OPTIMIZATION OF THE CRANKSHAFT JOURNAL BEARINGS EFFICIENCY

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Abstract: *The hermetic compressor is widely used in household refrigerators. About 10% (10-15W) of the power consumption of the compressor is converted to heat due to the friction losses in the journal bearings, and about 76.4 million kWh of energy is spent on friction losses in the annual energy consumption of refrigerators sold in one year. In this work, the energy efficiency of the refrigerator compressor is increased by the reduction of friction losses in journal bearings. The model for kinematic and dynamic analysis of the crankshaft-connecting rod in the hermetic compressor is established to determine the forces acting on the crankshaft and the load-bearing capacity in journal bearings. For all these investigations, in-house built MATLAB codes are used. The results of the kinematic analysis are compared to the CAD analysis. The eccentric ratio for hydrodynamic lubrication conditions is determined. The bearing dimensions are optimized to minimize energy loss.*

Keywords: *kinematic analysis, compressor, journal bearing, crankshaft, energy efficiency, friction*

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1. Introduction

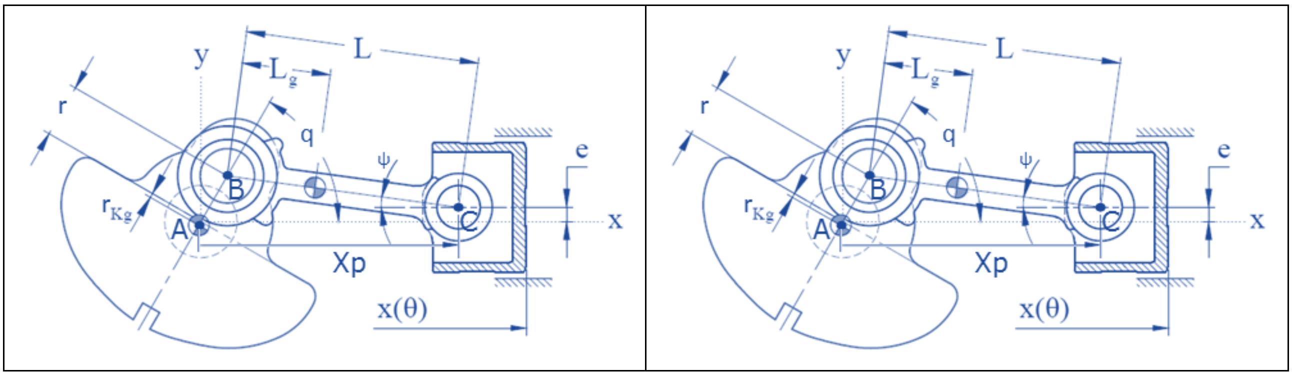
Increasingly the depletion of fossil fuels worldwide leads to an increase in research and development activities on energy efficiency and the usage of renewable energy. Efficient use of energy in residences is as essential as in industry. When it comes to the energy consumed in residences, 10-15% of this energy is consumed by refrigerators and freezers. Also, approximately 90% of the energy consumed by the refrigerator is consumed by the compressor. So, some researchers are also focused on the improvement of compressor technologies. One of them explains the requirements for energy saving by optimizing the compressor operation, which has the calculation of cooling processes (Hadziahmetovic, H., 2016). Reducing the vibration of the turned-on hermetic compressor by the derived MATLAB Simulink model and creating a new suspension system by verification and accuracy analysis is optimized (Kogani et al., 2021).

Another perspective is that the roughness parameters of the mechanical components, which specifically impact hydrodynamic lubrication, affect the surface quality and tribological behavior of dynamic systems (Durakbasa N.M et al., 2019; Durakbasa N.M et al., 2019). In addition, the effects of energy consumption labeling of refrigerators (ISO 5390, 1977; ISO 5149-2, 2014) in the world market force manufacturers to produce more efficient refrigerators (Krousgrill, C. M., 1998). Therefore, compressors designed with high energy efficiency are of utmost importance to produce high energy efficiency class devices. In this article, the friction losses which lead to energy costs in the radial journal bearing of the compressor are calculated analytically according to the hydrodynamic lubrication theory.

The kinematic and dynamic analysis of all the forces and moments, consisting of the axial journal bearing on the crank-pressing surface and the radial journal bearings of the crankshaft due to the pressure force, was carried out in both the MATLAB and ADAMS programs. Depending on the analysis results, the maximum eccentricity ratio in the present system and the eccentricity ratio which may occur in the hydrodynamic lubrication conditions were identified. Depending on the load-bearing capacity and knowing the analysis results, the optimum bearing length required to achieve the maximum eccentricity ratio, which produces the minimum friction, is presented. This article aims to design a more precise mathematical model that considers more variables that affect the performance and energy efficiency of journal bearings, such as refrigerators and piston cylinder mechanisms. Thus, the friction at the journal bearings reduces by optimizing the energy efficiency through a mathematical algorithm.

