**Thermal Analysis of Ceramic Conventional Ball Bearings**

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**Abstract**

**Objective:** To analyze the heat transfer in a ceramic conventional ball bearing and to study the heat dissipation, temperature profile, deformation and thermal stresses occurring in a bearing as a function of rotational speed.

**Methods/Statistical Analysis:** The Finite Element Method (FEM) was used to analyze the heat flow and other parameters in a bearing. Modeling of the system was done using SOLIDWORKS. The analysis was done to study the heat dissipation in the bearing. The model was imported in ANSYS by giving all the parameters. The temperature profile in the bearing was obtained by changing the properties. Structural analysis was performed and the deformation of the bearing at various points were calculated. **Findings:** It was observed that the temperature of the bearing increases with increase in the heat generation. The effects of temperature for different bearing speeds have been studied and it has been found that the rotational speeds have predominant effect on the temperature. The temperature of the bearing increases with the increase in speed. The maximum temperature is a function of heat generation. With the increase of the rotational speed the displacement increases which causes deformation and stresses. The rotational speed also has more effect on the stiffness.

**Application/Improvements:** Bearing is used to support the shaft. As an improvement the vibration characteristics can be analyzed and simulated.

**Keywords:** Ball Bearing, Deformation, Finite Element Method (FEM), Temperature Distribution, Thermal Analysis

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**1. Introduction**

The ball bearing is one of the main part of the mechanical component, and many joints exist in the ball bearing. When the surfaces are in contact, friction occurs. The friction causes a sudden heating of the balls and can have adverse effects. The increase of the temperature causes deformations and overheating of the coolant. The thermal heating causes deformation that subsequently affects the accuracy of the machine tool. The bearing allows the rotary motion. Figure 1 shows a ball bearing.

Anbazhagan et al. performed analysis of the crack in pressure vessel saddles using the Finite element method. Krishnamani et al. made thermal analysis of ceramic coated aluminum alloy piston using the Finite element method. Alfares et al. developed a mathematical model to study the axial preloading of ball bearings. Jedrzejewski et al. analyzed the axial forces in a bearing. Bao-min et al. studied the inner ring displacement based on elastic theory. Gao et al. simulated the shape, size and stress of ball bearings using finite element method. Jin et al. calculated the heat generation rate which is used in machine tools. Guo et al. made analysis on angular contact ball bearing. Wang et al. developed a method for calculating the contact subsurface stress of ceramic ball bearing. Yu et al. measured the temperature distribution in the bearing. In et al. simulated a deep groove ball bearing using ANSYS.

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2. Materials and Methods

Based on the literature review, the design and process parameters were selected. The steady state thermal solution was obtained. The static structural solutions were obtained. Figure 2 shows the flow chart for thermal and structural solutions.

2.1 Experimental Procedure

The model includes all the heat sources and thermal expansion in the system. Friction causes an increase in temperature of the bearing. If the heat generated is not removed the bearing may fail. To analyze the heat flow, a ball bearing has been modeled and analyzed using the finite element technique. The temperature is a function of heat dissipation and the speed.

The design model was created in SOLIDWORKS and was imported to ANSYS by giving the parameters and the steady state thermal analysis was done by giving the required conditions. The deformation of the bearing at various points were calculated.

2.1.1 Selection of the Bearing

A specific condition of the ball bearing was taken and the design calculation was done. The ball bearing was selected accordingly. A simply supported shaft, diameter 20mm, with a load of 10kN in the middle with the axial load of 3kN was taken. The speed of the shaft was taken as 1440rpm. Bearing was selected for 1000 hours of rotation. The radial and axial load factor selection are given in Table 1.

\[
P_r = 5 \text{kN} \\
P_a = 3 \text{kN}
\]

Life of the Bearing in millions of revolution.

\[
L_{10} = \frac{60 \times 1440 \times 1000}{10^6} = 86.4
\]

Equivalent load on the bearing is given by,

\[
P_e = X \cdot P_r + Y \cdot P_a
\]  (1)

From the above table \(e = 0.45\) by interpolation.

\[
P_{Pr} = \frac{3}{5} = 0.6 < e
\]

From the above table \(X = 1\) and \(Y = 0\)

Equivalent radial load, \(P_e = X \cdot P_r + Y \cdot P_a\)

\[
P_e = (1 \times 0.1 \times 5) + (0 \times 3)
\]

\[
P_e = 0.5 \text{kN}
\]

Basic dynamic load rating, \(C = P_e \left( L_{10} \right)^{\frac{1}{3}}\)  (2)

\[
C = 0.5 \left( 86.4 \right)^{\frac{1}{3}} = 11.05 \text{kN}
\]

By referring the bearing catalogue at 20 mm inner diameter and \(C = 11.05 \text{kN}\) the bearing was selected as SKF 6004.

