Thermal and Dynamic Analysis of Bearing Assembly for Small Size Steam Turbine

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Abstract--General solution procedure is given to enhance the loading capacity of 10 MW Siemens200 steam turbine(SST200) by adopting the 110/180 mm size bearing in 90/160 mm size bearing casing. First 90/160 mm bearing with casing is modeled in Pro-E wildfire 4.0 and analyzed in ANSYS workbench 12.0 for mechanical and thermal loads and benchmarked. Then modification is done in 90/160 mm bearing casing to accommodate 110/180 mm bearing. At last modified bearing casing is analyzed with 110/180 mm bearing and found suitable for use.

Index terms—Mechanical load, Thermal load, Turbine Bearing

I. INTRODUCTION

 Tilting pad type thrust bearings are used in a wide variety of rotating machinery where significant thrust loads must be accommodated [1]. Thrust load is transmitted from rotor to stator through hydrodynamic oil films that generate between a rotating collar on shaft and pads in bearing [2]. The lubricant used was ISO VG 46 turbine oil [3]. Also modal analysis is carried out and heat load is found out.

II. STATIC ANALYSIS

First of all the part drawings of tilting pad, thrust pad, radial bearing, axial bearing, bearing casing are provided. The 3-D solid models are prepared using Pro-E Wildfire 4.0 [4] of various parts. The forces are identified and the static analysis is carried out using Ansys Workbench 12.0.

Figure 1.1 and 1.2 shows modified upper and lower bearing casing. Also table 1.1 and 1.2 shows dimensions of bearing casing.

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Original Dimension, mm</th>
<th>Modified Dimension, mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>175</td>
<td>240</td>
</tr>
<tr>
<td>2</td>
<td>244</td>
<td>288</td>
</tr>
<tr>
<td>3</td>
<td>220</td>
<td>288</td>
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<td>4</td>
<td>294</td>
<td>302</td>
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<tr>
<td>5</td>
<td>34</td>
<td>44</td>
</tr>
<tr>
<td>6</td>
<td>168</td>
<td>171</td>
</tr>
</tbody>
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Table 1.1: Dimensions of Upper Bearing Casing of 90/160 mm Bearing

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<td>6</td>
<td>168</td>
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Table 1.2: Dimensions of Lower Bearing Casing of 90/160 mm Bearing

Figure1.3 and 1.4 shows the assembly drawings of 90/160 mm size bearing with casing and 110/180 mm size bearing with 90/160 mm bearing casing.
The stresses generated on axial pad and bearing casing due to axial and radial force.

Radial Load = 9751 N
Thrust Load = 39687 N

The von-Mises stress for 90/160 mm bearing is 51.056 MPa and for 110/180 mm bearing is 45.478 MPa. The stress values are coming nearer to same. So, 110/180 mm bearing is suitable in 90/160 mm bearing casing.

III. THERMAL ANALYSIS

Heat load calculation, oil outlet temperature and static analysis with temperature effect are carried out. First 90/160 mm bearing with casing is analyzed for benchmarked and then analyzed for 110/180 mm bearing with 90/160 mm bearing casing.

1) Heat Load by Free Convection

Heat generated at radial and axial of 90/160 mm size bearing is partly carried away by lubricating oil circulated in the bearing and partly by natural convection to atmosphere. Surface temperature of bearing casing $T_s$ is $67^\circ$C, Free stream temperature $T_1$ is $33^\circ$C, Diameter of bearing $D = [380-(240+225)/2] = 147.5 mm$, Characteristic length of bearing $L = 450 mm$, Properties of air at mean temperature $T_m$ is $(67+33)/2 = 50^\circ$C.
So, Rayleigh No is given by,
\[ R_u = \frac{g \cdot \beta \cdot (T_r - T_m) \cdot D^3}{V^2} \cdot P_r = 7.4090e8 \]
Nusselt No is given by, [6]
\[ N_u = [0.6 + \frac{0.387 \cdot R_{1/6}^{1/6}}{1+(\frac{P_r}{P_r})^{9/16}}]^{2} = 105.6332 \]
Therefore convective heat transfer coefficient is given by,
\[ h_{conv} = 6.4202 \frac{w}{m^2 k} \]
Heat loss is given by,
\[ Q = hA(T_T - T_m) = 148.2386w \]

2) Heat Carried away by Oil
The lubricating oil enters in the bearing, it absorbs heat. This high temperature oil is recirculated through oil cooler where it is cooled to predefined temperature. This cooled oil is again recirculate in the bearing.