2. Methods

Equations have already been obtained according to Fig. 1 using the kinematic and dynamic analyzes, which are used for creating the pattern of the system in MATLAB.



a. b.

Fig. 1. a) The analyzed crankshaft-connecting rod mechanism, b) Crankshaft center of bearing dimensions (Bodur, O., 2014; Özen, S., 2015).

2.1 The Kinematic Analyzes of the Mechanism

Each component's position, speed and acceleration values are calculated as a function of time to determine the joint forces of the mechanism by performing dynamic analysis. The connecting rod angle (q), Crankshaft angle (ψ) and all other dimensional parameters are given in Fig.1a, where A is the center of Crankshaft rotation, B is the center of the hinged assembly between the Crank and the Connecting Rod. Similarly, C is defined as the center of the hinged assembly between the Connecting Rod and the Slider (Piston) (Bodur, O., 2014; Top A.B., 2011). The slider displacement is given as in Eq. 1. depending on the Crankshaft angle (q) (Bodur, O., 2014). The slider velocity and acceleration are calculated as the first and second derivatives of the Slider displacement with respect to time (Bodur, O., 2014).

$$X_p = r \cdot \cos(q) + L \cdot \sqrt{1 - \left(\frac{e + r \cdot \sin(q)}{L}\right)^2} \tag{1}$$

2.2 The Dynamic Analysis of the Mechanism

Reaction forces of the joints are obtained concerning time in dynamic analysis. The reaction forces can be obtained using Newton's second law (Kurka et al., 2012). Equations are obtained by using the moment balance to define the forces which act on the crankshaft in Fig. 1b (Kurka et al., 2012).

2.3 The Alteration of Eccentric Ratio in The Existing Crank Shaft

By analyzing the existing journal bearings of the system, which is dependent on the angle when the eccentric ratio is calculated, the situation of the journal bearings should be characterized by depending on the eccentric ratio and the Sommerfeld numbers. By taking the consideration of its current situation, the length of the upper journal bearing should be optimized to reach better conditions in the hydrodynamic lubrication.

The formula of the load bearing capacity, which is depicted below, is described based on the research in that field by taking the consideration of eccentricity ratio of the bearings, the viscosity of the oil that is used for the bearings, radial bearing clearance and the geometric parameters of the bearing (Harnoy, A., 2002; ISO 7902-1, 2020):

$$W = \frac{\mu U L^3}{4C^2} \cdot \frac{\varepsilon}{(1-\varepsilon^2)^2} [\pi^2(1-\varepsilon^2) + 16\varepsilon^2]^{1/2} \tag{2}$$

Within the results of the kinematic and dynamic analysis that is carried out for crankshaft- connecting rod, the alteration of the forces, which depend on the angle of the crankshaft, that are applied to both upper and lower bearings are calculated by using MATLAB or ADAMS. For that reason, the alteration of the eccentric ratio is going to be assessed based on the result of that two analyses.

Afterward, the maximum eccentric ratio is designated according to the maximum forces in the system. The alteration of the load-bearing capacity for both bearings, in the condition of a hydrodynamic lubrication system, is found as a result of the kinematic and dynamic analyses, and it is utilized for the change in the maximum eccentric ratio in the system. The change of the eccentric ratio is calculated by using Eq. 2. Therefore, interpolation calculation is used in MATLAB.

2.4 Designation of The Minimum Lubrication Thickness

To reach the condition of hydrodynamic lubrication, the surface roughness of both crankshaft and bearing shouldn't contact each other in any case, and there should be a lubrication thickness between them. For this reason, the designation of the minimum lubrication thickness in the existing system leads us to the maximum eccentric ratio that occurs with the maximum force application.

Maximum radial clearance that occurs in the surface roughness, in other words, minimum clearance for the occurrence of hydrodynamic lubrication, is calculated by using Eq. 3. According to ISO 4287 and 4288; by using Ra surface roughness with the equivalence of its Rt values which should be used in the equation below (ISO 4287, 2002; ISO 4288, 1998):

$$\mathrm{h_{0min}} = 1{,}2(\mathrm{R_{tM}} + \mathrm{R_{tY}}) \tag{3}$$

Where h is the variable of film thickness due to the journal eccentricity ($\varepsilon = e/C$) (Harnoy, A., 2002). The minimum lubrication thickness in the existing system occurs at the maximum eccentric ratio (Bodur, O., 2014):

$$\mathrm{h_{min}} = \mathrm{C}\,(1 - \varepsilon_{\mathrm{maks}}) \tag{4}$$

In the condition of hydrodynamic lubrication, to calculate the friction coefficient and the loss of power for the upper bearing, on which the eccentric ratio is calculated, Sommerfeld numbers should be used from an eccentric ratio to Sommerfeld numbers graph (Stolarski T.A., 1990; Byung et al., 2002).