Table 1. Radial and axial load factor selection

| \(\frac{P_a}{Co}\) | \(\frac{P_e}{Pr}\) ≤ \(e\) | \(\frac{P_e}{Pr}\) ≥ \(e\) |
|---|---|---|
| X | Y | X | Y |
| 0.021 | 0.21 | 1.0 | 0.0 | 0.56 | 2.15 |
| 0.110 | 0.30 | 1.0 | 0.0 | 0.56 | 1.45 |
| 0.560 | 0.44 | 1.0 | 0.0 | 0.56 | 1.00 |
2.1.2 Heat Generation in the Bearing

The major source of heat generation is the machining process and the friction between the balls and the races. The major portion of the heat is taken away by the coolant and the chips. In ball bearings heat is generated by three sources. First is the load related heat generation, second source is the viscous shear of lubricants between the solid bodies, known as viscous heat dissipation. The third source of heat is known as spin related heat generation. Considering this, analytical formulation for heat generation in a bearing was developed. The heat generated in a bearing is given as

\[ H_f = 1 \times 10^{-4} \cdot n \cdot M \]  

where, \( H_f \) is the heat generation due to friction in Watts, \( n \) is the rotational speed (rpm), \( M \) is the total frictional torque (N mm). Rotational speeds of 50, 100, 150, 200, 250, 300, 350, 400, 450, 500, 550 were taken and the total frictional torque as 100 N-mm. The internal heat generation can be calculated by using the formula, \( \text{Internal Heat Generation} = \frac{H_f}{V} \). The volume of the Ball bearing was calculated as 56.54 mm\(^3\). The values were tabulated and the internal heat generation was calculated. The internal heat generation for different speeds is shown in Table 2.

| Speed (rpm) | Heat Generation (W) | Volume of bearing (mm\(^3\)) | Internal Heat Generation (W/mm\(^3\)) |
|-------------|---------------------|------------------------------|-------------------------------------|
| 50          | 0.5253              | 56.54                        | 0.009258932                         |
| 100         | 1.047               | 56.54                        | 0.018517863                         |
| 150         | 1.5705              | 56.54                        | 0.027776795                         |
| 200         | 2.094               | 56.54                        | 0.037035727                         |
| 250         | 2.6175              | 56.54                        | 0.046294659                         |
| 300         | 3.141               | 56.54                        | 0.05555359                          |
| 350         | 3.6645              | 56.54                        | 0.064812522                         |
| 400         | 4.188               | 56.54                        | 0.074071454                         |
| 450         | 4.7115              | 56.54                        | 0.083330386                         |
| 500         | 5.235               | 56.54                        | 0.092589317                         |
| 550         | 5.7585              | 56.54                        | 0.101848249                         |

2.1.3 Modeling in SOLIDWORKS

The bearing consists of ceramic balls of mass 0.067kg. The outer and inner diameter are \( D = 42 \text{ mm} \), \( d = 20 \text{ mm} \) respectively, \( B = 12 \text{ mm} \), \( a_o = 15^\circ \) and \( Z = 9 \) balls. The bearing operates under dynamic load rating of \( C = 9.95 \text{kN} \) and static load rating of \( C_o = 5 \text{kN} \) and at a rotational speed of 50-550rpm.

Using these specifications, the ball bearing was modeled in SOLIDWORKS. The design process consists of four parts and one assembly. The four parts namely outer race, inner race, separator and cage were designed first individually and these parts were assembled thereafter. Figures 3-6 indicates the model of the outer race, inner race, cage, ball respectively and Figure 7 indicates the ball bearing assembly.

Figure 3. Outer race.

Figure 4. Inner race.

Figure 5. Cage.

Figure 6. Ball.
Thermal Analysis of Ceramic Conventional Ball Bearings

2.1.4 Simulation

The model made in SOLIDWORKS was imported in ANSYS. The node has three degrees of freedom; translation in the x, y, and z directions. The simulation was performed. The solid model was imported in ANSYS workbench. The steady state thermal analysis containing Engineering Data, Geometry, and Model was performed and the solution was obtained.

The properties of ceramic are given as follows. density = 3200 kgm$^{-3}$, coefficient of thermal expansion = $2.9 \times 10^{-6}$ $^\circ$C$^{-1}$, Young's modulus = 320E9 Pa, Poisson's ratio = 0.27.

The meshing of the model was then done. Figure 8 shows the imported data in ANSYS, Figure 9 shows the Ball Bearing geometry and Figure 10 shows the meshing of the assembly.

3. Results and Discussion

The simulation gave the required parameters which are temperature, total heat flux, total deformation, equivalent stress and equivalent elastic strain at various rotational speeds from 50-550 rpm. Figures 11-15 indicates the temperature profile, heat flux, total deformation, equivalent stress and equivalent strain respectively at 50 rpm. Figures 16-20 shows temperature profile, heat flux, total deformation, equivalent stress and equivalent strain respectively at 550 rpm.