- Volume flow rate of oil: 122 lit/min
- Specific heat of oil: 1976.1696 J/kg°C
- Density of oil: 861 kg/m³
- Inlet temperature of oil: 49°C
- Outlet temperature of oil: 68°C

Hence heat carried away by oil,
\[ Q = mC_p \Delta T = 65733.9223w \]

3) Heat load by Force Convection
Heat is transferred by force convection at axial and radial bearing. [7]

- Surface temperature of rotor: 80°C
- Free steam temperature of air: 49°C
- Properties of oil at mean temperature \( T_m \):
  - \( (80+49)/2=64.5°C \)
- Prandtl No is given by, [8]
  \[ \mu C_p \cdot P_r = 227.3546 \]
For Radial Bearing, Reynold No,
\[ R_{re} = \frac{DvD_{sh}}{\mu} = 6.32014e5 \]
Nusselt No. [9]
\[ N_u = \frac{(\frac{f}{2}) \cdot R_{e} \cdot P_r}{1.07 + 12.7 \cdot (\frac{f}{2})^{3/2} \cdot [P_r^{2/3} - 1]} = 11553.0332 \]
Hence, Convective heat transfer coefficient
\[ N_u = \frac{h \cdot D_e}{K} \]
\[ \Rightarrow h_1 = 1993.7613 \frac{W}{m^2 K} \]
Same procedure is done three times so,
\[ h_2 = 2336.7974 \frac{W}{m^2 K} \]
\[ h_3 = 1993.7613 \frac{W}{m^2 K} \]
\[ h_4 = 3268.7550 \frac{W}{m^2 K} \]
Heat loss is given by,
\[ Q_{radial} = [hA + h_1A_1 + h_2A_2 + h_3A_3] \cdot \Delta T = 2541.5120w \]
Similarly procedure is carried out for axial bearing. Hence heat loss is given by,
\[ Q_{axial} = 2992.2787W \]
Total heat loss,
\[ Q_{total} = Q_{axial} + Q_{radial} = 5533.7907w \]

4) Heat Generation by Friction
When rotor rotates in bearing at that time due to friction heat is generated.
- Axial Force = 47712.5 N
- Radial Force = 9751.6 N
- Radius = 0.045 m
- Speed = 12000 rpm
- Coefficient of friction = 0.0176
Heat load by radial bearing,
\[ Q_{radial} = T \cdot \omega = 9772.7423w \]
Same for axial bearing,
\[ Q_{axial} = 47486.1797w \]
Hence total heat load is generated due to frictional effect,
\[ Q_{total} = Q_{radial} + Q_{axial} = 57258.922w \]
Balancing heat loss,
\[ Q_{frictional} + Q_{force} = Q_{oil} + Q_{free} \]
\[ \Rightarrow 62792.7127w = 65882.1609w \]

5) Oil Outlet Temperature of 90/160 mm bearing
\[ Q = mC_p \Delta T \]
\[ \therefore 44000 = \frac{102.5e-3 \cdot 861 \cdot 1976.1696 \cdot (T_o - 49)}{60} \]
\[ \Rightarrow T_o = 64.14°C \]
Heat Load from Turbine Casing body to bearing body by conduction,

\[ Q = KA \frac{dt}{dx} = 934.14w \]

Hence, outlet temperature is given by,

\[ \Rightarrow T_o = 64.46^\circ C = 65^\circ C \]

5.1) Thermal Analysis of 90/160 mm bearing

Material for Radial bearing: 42CrMoS4V

- Density: 7.80 gm/cm$^3$
- Young modulus of Elasticity: 2.1e5N/mm$^2$
- Coefficient of thermal expansion: 11.7e-6 K$^{-1}$
- Yield strength: 500MPa
- Thermal conductivity: 36W/mK