Our purpose is to increase the eccentric ratio. However, it shouldn't be forgotten that when the system is running, the lubrication thickness should be more than the maximum clearance that happens between surface roughness since the transition eccentric ratio through to the condition of hydrodynamic lubrication (ε_g) is going to be calculated in parallel with that (Bodur, O., 2014):

$$h_{0min} = C\left(1 - \varepsilon_g\right) \tag{5}$$

The friction coefficient, which depends on the Sommerfeld number, occurs on the bearing because of the lubrication and is shown below in the equation; in the case of So<1 (Stolarski, T. A., 1990):

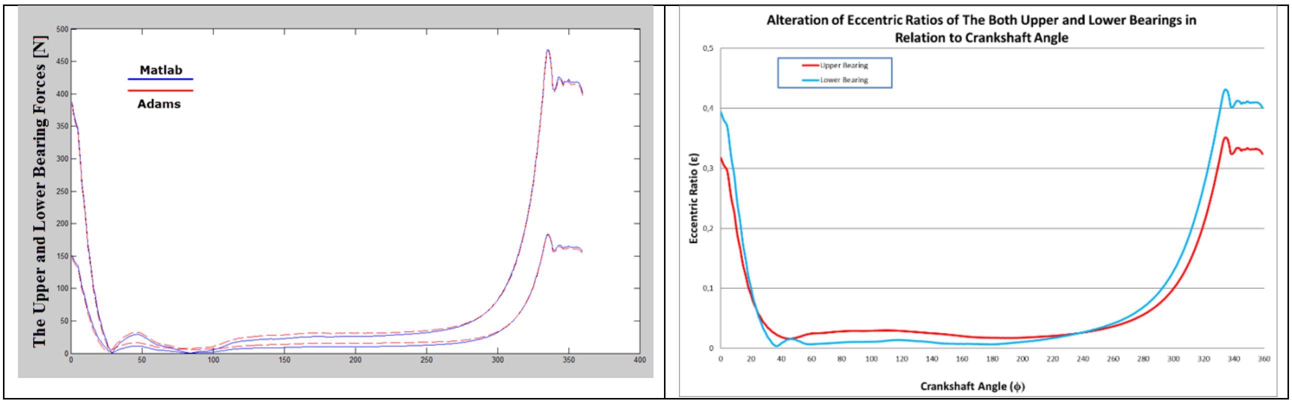
$$\mu_f = 3\psi/So \tag{6}$$

The loss of power that occurs due to friction is calculated by using the equation below (Stolarski, T. A., 1990):

$$P_s = \mu_f \, F \, U \tag{7}$$

3. Results and discussion

By creating the CAD model of the system, which is prepared using the ADAMS program with considering the same conditions, kinematic and dynamic analyses are carried out for crankshaft bearings in Fig. 2a.



a. b.

Fig. 2. a) Effect of the upper and lower bearing forces on the radial journal bearing of the crankshaft, b) Alteration of eccentric ratios of upper and lower bearings in relation to Crankshaft Angle.

According to the comparison, the analysis results of the forces, which are applied to both upper and lower bearings, are very close to each other (maximum force difference is around 1N). The alteration of the eccentric ratios of both upper and lower bearings is obtained under variable forces that depend on the crankshaft angle, as shown in Fig. 2b, according to both MATLAB and ADAMS results. The angle of the highest eccentric ratio of both the upper and lower bearings is the angle that maximum force affects. According to Eq. 5, $\mathbf{h_{0min}}$ can be found.

When the eccentric ratio is at maximum, which is $\boldsymbol{\varepsilon}_{\mathbf{maks.(Matlab)}}$, existing lubrication thickness can be found by using Eq. 4. According to that, $\mathbf{h_{min}}$ is calculated for MATLAB and ADAMS analyses. In this case, since $\mathbf{h_{min}} > \mathbf{h_{0min}}$ hydrodynamic lubrication conditions are provided. The minimum lubrication thickness ($\mathbf{h_{min}}$) that is occurred in the lower bearing, is compared with the minimum clearance ($\mathbf{h_{0min}}$) that occurred for the hydrodynamic lubrication according to Eq. 3. The maximum eccentric ratio, which is already calculated in MATLAB, is used to calculate the minimum lubrication thickness that occurred in the lower bearing.

