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Research on optimization of low-frequency mechanical efficiency of Reciprocating Compressor for Refrigerator

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Abstract. Lubricating oil and bearing clearance are important parameters that affect the lubrication state of the compressor at this time. The main purpose of this paper is to study and optimize the influence of the lubricating oil and bearing clearance on the performance of the compressor. The losses of the different friction parts were obtained through simulation research. At the same time, via analysing the viscosity of the lubricating oil and the bearing clearance, this paper has got the best parameters and greatly improved the compressor low frequency efficiency. Through further experimental analysis, the results are in good agreement with the simulation analysis conclusions. This study is helpful to increase the low frequency efficiency in guiding the design of reciprocating compressors.

1 Introduction

The implementation of the new energy efficiency standard for Chinese refrigerators in October 2016 (GB 12021.2-2015) puts forward higher requirements on the energy efficiency of compressors. Experimental research shows that low speed operation is a better choice. Under this condition, the system load is lighter, the compressor operating rate is higher, the start-stop loss is small, and the energy efficiency is better. At the same time, the temperature control is more stable and the noise is smaller. Through experimental research, the power distribution of a compressor at 16HZ (960rpm) ASHRAE is shown in Figure 1, which shows the proportional relationship between motor loss, controller loss, mechanical loss and indicated power loss clearly. The mechanical loss under this condition is up to 38%. Optimizing mechanical efficiency is an important means to improve the COP (coefficient of performance) at low frequency.

Figure 2 shows the components of the compressor mechanical losses clearly. The mechanical frictional power dissipation is generated in the crankshaft/crankcase, crankshaft pin bearing, piston pin bearing, thrust bearing and piston/cylinder bearing parts. In addition to insulate and support of the bearing surface components, the lubricating oil located in the bearing part can also take away the heat generated by the bearing and create conditions for the stable and reliable operation of the compressor. How to improve the cop at low frequency and reduce bearing losses is worth studying and solving.



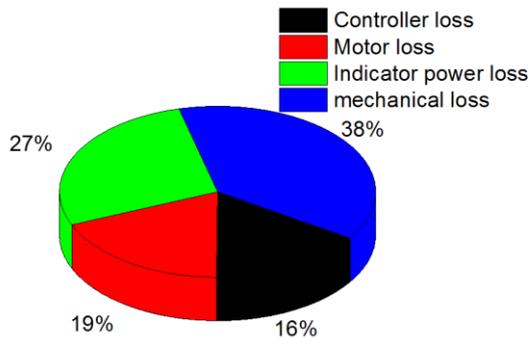


Figure 1. Power loss distribution of compressor

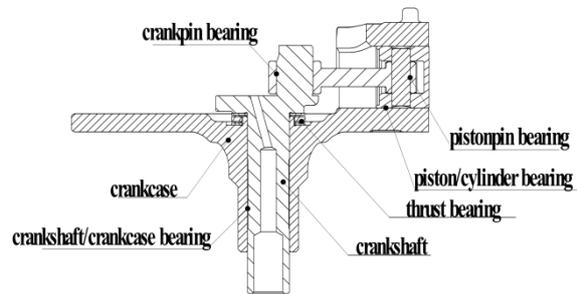


Figure 2. Schematic diagram of friction part of reciprocating compressor

Masaru MATSUI^[5] studied a numerical calculation method for calculating the bearing loss in the mixed lubrication state of the compressor. Rodrigo LINK^[6] calculated the cone cylinder of the compressor numerically and obtained the piston cylinder power consumption and its regularity for the operating condition of 50HZ/60HZ. Takuma TSUJI^[7] also used theoretical numerical methods to optimize the power consumption of compressors with different ratios of cylinder eccentricity.

