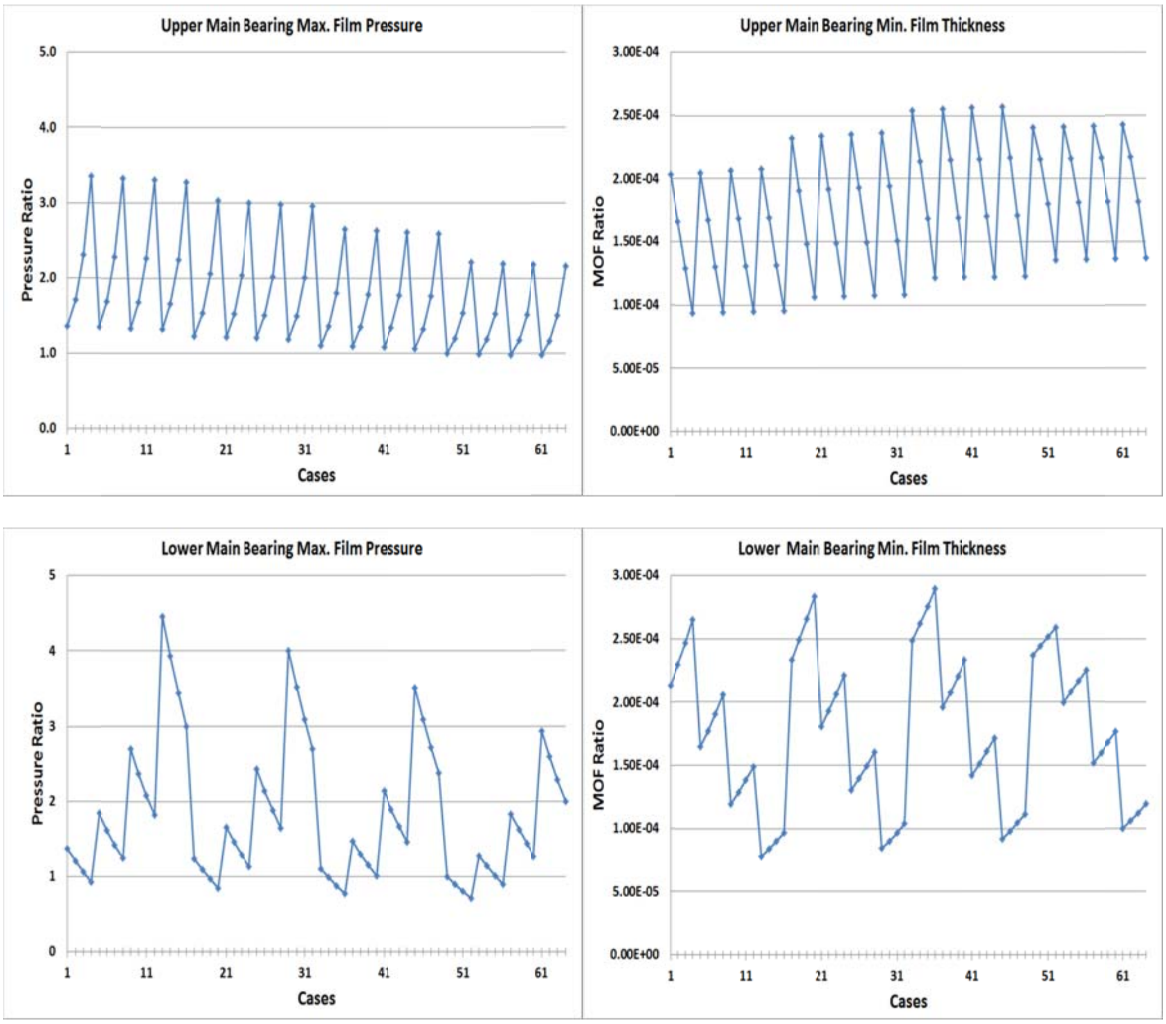

Figure 6: Max. film pressure and min. film thickness ratio results of upper/lower main bearing

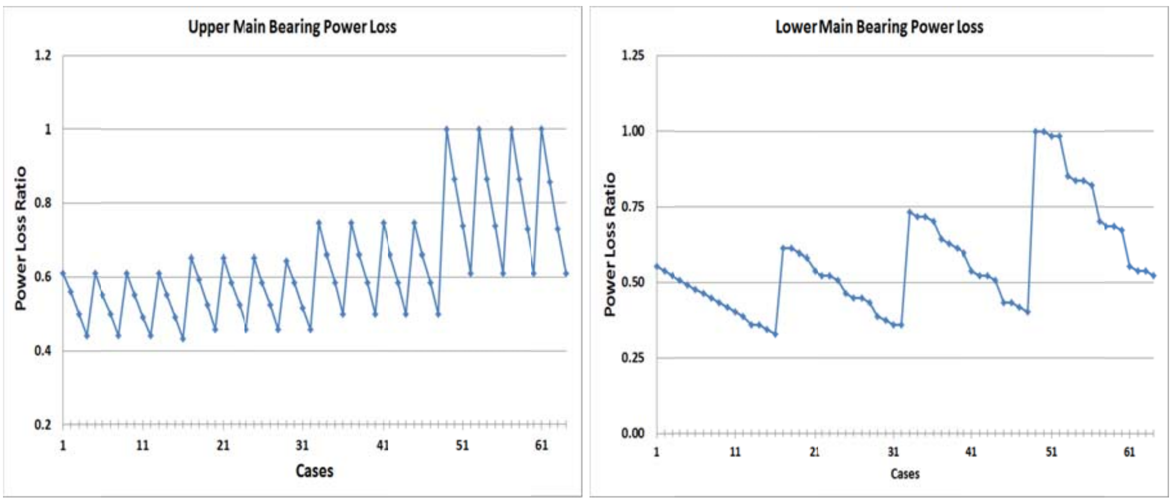

Figure 7: Power loss ratio results of upper/lower main bearing

All the results of the numerical analysis were investigated in detail with six sigma (6σ) techniques. Some of the cases given in Table 4 were selected for summarizing the interaction between the parameters. Case33 and Case 49 represent the base models. Total power loss difference was evaluated with regard to base models. Parameters for the experimental studies were defined according to the interaction between the results of the cases. Detailed variation of the maximum oil film pressure and minimum clearance with respect to the crank and bearing angle also investigated for different cases during this study.