According to Eq. 4; when the minimum lubrication thicknesses are compared with each other, $\mathbf{h_{min(Matlab)}} < \mathbf{h_{0min}}$, mixed friction conditions are available (Dagilis et al., 2009; Matsui et al., 2010). The system works in the condition of elastohydrodynamic conditions. It can also be defined as a situation where no lubrication can completely separate the particles between the surfaces (Dagilis et al., 2009; Matsui et al., 2010).

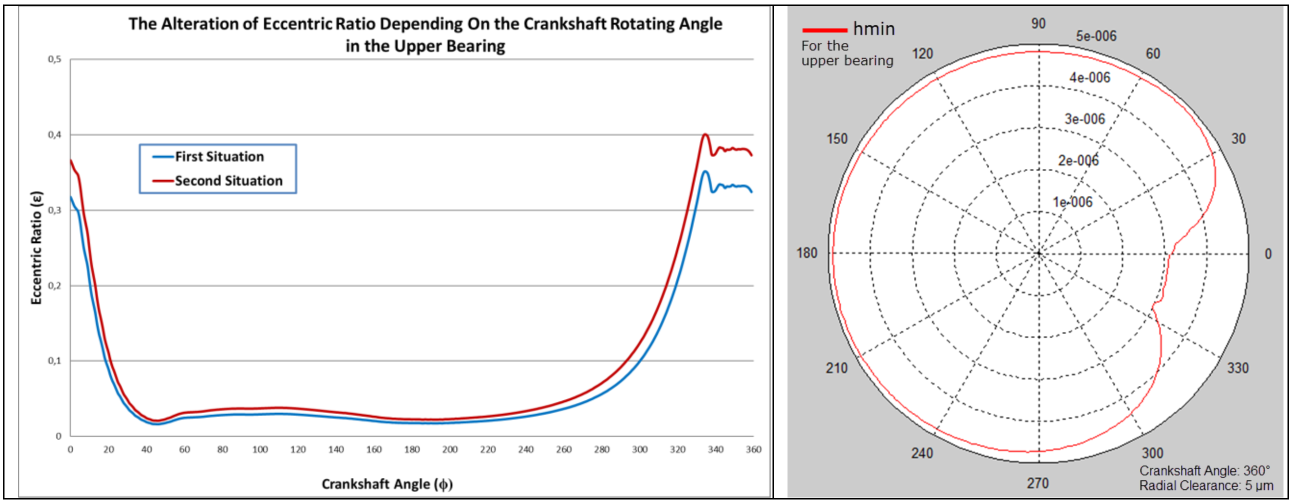
3.1 Optimization of the Length of the Upper Bearing

When the existing upper bearing is assessed, it reaches the hydrodynamic lubrication conditions. Optimization of the bearing is carried out to reach suitable lubrication conditions. Eq. 5 is used depending on the transition eccentric ratio through to the hydrodynamic lubrication system to draw close to a stable region. As mentioned in Eq. 3, by using the values of the maximum clearance that occurs between radial clearance and surface roughness, the transition eccentric ratio is calculated as (ε_g).

When the measurements of the existing bearing are considered, it cannot surpass that ratio. When we assess Eq. 2, it is observed that there is an inverse ratio between the length of the bearing and the eccentric ratio. This also means that if an increase in the eccentric ratio is needed, it can be done with a decrease in the length of the bearing. As long as the inputs except for the length of the bearing are fixed, the calculation of the interpolation by using MATLAB can be carried out, which also means that the optimized length of the bearing, which provide us with the transition eccentric ratio, can be calculated. When the optimization is done according to Eq. 2, the optimized length of the bearing can be found.

Bearings that are smaller than this length causes an increase in the eccentric ratio and decrease the minimum lubrication thickness. However, the values which are higher than this number should be defined as design parameters.

The pressure on the lubrication is altered due to the change in the length of the bearing. Moreover, Sommerfeld numbers are affected and increased. The alteration of the eccentric ratio, which depends on the crankshaft rotating angle, as a result of the change in the length of bearing in Fig. 3a and the alteration of lubrication thickness, which also depends on the crankshaft rotating angle, are shown in Fig. 3b.



a. b.

Fig. 3. a) The alteration of eccentric ratio depending on the crankshaft rotating angle for a decrease in length of the upper bearing, b) The alteration of minimum lubrication thickness depending on the crankshaft angle for a decrease in the length of the bearing.