The traditional researches on the reciprocating compressor bearing are often directed to 50Hz/60Hz running condition, the compressor lubrication state is good, the regularity of the shaft loss of different models tends to have good consistency and predictability and there are still few studies for the industry to low frequency shafting. When the compressor is running at low speed, the lubrication condition deteriorates, and the influence of roughness and surface texture and the shafting material on frictional power consumption becomes more prominent. The influence of lubricating oil properties on the shafting system will show different rules from the 50Hz/60Hz operation. In view of this, it is of great significance to investigate low-frequency state bearing systems.

In view of many parameters such as bearing width, diameter, clearance, oil, material, surface roughness and stroke cylinder diameter ratio affecting on small reciprocating compressor bearings, this paper mainly deals with the influence of two important parameters that oil and bearing clearance on the power loss of different bearing parts for the compressor during under 960r/min operation.

2. Preliminary research

To obtain the input load of the compressor, this paper has established a cylinder pressure calculation method considering the coupling process between the valve and the refrigerant and obtained the cylinder pressure as a function of the rotation angle as shown in Figure 3. The calculation works for the 9.1cc displacement prototype under standard operating condition. The speed is 960r/min and the cylinder pressure has an important relationship with the mechanical loss of the bearing system.

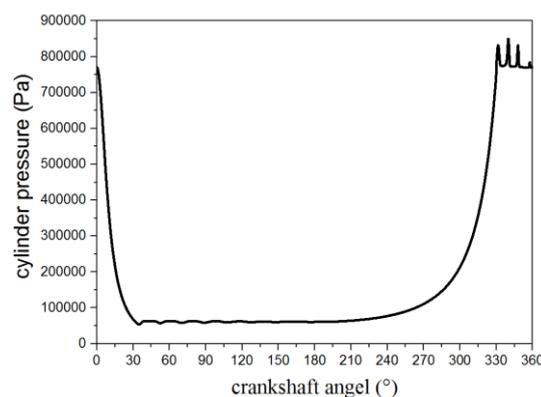


Figure 3. Diagram of Cylinder Pressure Change with Angle

3. Analytical model

3.1. Theoretical background

The model of the sliding bearing liquid force in contact with the microscopic surface is based on the average Reynolds equation of the elastic fluid lubrication theory, assuming that the liquid is incompressible. The oil pressure calculation of the oil film is based on the analytical formula of Patir/Chang^[1,2] (1978 and 1979), which is applicable to the Gaussian surface distribution hypothesis.

$$\frac{\partial}{\partial x} \left(\varphi_x \frac{h^3}{12\eta} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\varphi_y \frac{h^3}{12\eta} \frac{\partial p}{\partial y} \right) = U \frac{\partial}{\partial x} [\theta(h_T + \sigma\varphi_s)] + \frac{\partial(\theta h_T)}{\partial t} \quad (1)$$

At the same time, the solid contact model is based on the calculation theory of Greenwood and Tripp (1970)^[5], and the contact pressure considers the elastic deformation of the contact surface. The contact pressure is obtained by shear stress and integrated on the bearing surface to obtain the corresponding contact power consumption. Detailed descriptions and the meaning of the relevant parameters can be found in the relevant literature^[1,2,5].

$$P_a = \frac{16\sqrt{2}\pi}{15} (\sigma_s \bar{\beta} \eta_s)^2 \sqrt{\frac{\sigma_s}{\beta}} E^* F_{\frac{5}{2}}(H_s) \quad (2)$$

3.2. Problem analysis and modeling

When the compressor was operated at low speed under the ASHRAE condition, the experiment indicated that a rapid drop of COP. The analysis of the power consumption of the compressor showed that the low-frequency mechanical loss had an increasing trend. In order to analyse the mechanical loss, the calculation model has been established as shown in Figure 4. It includes the main bearing and the rotary motion part of the crank pin, the swing of the piston pin and the small end of the connecting rod and the reciprocating linear motion friction between the piston and the cylinder. The lubricating oil passes through the spiral groove from the bottom of the crankshaft. It is added to the main bearing and crank pin bearing parts and is inserted into the piston pin and piston by splash lubrication. This simulation model considers the interaction between shafting dynamics and elastic fluid lubrication.