The temperature, total heat flux, total deformation, equivalent stress and equivalent elastic strain values
were taken from the simulated models with respect to rotational speed and were tabulated for obtaining the heat generation rate in the ball bearing. The heat generation was calculated. The simulation results are shown in Table 3. Figure 21 shows the rotational speed (rpm) vs. Temperature plot for various range of speeds. Figure 22 represents the rotational speed (rpm) vs. total heat flux (W/mm²) plot for different speeds. Figure 23 indicates the rotational speed (rpm) vs. total deformation (mm) plot. Figures 24 and 25 shows the plot of rotational speed (rpm) vs. equivalent stress (MPa) and equivalent strain respectively.
Figure 18. Total deformation (550 rpm).

Figure 19. Equivalent stress (550 rpm).

Figure 20. Equivalent strain (550 rpm).

Figure 21. Rotational speed vs temperature.

Figure 22. Rotational speed vs total heat flux.

Table 3. Simulation results

| Rotational Speed (rpm) | Temperature (°C) | Total heat flux (W/ mm²) | Total deformation (mm) | Equivalent stress (MPa) | Equivalent elastic strain (mm/mm) |
|------------------------|------------------|--------------------------|-----------------------|------------------------|----------------------------------|
|                        | Min              | Max                      | Min                   | Max                    | Min                              | Max                          |
| 50                     | 40.977           | 42.995                   | 0.0054                | 0.1079                 | 0.0014                           | 0.0053                       | 1.1642 | 108.60 | 0.0001 | 0.0005 |
| 100                    | 51.477           | 55.426                   | 0.0106                | 0.2112                 | 0.0021                           | 0.0082                       | 1.8692 | 169.94 | 0.0001 | 0.0009 |
| 150                    | 62.216           | 68.138                   | 0.0159                | 0.3168                 | 0.0029                           | 0.0113                       | 2.5995 | 232.67 | 0.0001 | 0.0012 |
| 200                    | 74.147           | 82.264                   | 0.0219                | 0.4341                 | 0.0038                           | 0.0147                       | 3.4147 | 302.38 | 0.0001 | 0.0015 |
| 250                    | 84.886           | 94.977                   | 0.0074                | 0.5397                 | 0.0046                           | 0.0177                       | 4.1499 | 365.11 | 0.0002 | 0.0019 |
| 300                    | 95.624           | 107.69                   | 0.0088                | 0.6453                 | 0.0054                           | 0.0207                       | 4.8859 | 427.84 | 0.0002 | 0.0022 |
| 350                    | 106.36           | 120.40                   | 0.0103                | 0.7510                 | 0.0062                           | 0.0238                       | 5.6223 | 490.57 | 0.0002 | 0.0025 |
| 400                    | 118.29           | 134.53                   | 0.0121                | 0.8683                 | 0.0071                           | 0.0271                       | 6.4410 | 560.28 | 0.0003 | 0.0028 |
| 450                    | 129.39           | 147.66                   | 0.0134                | 0.9774                 | 0.0079                           | 0.0303                       | 7.1979 | 621.10 | 0.0003 | 0.0032 |
| 500                    | 139.77           | 159.95                   | 0.0148                | 1.0812                 | 0.0088                           | 0.0332                       | 7.9115 | 684.79 | 0.0004 | 0.0035 |
| 550                    | 150.51           | 172.67                   | 0.0162                | 1.1851                 | 0.0094                           | 0.0362                       | 8.6251 | 748.48 | 0.0004 | 0.0038 |
In the numerical analysis temperature distribution was measured for a series of rotational speeds. The heat generation is due to the torque developed. Here two types of torques were considered, one is the load torque and other is the viscous torque. The heat generation value was inputted with the required conditions and the temperature profile; total heat flux of the entire model was measured. The obtained temperature from the thermal analysis was inputted by updating the conditions. The deformation of the model and the maximum stress distribution at the contact were measured.

At higher speeds the dynamic response was significant and the thermal effects have to be considered. The thermal load affects the stiffness. The heat generation in the ball bearing is a major cause of thermal expansion.

4. Conclusion

The heat generation rate, temperature profile, deformation and thermal stress of the bearing were measured. The simulation was performed, and the temperature increases with heat generation. The effects of temperature for different bearing speeds are studied and the rotational speeds have major effect on the temperature which tends to increase with the increase of speeds. If the groove diameter is reduced, deformation and stresses can be minimized. From the results, deformation is within the limit. This study is used to analyze the failure of bearing.

5. References

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6. Nomenclature

- **C** – Dynamic load
- **d** – Diameter of the ball
- **H_f** – Heat generated due to friction
- **M** – Total frictional torque
- **n** – Rotational speed
- **P_r** – Radial load
- **P_a** – Axial load
- **P_e** – Equivalent load
- **X** – Load factor - radial
- **Y** – Load factor - axial