3.2 Optimization of the Length of the Lower Bearing

The eccentric ratio of the lower bearing is higher than the transition eccentric ratio due to the low length of the lower bearing. For that reason, the eccentric ratio of bearing is optimized by considering the transition of eccentric ratio.

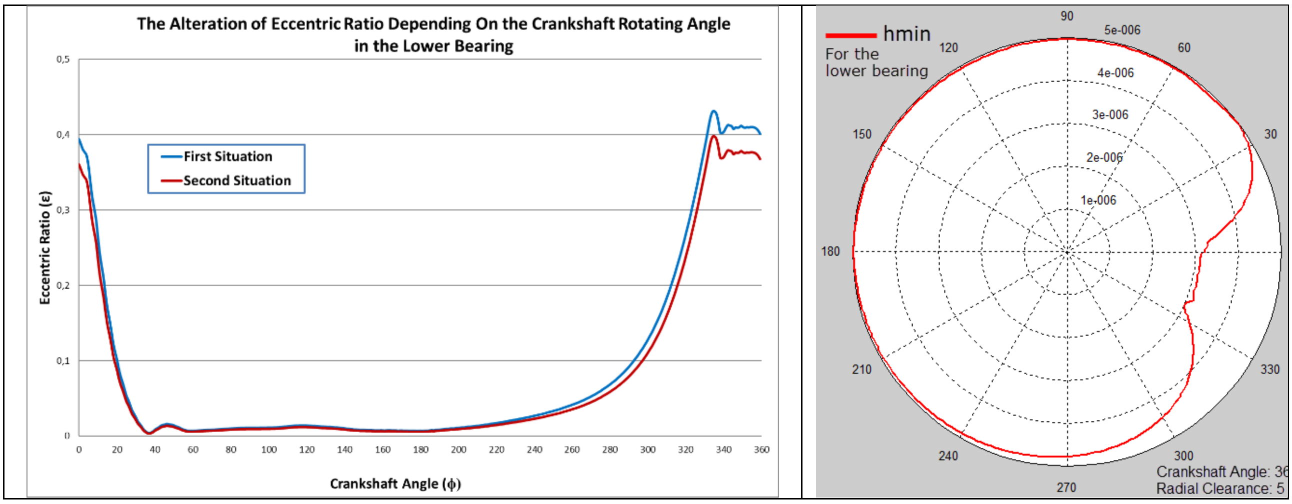


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Fig. 4. a) The alteration of the eccentric ratio depending on crankshaft rotating speed for an increase in length of the lower bearing, b) The alteration of the minimum lubrication thickness for a decrease in the length of bearing.

Thus, the length of the bearing that is suitable for the hydrodynamic conditions can be determined. The optimization of the lower bearings can be carried out by using the same interpolation calculation that is already used in MATLAB for upper bearings. The length of the lower bearing that provides us with the transition eccentric ratio can be calculated.

The pressure on the bearing is changed due to the alteration of the length of the bearing, and Sommerfeld numbers are affected. As a result of the alteration of the length of the bearing, the alteration of eccentric ratio in relation to the crankshaft rotating angle in Fig. 4a and the alteration of the minimum lubrication thickness in relation to the crankshaft rotating speed is shown in Fig. 4b.

4. Conclusion

The reason that explains why the forces that affect bearings are calculated by using analyzes of dynamic and kinematics both in MATLAB and ADAMS is because of the consistency in the system. By creating a MATLAB pattern, forces of the bearings of the different crankshaft mechanisms can be calculated, and systems, which are in the engine, are also included.

When the length of the upper bearing is decreased and the length of the lower bearing is increased, which can be seen in the upward figures, lubrication thickness is observed in a better way. As a result of two optimizations, it can be observed that the friction coefficient is decreased, and the loss of power is comparatively decreased.

Alteration of the upper bearing length prompts the friction coefficient to be decreased in the condition of hydrodynamic lubrication. Compared to that, the loss of power that occurs in the upper bearing is decreased by 11%. Alteration of the lower bearing length in the hydrodynamic lubrication condition, except mechanical losses, causes the friction coefficient that occurs before the lubrication takes place to be reduced and comparatively to that, the loss of power is reduced by 8%. In this regard, the losses that occur as a result of the contact of the crankshaft with the journal bearing can also be observed.

This research is able to be used to predict and optimize the parameters that affect the tribological behavior of journal bearings in piston cylinder mechanisms. Thus, the energy consumption and friction of the compressor in refrigerators or other similar systems are likely to be optimized. In the subsequent scientific studies, some accurate mathematical algorithms would be generated, including the detailed roughness parameters, surface quality, manufactured material properties and other variables that involve journal bearings' power losses.

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