To study the effect of lubricant and bearing clearance on low frequency mechanical loss, six lubricants were collected and their viscosity and density data are listed in Table 1. It is assumed that the viscosity of the lubricant is a single function of temperature. At the same time, the oil temperature during the operation of the compressor was tested. Table 2 demonstrates the bearing clearance data for this calculation. Other parameters such as roughness, material elastic modulus, Poisson's ratio and mass properties as well as bearing sizes, working temperature and pressure conditions are input according to actual conditions and the calculation consistency is guaranteed.

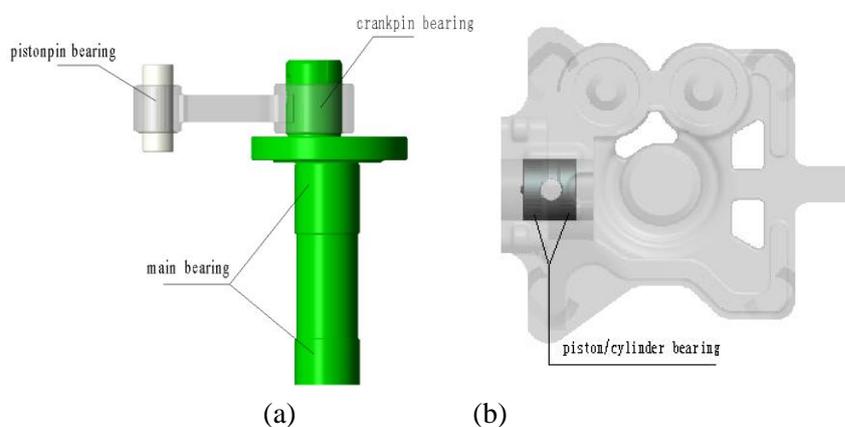


Figure 4. Schematic diagram of bearing parts for analysis

Table 1. Different types of oil kinematic viscosities (cst) and density (g/cm³) data

	Oil type 1	Oil type 2	Oil type 3	Oil type 4	Oil type 5	Oil type 6
Viscosity@40° C	2.2	3.2	4.22	5.65	7.98	11.63
Viscosity@100 °C	0.98	1.2	1.42	1.69	2.22	2.7
density	0.86	0.861	0.861	0.8649	0.8686	0.873

Table 2. Different bearing parts clearance (mm)

crankshaft bearing	crankpin bearing	Piston pin bearing	piston/cylinder
0.008/0.012/0.016/0.02	0.02	0.005	0.007

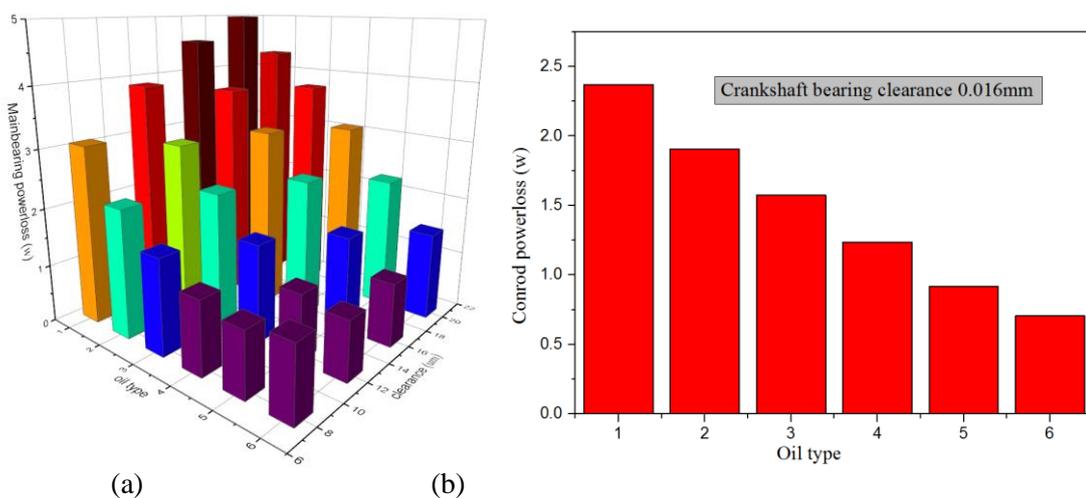
In order to obtain the analysis results, it is necessary to make some basic assumptions about the calculation model:

- 1) The oil supply temperature is a constant value and is obtained experimentally, and the temperature of the bearing surface also has a constant value;
- 2) Lubricating oil is not compressible, has a constant density and specific heat properties;
- 3) Each component is an elastomer;
- 4) The internal flow of the bearing is laminar.

4. Simulation results

4.1. crankshaft/conrod part

The calculation model simulates the energy losses of the compressor under ASHRAE at 960rpm, and the cylinder pressure condition is used as the model input load. In this paper, different types of lubricating oil and 24 combinations under the gap are used to calculate the power consumption changes of the main bearing, the connecting rod big head and the connecting rod small head given in figure 4.a and draw a graph given in figure 5. In addition to the liquid internal friction power consumption, the total power consumption value of each example also includes the rough contact power consumption of the bearing surface.

**Figure 5.** Power consumption diagram of main bearing and conrod

In the low frequency operation, the variation rule of the crankshaft/conrod mechanical loss for the compressor was shown in Figure 5, the main bearing power consumption accounts for the main part, while the loss of the connecting rod component is relatively low. As the viscosity of the lubricating oil

increases or the main bearing clearance decreases, the value of viscous powerloss increases, meanwhile, the oil film carrying capacity is improved, which is advantageous for improving the contact state of the bearing. Especially in the initial stage of the exhaust process and the expansion process, at this time, the pressure in the cylinder is high, the load is large, the bearing working condition is poor, and the peak contact of the bearing surface is severe, resulting in an increase in power consumption or even low-frequency wear. Since the variation of the viscous power consumption is converse from the contact power consumption, which leads to the existence of minimal powerloss. Therefore, increasing the viscosity of the lubricating oil or reducing the main bearing clearance is advantageous in reducing bearing loss at low frequency.

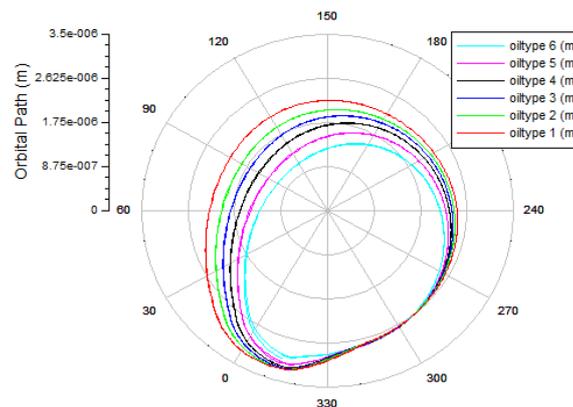


Figure 6. Shaft trajectory change with different types of lubricating oil diagram

The above simulation results can be explained that the change of the bearing attitude affects the contact state of the bearing, thereby affecting the contact loss. Figure 6 shows the center axis trajectory of the lower main bearing part for the same main bearing clearance with different types of oil, indicating that the increase in the viscosity of the lubricant oil is beneficial to reduce the misalignment of the crankshaft motion process, making the crankshaft motion closer to the center position, thereby reducing Contact pressure and contact loss between the crankshaft and the cylinder.