Table 4: Summary of the numerical analyses results

ASHRAE	Upper Main Bearing			Lower Main Bearing			TOTAL	Dimensions		
5 cSt	Max. film pres. Ratio	Min. film thickness Ratio	Power loss Ratio	Max. film pres. Ratio	Min. film thickness Ratio	Power loss Ratio	Power Loss Ratio	Clearance / Shaft Diameter	Up. M.Bearing / Shaft Diameter	Low. M.Bearing / Shaft Diameter
Case16	3.30	9.53E-05	0.43	3.00	9.7E-05	0.33	0.56	1.7E-03	0.8	0.4
Case11	2.26	1.31E-04	0.49	2.08	1.38E-04	0.40	0.61	1.7E-03	1.1	0.6
Case3	2.30	1.29E-04	0.50	1.06	2.47E-04	0.52	0.64	1.7E-03	1.1	0.9
Case43	1.77	1.70E-04	0.58	1.67	1.61E-04	0.52	0.68	1.0E-03	1.1	0.6
Case33	1.10	2.54E-04	0.75	1.11	2.49E-04	0.73	0.82	1.0E-03	1.6	0.9
Case49	**1.00**	**2.41E-04**	**1**	**1**	**2.37E-04**	**1**	**1**	6.7E-04	1.6	0.9

Experimental Results:

In this experimental study different combination of the parameters given in Table 5 were used for preparing appropriate reciprocating compressor samples. Parameters were defined according to the interaction between the results of the numerical analysis.

Table 5: Defined experimental cases

Parameters	
Clearance/Shaft Diameter Ratio	1.7E-03 / 1.0E-03 / 6.7E-04
Upper Main Bearing/Shaft Diameter Ratio	1.6 / 1.1 / 1.0
Lower Main Bearing/Shaft Diameter Ratio	0.9 / 0.6
Crank Pin Bearing/Shaft Diameter Ratio	0.9
Piston Pin Bearing/Shaft Diameter Ratio	0.8
Kinematic Viscosity (at 40°C)	5 cSt
Working condition	ASHRAE

Calorimeter measurements were conducted at ASHRAE test conditions to examine the performance characteristics of the investigated compressor samples. The results of the performance measurements of the reciprocating compressor samples are shown in Table 6. According to the experimental results decreasing the length of the bearing and increasing the clearance between the crankshaft journal and its bearing reduces the total power loss. Performance measurements of the compressor with the improved bearing design (Case 1) showed between 3.2% and 4.5 % increase in the coefficient of performance (COP) with respect to the compressor with previous bearing design (Case4 and Case 5).

Table 6: Experimental test results of the samples

ASHRAE		Dimensions		
5 cSt	Average ΔCOP %	Clearance / Shaft Diameter	Up. M.Bearing L. / Shaft Diameter	Low. M.Bearing L. / Shaft Diameter
Case 1	**4.5**	**1.7E-03**	**1.1**	**0.6**
Case 2	**3.2**	**1.7E-03**	**1.6**	**0.9**
Case 3	**2.6**	**1.0E-03**	**1.1**	**0.6**
Case 4	**1.3**	**1.0E-03**	**1.6**	**0.9**
Case 5	**0.0**	**6.7E-04**	**1.6**	**0.9**

After the optimization of the main bearings, this study will be extended for the optimization of the other bearings *(Crank Pin Bearing and piston pin bearing).* Numerical analysis and experimental studies with respect to the parametric variation of clearance and length of the other bearings will be useful for the finalizing the complete bearing designs of the compressor. In addition to this investigation of the interaction between the bearings with different working conditions and oil viscosities can provide useful design information.

CONCLUSIONS

In this paper crankshaft bearing optimization were investigated. Bearing parameters under dynamic working conditions of the compressor were also studied in detail.

- Decreasing the length of the bearing reduces the total power loss, 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 and boundary contact losses and wear in the compressor.
- The results of the numerical analysis evaluated with six sigma (6σ) techniques have shown that the numerically calculated calculated total power loss rate is close to efficiency gain measured with a calorimeter test system.
- Measurement of the efficiency of the compressor with the improved bearing design showed between 3.2% and 4.5 % increase in the coefficient of performance (COP) with respect to the compressor with previous bearing design.
- This study shows that the mechanical loss characteristics are significantly influenced by the length of the bearings, clearance between the crankshaft journal and its bearing. Bearing analysis results help to characterize the optimum journal bearing parameters which lead to improved mechanical efficiency of the compressor.

This study will be extended in the second step of this optimization:
Numerical analysis and experimental studies with respect to the parametric variation of ;

- Clearance at Crank Pin Bearing and piston pin bearing;
- Lenght of Crank Pin Bearing and piston pin bearing;
- Working conditions
- Oil viscosities,

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ACKNOWLEDGEMENT

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