Analysis of the effect of friction of hybrid ball bearings on grease evaporation in cold compressors

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Abstract. Bearings are the essential part of the cold compressor rotor system, which has a great influence on the operation of cold compressor. Hybrid ball bearing has been successfully applied in cold compressors with the lubrication of grease. With the high-speed rotation of the cold compressor, the bearing will be affected by the axial thrust of the impeller, which will cause friction, and the grease will be impacted by the frictional heat. To understand the influence of frictional heat on grease evaporation, it is important to study the bearing heat generation. In this article, the application of grease-lubricated hybrid ball bearings in cold compressors was introduced. Based on the parameters of a successfully applied impeller and a ball bearing of the cold compressor, the axial thrust of impeller was calculated, and a bearing frictional heat generation model was proposed, which considered the dynamic change of the axial thrust of the cold compressor impeller. The heat generation and temperature of the bearing was analyzed and calculated using the proposed model. According to the results of the calculation and practical experience, a test was performed on high-speed greases, and the adding amount of grease was estimated with reference to the bearing temperature.

1. Introduction  
For the production of superfluid helium, there are several solutions according to the needs of cryogenic systems. In large scale cryogenic systems, the cold compressor with a centrifugal impeller, which has been applied for many years, is the common equipment to depressurize the liquid helium tank in order to maintain the vacuum and obtain Helium II (superfluid helium) [1-3].  

At present, the core components of the mainstream cold compressor are centrifugal impeller, high-speed motor and rotor-bearing system, and there are three types of bearings: active magnetic bearing, grease-lubricated rolling bearing and gas bearing. For the application of rolling bearing in cold compressor, relevant cases and descriptions have been recorded in the literature [3, 4]. As the cold compressor runs at high speed, bearings are the main heat source in the motor cavity of the room temperature end. The research focus of this paper is a high-speed angular contact hybrid ceramic ball bearing (HCBB) used in cold compressor. The rolling element of HCBB is made of silicon nitride and the inner and outer rings are of bearing steel material. Compared with traditional all-steel bearings, HCBB has the characteristics of low heat generation and relatively long service life at high speed, which
can be attributed to the higher hardness and significantly lower density of the silicon nitride balls, and for this reason the loss of bearing during rotation is relatively small.

Based on the traditional quasi-static model, a HCBB analysis model considering the grease film friction between rolling element and raceway was established, and the friction power loss of HCBB was also modelled according to the actual operation characteristics of HCBB rotating at high speed. Using the model above and combined with the axial force calculation of the centrifugal impeller of cold compressor, the friction heat generation of HCBB was estimated. The evaporation of high-speed grease for HCBB was also evaluated through test, for which the reference grease temperature was predicted by a thermal network model built for HCBB. The results of the test provided a preliminary reference for the evaporation contamination of grease-lubricated bearing in cold compressor.

2. Frictional loss calculation based on quasi-static model

2.1. Quasi-static model of cold compressor HCBB

To date, quasi-static model has a lot of applications in rolling bearing analysis [5-7]. The coordinate system of the model established in this paper is shown in Figure 1(a). The bearing coordinate system is divided into the global coordinated system $oxyz$ and the local coordinated system $'o'x'y'z'$, among which the latter one is of the greatest importance. In the local coordinate system, the various forces and moments acting on the rolling element are balanced, and finally the external axial load is balanced in the global system. After solving the local balance equation and the global balance equation by iteration method, the dynamic characteristics of HCBB can be obtained.

When the cold compressor rotates at high speed, the inner ring of HCBB rotates with the rotor while the outer ring is fixed. At this moment, the bearing inner ring is subjected to the axial load $F_a$ from the impeller and the load is delivered to all rolling elements of the bearing (the number of rolling elements is $Z$). Inside the bearing, the rolling element is loaded with the contact force $Q_{i/o}$ caused by the elastic deformation between the rolling element and the raceway, as well as the centrifugal force $F_c$ and the gyroscopic moments $M_{y/o}, M_{z/o}$. HCBB is lubricated with grease, therefore, frictions $F_{x/o}, F_{y/o}$ exist in the contact area between the rolling element and the raceway, as well as closely related to the viscosity of the lubricating grease film. Meanwhile the rolling element is also affected by torques caused by grease film friction $M_{y/o}, M_{z/o}$. Due to the centrifugal effect, the rolling element moves to the groove bottom of the outer raceway. At this moment, the contact angle $\alpha_i$ between the rolling element and the inner raceway increases, while the inner one $\alpha_o$ increases, from which it can be concluded that the inner and outer contact angle are not equal during the bearing operation. According to the analysis above, the coordinate system and the balance relationship of single rolling element are shown in Figure 1(b).