4.2. Piston/Cylinder part

The simulation of this object was the linear motion of the piston and cylinder friction using the same cylinder pressure as input condition, shown in Figure 4b. The effect of different types of lubricant oil on the loss was analysed, as shown in Figure 7, the results showed that the viscosity of the lubricating oil is positively correlated with the loss, which is opposite to the law of the main bearing. This can be explained by the fact that the lateral force of the friction portion of the piston cylinder is relatively low, and the proportion of the viscous loss of the oil film is large.

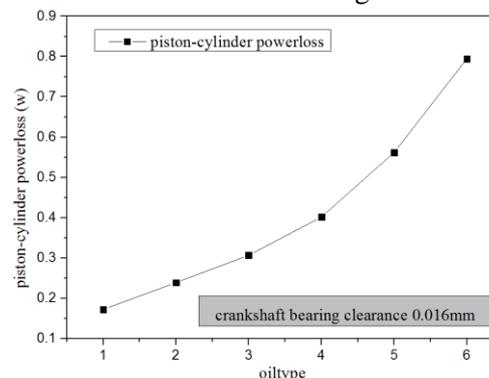


Figure 7. Relationship between different types of lubricating oil and frictional power consumption of piston-cylinder

For the total power consumption of the compressor bearing lubrication system, the change law is in good agreement with the main bearing as shown in Figure 8. The results show that under ASHRAE

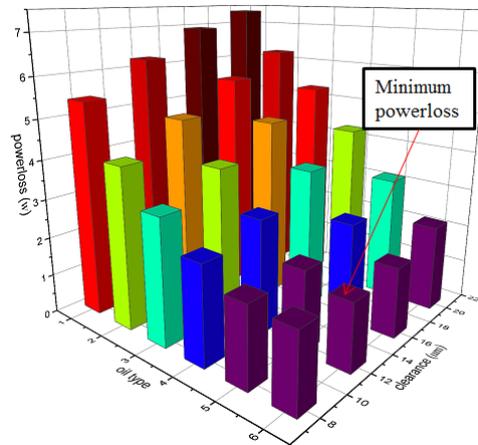


Figure 8. Total power consumption diagram under different viscosity and clearance

960rpm operating condition, using a clearance of 12μm and 6th type of lubricant oil, total power consumption reach the minimum.

4.3. Analysis of Simulation Results

The viscosity of lubricant oil and bearing clearance are two important indicators affecting compressor loss. Figure 9 depicts the contact loss and hydrodynamic power loss of the main bearing part at 960rpm operation condition with different clearances and viscosities of lubricant oil at 40°C. It shows that the hydrodynamic loss has an approximate linear positive correlation with the viscosity, and the contact loss has an approximate exponential decreasing relationship with the increase of viscosity. In the low viscosity region, contact loss accounts for a large proportion, and reducing contact loss is an important way to optimize the performance of the compressor under these conditions.