For axial bearing: 16MnCr5

- Density: 7.85 gm/cm$^3$
- Young modulus of Elasticity: 2.1e5N/mm$^2$
- Coefficient of thermal expansion: 11.7e-6 K$^{-1}$
- Yield strength: 500MPa
- Thermal conductivity: 41W/mK

For casing: Cast Iron

- Density: 7.20 gm/cm$^3$
- Young modulus of Elasticity: 240N/mm$^2$
- Coefficient of thermal expansion: 1.1e-5 K$^{-1}$
- Yield strength: 250MPa
- Thermal conductivity: 52W/mK

Inlet, outlet and temperature at fluid film interface is 49°C, 64°C and 92°C are applied. Hence, temperature profile is generated in Ansys Workbench 12.0[10] as shown below.

Figure 2.1 shows that the temperature at all the sections of 90/160 mm bearing is less than 80°C except at fluid film interface where temperature is about 92°C. This indicates that these temperatures are within limit.

Now, applied axial and radial forces on pads. Also friction force is applied on pad.

For axial bearing,

\[ F_r = \mu r o = 11.1696N \]

And for radial bearing,

\[ F_r = 10.9705N \]

Stress analysis is shown in figure 2.2. Maximum stress shows at axial pad’s contact surface.

Fig. 2.1: Temperature at different section of 90/160 mm bearing

Fig. 2.2: Stress value of 90/160 mm bearing with casing
6) Oil Outlet Temperature of 110/180 mm bearing
In this section same procedure is carried out which is done in section – 5.

\[ Q = mC_p\Delta T \]
\[ \Rightarrow T_o = 68.08^\circ C \]

Also, heat Load from Turbine Casing body to bearing body by conduction is 934.14 w. Hence, outlet oil temperature is,
\[ \Rightarrow T_o = 68.35^\circ C \pm 69^\circ C \]

6.1) Thermal Analysis of 110/180 mm bearing with 90/160 mm bearing casing
Same procedure is carried out as per section 5.1. Hence, temperature profile shows in below figure.

![Temperature Profile](image)

Also, Stress analysis is shown in figure 2.4. Maximum stress shows at axial pad's contact surface.

![Stress Analysis](image)

After doing all analysis results are shown below. Maximum stress generated at axial pad's contact surface in both case.

<table>
<thead>
<tr>
<th></th>
<th>90/160 mm bearing with casing</th>
<th>110/180 mm bearing with 90/160 mm bearing casing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>65°C</td>
<td>69°C</td>
</tr>
<tr>
<td>von-Mises Stress</td>
<td>80.467MPa</td>
<td>46.898MPa</td>
</tr>
</tbody>
</table>

IV. DYNAMIC ANALYSIS

Basically mode shapes and frequency is finding out in modal analysis. From that we can decide the nature of rotor during working condition. [11] During working condition, some unbalance weight will generate and it will act at any distance which can be found as per ISO 1940/1. [12]

![Modal Analysis](image)

Unbalance weight is given by,
\[ W = \frac{U_{per}}{r} = 12.79 \text{ gm} \]
Now, in this case unbalance weight have kept at centre of rotor. So, due to unbalance weight new mode shape of rotor is shown in Figure 3.2.

Due to unbalance weight natural frequency as well as displacement is decreasing in modal analysis.

<table>
<thead>
<tr>
<th>Actual Rotor</th>
<th>After adding unbalance weight</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency</td>
<td>182.9 Hz</td>
</tr>
<tr>
<td></td>
<td>181.5 Hz</td>
</tr>
<tr>
<td>Max. Displacement</td>
<td>42.257 mm</td>
</tr>
</tbody>
</table>

V. CONCLUSION

In order to enhance the loading capacity of SST 200, it is possible to use 110/180 mm bearing in modified 90/160 mm bearing casing. The earlier loading capacity of SST 200 is 4 MW. After the modification, loading capacity should 8 MW which should be increased by 200%

VI. REFERENCES

[1] Yong Joo Cho Kyung Bo Bang, Jeong Hun Kim. Comparison of power loss and pad temperature for leading edge groove tilting pad journal bearings and conventional tilting pad bearings. Department of Mechanical Engineering, Pusan National University, 2009
[4] Tickoo Sham, Pro - e wildfire 4.0
[10] Ansys thermal analysis guide 12.0