![Figure 1](image-url)
Based on the Jones’s and Harris’s theory [8, 9], as well as the analysis above, each rolling element in HCBB has the following balance equation:

\[
\begin{align*}
Q_i \cos \alpha_i - Q_o \cos \alpha_o + F_{x_i} \sin \alpha_i - F_{y_i} \sin \alpha_i + F_{v_i} = 0 \\
Q_o \sin \alpha_i - Q_i \sin \alpha_o - F_{x_o} \cos \alpha_i + F_{y_o} \cos \alpha_i = 0 \\
F_{y_i} - F_{y_o} = 0 \\
M_{x_i} + M_{x_o} = 0 \\
M_{y_i} + M_{y_o} - M_{gy} = 0 \\
M_{z_i} + M_{z_o} - M_{gz} = 0
\end{align*}
\] (1)

For the axial thrust of the impeller, since the cold compressor has a vertical arrangement and the impeller is at the lower end of the shaft, the axial thrust has seven parts: the axial force caused by gas pressure and the momentum of the inlet gas, the force of inlet gas pressure acting on the front end of the impeller, the gravity and the gas pressure of the motor cavity which acts on the end of the rotor, as shown in Figure 2.

![Figure 2. Axial thrust of cold compressor rotor-impeller system.](image)

Since all the loads of the rolling elements are derived from the axial load of the impeller-rotor, the following formulas are used to describe the balance relationship between all the rolling elements and the axial load of the inner ring:

\[
\begin{align*}
F_{a} = F_{t} + F_{s} - F_{3} + F_{l} + F_{momentum} - F_{gravity} - F_{shaft} \\
F_{a} - Z(Q \sin \alpha_o + F_{x_o} \cos \alpha_o) = 0
\end{align*}
\] (2)

As the rotor rotates at a high speed, a swirl appears in the cavity between the semi-open impeller and the volute, which causes static pressure at the impeller back to be lower than that of the impeller outlet. Generally, the following formula can be used to calculate the pressure variation in the cavity [10]:

\[
\rho(r) = p_2 - \frac{1}{2} \rho \Omega^2 (r_2^2 - r^2)
\] (3)

where \( r_2 \) is the outlet radius of the impeller, \( p_2 \) is the outlet pressure, \( \Omega \) is the rational angular speed of the impeller, \( \rho \) denotes the gas density, and \( \Omega \) represents the swirl coefficient [10], of which the determination is complicated. It is generally assumed that the swirl coefficient is a constant value, in industry the value is often set as \( \Omega = 0.5 \) [10, 11], D. Japikse [12] gives a range of \( \Omega = 0.3-0.9 \) based on application experience. The common practice to determine the swirl coefficient is CFD analysis combine with experimental measurement [10, 13].

2.2. The friction loss model of HCBB

On the basis of results obtained through the quasi-static model, the friction loss model can be established. As the grease-lubricated bearing rotating, the friction loss mainly comes from the friction torque \( M_i \) caused by the ball-raceway contact load and the viscous torque \( M_v \) of the grease. For these torques, Palmgren [14] proposed two calculation formulas based on test data, as shown below:

\[
\begin{align*}
M_i = f_i P l d_m \\
M_v = \begin{cases} 
10^{-7} f_0 (vn)^{2/3} d_m^3 & \text{vn} \geq 2000 \\
160 \times 10^{-7} f_0 d_m^3 & \text{vn} < 2000
\end{cases}
\end{align*}
\] (4)
among which $P_1$ represents the equivalent dynamic load of the bearing, $f_1$ is the coefficient related to the bearing type load, $v$ is the kinematic viscosity of the grease, $f_0$ is determined by the bearing type, $n$ is the rotational speed, and $d_m$ means the pitch diameter of the bearing. The details of the definition of $f_0$ and $f_1$ can be found in the literature [9, 14].

The friction loss can be calculated as follows, which also represents the heat generation of the bearing:

$$H = M \cdot \omega$$  \hspace{1cm} (5)

The parameters of the HCBB and the grease involved in the research are listed in Table 1 and Table 2.