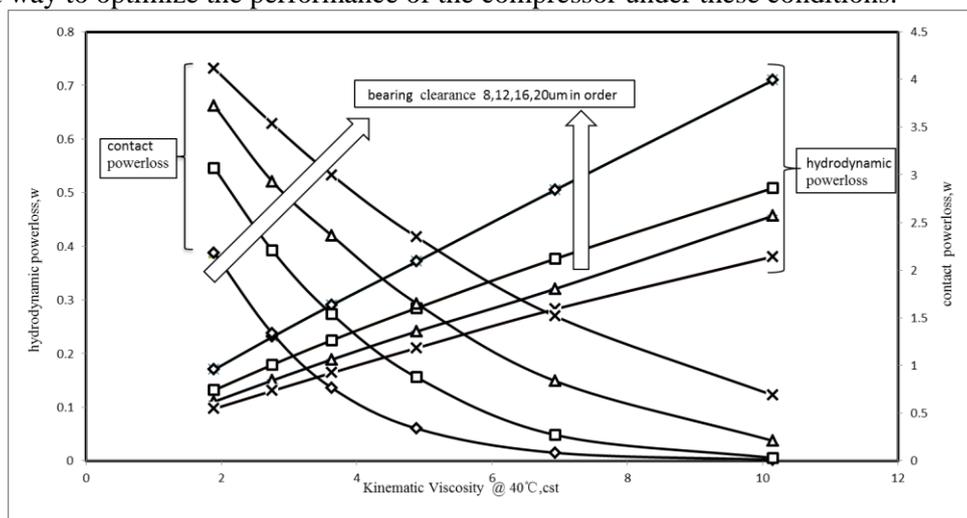


Figure 9. The contact loss and hydrodynamic powerloss of the main bearing part

According to the analysis above, the relationship between the different friction parts of the compressor and the viscosity is analysed by regression analysis and the relationship of total loss can be obtained, which can be used to analyse and predict optimum viscosity under different main bearing clearances:

$$w = kx + \sum_{i=1}^3 a_i e^{b_i x} + c \tag{3}$$

Here $w(w)$ and $x(\text{cst})$ represent the total power consumption of the bearing and the viscosity (cst) of the lubricant oil at 40°C, respectively, k, a, b, c are the undetermined coefficients, which are related to the bearing clearance, specific parameter of other bearings and the compressor load. Taking 8 μm as an example, the equation of shaft loss can be expressed as following:

$$W=0.2x+6.853e^{-0.476x}+3.14e^{-0.3889x}+0.9187e^{-0.08087x}+0.1605 \quad (4)$$

The correlation coefficient of formula predictions and simulation data is 0.998. From this, the minimum value shows that the power consumption of the lubricant oil using 7.47cst under the clearance of 8 μm is the lowest when operating at 960rpm. This method can be used to guide the optimization of compressor parameters and prediction for bearing performance.

5. Experimental analysis

To verify the simulation conclusions, the clearance of the main bearing lubricants and experimental analysis of the prototype performance differences under ASHRAE condition. The basic compressor information is shown in Table 3. The test bench diagram is shown in Figure 10. The refrigeration capacity is synthesized by heat balance and enthalpy difference method. The input power is measured by Zes-Zimmer LMG500 with a full range error of 0.01%. The flow meter adopts Emerson CFMS007M, the pressure sensor adopts keller PAA-33X, the error is controlled less than 0.05%. The temperature adopts PT100 and the specified precision is +/-0.3 degree.

Table 3. Basic compressor information

Parameter	Value	Unit
Volume of Cylinder	9100	mm ³
Operation speed	960~4200	rpm
Bore of Cylinder	24	mm
Cooling capacity	51@960 rpm	W
Refrigerant	R600a	

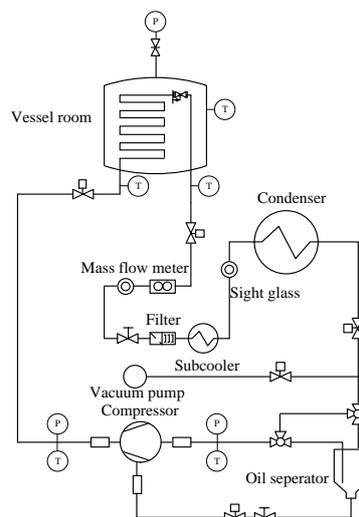


Figure 10. Schematic of designed test bench

The test was carried out using 2 prototypes. There are five combinations of main bearing clearance and types of lubricant oil and the other parameters of the prototype remained unchanged. The results are shown in Table 4. Here, the design parameters of the traditional prototype, as shown in sample 0,

are taken as a reference data, and the cop change rate of other samples is calculated based on the cop of this sample.

The relationship between the change of bearing clearance, the oil type and the cop was shown in Figure 11a. The consistency of two prototypes is good and compared to the conventional parameter setting, the cop increase by nearly 5% in the 8 μm of bearing clearance and nearly 7% using oil type 6 as shown in Figure 11b, increasing the viscosity also helps to improve the energy efficiency at 16Hz ASHRAE operating conditions.