### Table 1. Parameters of the HCBB.

| Type                  | Unit   | 71905 |
|-----------------------|--------|-------|
| Material of ring      | Steel  |       |
| Material of ball      | Silicon Nitride | |
| Inner diameter        | mm     | 25    |
| Outer diameter        | mm     | 42    |
| Ball diameter         | mm     | 5.5   |
| Number of balls       |        | 15    |
| Contact angle         | degree | 15    |

### Table 2. Parameters of the HCBB.

| Characteristics       | Unit   | Value  |
|-----------------------|--------|--------|
| Density               | kg/m$^3$ | 940    |
| Thickener             |        | Lithium Complex |
| Base oil              |        | SHC/Ester oil |
| Base oil viscosity at 40°C | mm$^2$/s | 25 |
| Base oil viscosity at 100°C | mm$^2$/s | 6   |

2.3. Results and discussion

The axial thrust generated by the impeller-rotor is not only related to the pressure and the swirl coefficient, but also depends on the impeller rotational speed. According to the speed variation range under the design condition, the axial thrust is calculated as follows:

**Figure 3.** The axial thrust of the impeller with different swirl coefficients and rotational speeds.

The data in Figure 3 shows that the swirl coefficient and rotational speed have a grease impact on the axial thrust, which increases significantly with the rise of the swirl coefficient. As the rotational speed grows, the axial thrust increases more rapidly. At the same time, within the lower range of the swirl coefficient, the axial thrust is larger at low rotational speed, and this situation gradually changes with the swirl coefficient increasing.
The ball-raceway contact load grows with the increase of the swirl coefficient, and this can be attributed to the rise in the axial thrust induced by the increase of swirl coefficient, as shown in Figure 4. Meanwhile, in the case of swirl coefficient $f = 0.5$, the contact load decreases as the speed increases. This is because the axial thrust decreases with the increase of the rotational speed, in the lower range of the swirl coefficient, as shown in Figure 5. From the trend shown in Figure 3, it can be predicted that when the swirl coefficient reaches higher value, the contact load will start to change in the opposite way to Figure 5.

Figure 4. The effect of swirl coefficient on axial thrust and contact load ($n=5000\text{r/min}$).

Figure 5. The effect of rotational speed on axial thrust and contact load ($f=0.5$).

According to the analysis of Figure 3, the axial thrust of the impeller-rotor varies from 17.6N to 69.5N. Based on this, the HCBB frictional heat generation is shown in Figure 6. It can be observed that the heat generation increases with the increase of the axial thrust, and the rise of the rotational speed has a more obvious influence on the increase of the heat generation.

Figure 6. The frictional heat generation of HCBB.

3. Evaporation test
The grease is in a negative pressure environment when HCBB runs in the cold compressor. The evaporation of the grease may contaminate the motor cavity, thereby affecting the helium working fluid. Therefore, it is necessary to evaluate the application feasibility of the grease by evaporation test.

3.1. Thermal network of HCBB
In order to carry out the evaporation test, the bearing temperature field is required. Therefore, a heat transfer model containing several thermal nodes of HCBB was established, as shown in Figure 7. In general, it is assumed that 50% of the frictional heat is transferred to the rolling element, and the other half is transferred to the inner/outer ring.
Figure 7. Thermal nodes distribution of HCBB.

The nodes of the model are distributed in the shaft \( P_s \), inner ring \( P_i \), inner contact area \( P_{ci} \), rolling element \( P_b \), outer contact area \( P_{co} \), outer ring \( P_o \), bearing seat \( P_h \) and the environment \( A \). The heat transfer balance equation is as follows:

\[
\begin{align*}
\frac{T_o - T_A}{R_o + R_{o-A}} &= T_{co} - T_o \\
\frac{T_{co} - T_{A}}{R_{co} + R_{co-A}} + \frac{T_{co} - T_b}{R_{co}} + \frac{T_{co} - T_A}{R_{co}} &= 0.5H_o \\
\frac{T_b - T_A}{R_b} + \frac{T_{co} - T_b}{R_{co}} + \frac{T_{co} - T_A}{R_{co}} &= 0.5H_i + 0.5H_o \\
\frac{T_{co} - T_{co-A}}{R_{co-A}} &= 0.5H_i \\
\frac{T_{co} - T_{co-A}}{R_{co-A}} &= 0.5H_i \\
\frac{T_i - T_A}{R_i} &= R_{i-A} \\
\end{align*}
\]

(6)

where \( T \) is the temperature of each node, \( R \) is the thermal resistance between the nodes, \( H_{io} \) is the friction loss at the contact area of inner/outer raceway. In order to prevent the permanent magnets of the motor from being demagnetized, the temperature in the motor cavity of the cold compressor is controlled within 80°C, which should be recognized as an extreme value.