The numerical analysis and experimental results in this paper have a good consistency with the prediction of the law, indicating that the proper reduction of the clearance or the increase of the viscosity of the lubricant oil is beneficial to reduce the contact loss under low-frequency operation, thereby improving the low-frequency energy efficiency of the compressor. For the influence of compressor power, covering all bearing lubricants, in addition to the analysis above, the analysis of others such as the thrust bearing portion distribution and the change law of the compressor mechanical power is meaningful.

Table 4. Specific experimental sample data

Sample/parameter	Mainbearing clearance , μm	Oil type
Sample 0	16	3
Sample 1	12	3
Sample 2	8	3
Sample 3	16	5
Sample 4	16	6

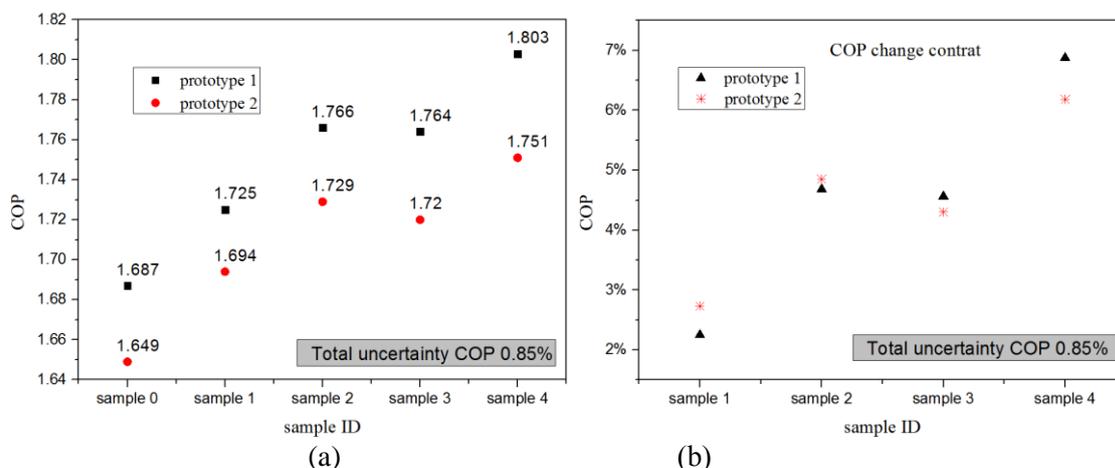


Figure 11. Change law of cop for different samples

6. Conclusion

In this paper, the power consumption of the compressor bearing system under the 960rpm low frequency ASHRAE operating condition was optimized. The computer model was used to analyse different friction losses for the reciprocating compressor using in small refrigerator between the piston and cylinder, the crankshaft and the connecting rod, the cylinder and the crankshaft. The loss of friction power varies with the types of lubricant oil and the clearance of the main bearing. Conclusion can be drawn as below:

1) Mechanical loss is the main factor affecting the power consumption of the piston at low-frequency. Simulation and experimental studies have shown that the different viscosities of lubricant oil and the clearance of main bearing have a greater impact on power consumption at low-frequency;

2) For the piston and cylinder, reducing the viscosity will help reduce the power consumption here; for the main bearing and connecting rod, the loss can be reduced by appropriately reducing the clearance of bearing or increasing the viscosity of the lubricant oil. This difference is determined by the relationship between the bearing load and the contact state.

3) Properly reducing the bearing clearance or increasing the viscosity of the lubricant oil is beneficial to improve the overall energy efficiency at low frequency. Through the test, the optimized clearance of bearing and viscosity of lubricant oil can increase the cop of the piston compressor by almost 7%.

In this paper, only the relationship between the mechanical loss at low-frequency ASHRAE condition and the clearance of bearing and lubricant oil were studied. The other operating conditions at low-frequency, high-frequency operating conditions and other parameters of the bearing will be studied in the next step.

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