Taking air with negative pressure (5kPa) as the motor cavity environment, under the condition that the bearing is subjected to the maximum axial thrust (69.5N), the temperature of HCBB in the working speed range was calculated using the thermal network model. The result is shown in Figure 8.

Figure 8. Temperature variation at each node of HCBB.

The calculation results can provide a reference for the temperature prediction of the lubricating grease. Figure 8 shows the temperature of each node rises with the increase of speed, and the temperature of
rolling element is the lowest among the nodes. It can be concluded that the maximum temperature of all
the nodes is below 84°C. From the result it can be predicted that the grease temperature will not exceed
90°C during operation, which provides a reference for the subsequent experiment.

3.2. Method
The above-mentioned high-speed grease was used for test samples. The base oil of the grease is
SHC/Ester oil, the thickener is lithium complex soap, and the working temperature of the grease is 50-
120°C. A vacuum drying oven with the function of continuous depressurization while maintaining a
certain negative pressure was used as the evaporation equipment. A high-precision electronic balance
was used as the weighing device (with a precision of 0.1mg). Four glassware were used to hold the
samples which were evenly applied to ensure a uniform evaporation surface area. The test temperature
was set to 90°C according to calculation in section 3.1. In addition, based on the operating experience
of cold compressor, the test environment pressure and the duration were set to be 5kPa and 288 hours,
respectively.

3.3. Results and discussion
After 288 hours of testing, the grease lost a certain amount of mass. An abnormal sample data was
excluded due to operation error, and the results are shown in Figure 9. It can be estimated that the total
loss of the grease is between 0.67% and 0.73%, and the evaporation rate of the grease decreased
significantly over time.

![Figure 9. Mass loss ratio of the grease.](image)

![Figure 10. Average loss rate of the grease.](image)

In order to estimate the variation of the grease evaporation rate, the samples were weighed several
times during the test, and the average evaporation rate between adjacent time spots was calculated, as
shown in Figure 10. The average evaporation rate in the early stage decreased rapidly and then gradually
stabilized at a very low level. At the last test period of 240-288h, the average loss rate is $4.5 \times 10^{-5}$ g/h,
which indicates the evaporation keeps a very low loss rate during long-term operation.

For the suitable amount of grease added to the bearing, the manufacturer SKF recommends the
amount should be less than 30% of the available space inside the bearing [15]. Considering the structure
of HCBB and the grease density, it can be confirmed that the grease amount should not exceed 1.1g.

Combining the grease addition of 1.1g and the total grease loss (0.73%) from the above result, the
amount of grease loss in 288 hours is about $8 \times 10^{-3}$g. The flow rate of the cold compressor involved in
this article is about 200g/h, therefore the total mass of the helium during the test period is $2.2 \times 10^8$g.
According the two data above, it can be calculated the contamination level during the test is less than $4 \times 10^5$ppm. With 0.1ppm as the evaluation standard, the calculation results indicate the contamination
of the grease fully meets the requirement.
4. Conclusion
In this paper, a quasi-static model of grease-lubricated HCBB was established. The model considers not only the centrifugal and gyroscopic effects, but also the friction of grease film within the contact area of ball-raceway. The model also includes the dynamic change of the axial thrust as an important factor. The effects of swirl and rotational speed on axial thrust were studied and the contact load inside HCBB as well as the heat generation were also analysed. In addition, a bearing thermal network was modelled to calculate the temperature distribution of HCBB. Finally, an evaporation test was performed.

The following results were obtained from the above work:

The increase of swirl coefficient will lead to a significant increase of impeller axial thrust. At the same time, with a lower swirl coefficient, the axial thrust will decrease with the rise of rotational speed. When the swirl coefficient becomes greater, the axial thrust change in the opposite direction as the speed grows. Additionally, the contact load of ball-raceway and the heat generation of HCBB increases with the increase of axial thrust.

The temperatures of the nodes of HCBB are close in value, among which the temperature of the rolling element is the lowest. The increase of speed causes the increase of bearing temperature. Meanwhile, the evaporation rate of the high-speed grease will decrease significantly and maintain at a very low level during the long-term operation. Finally, for the helium flowing through the cold compressor, the contamination from grease evaporation is very small, which can satisfy the operation requirement.

5. References
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