STEAM TURBINE VIBRATION

Newly machined bearing, new coupling, and redesigned gearbox did not help in eliminating running speed vibration.

A large US gas production company reported continuous vibration problems with a steam turbine driven fan. The steam turbine fan was of a unique design and was one in only a dozen made in the world. During a company internal review it was decided that to ensure “better reliability” a new gearbox with different gear ratio should be installed and the coupling should be exchanged. The intent was to slow down the turbine with no effect to the fan speed. At the same time, a new foundation and inertia (mass) concrete block was fabricated and poured to accommodate the new unit assembly. At the first start the unit, which should have operated up to about 5300 RPM, exhibited high instability and unacceptable vibration above 4000 – 4200 RPM. Numerous operational parameter measurements and procedures were attempted without success. The company then made the conscious decision to run the unit below what they dubbed “critical speed” and seek professional machinery vibration help from Cascade MVS.

CASCADE MVS
DICKINSON TEXAS (281) 482-2727

ROTOR-DYNAMIC PROGRAM

Cascade MVS uses the Finite Element-based, powerful, and industry proven DyRoBeS program. The program incorporates fully-integrated rotor, bearing, and gear load modules with excellent visual presentation and ease of use. The program was developed by industry leader Wen Jeng Chen, Ph.D., P.E., and is supported by a core group of industry leaders. It represents the best available R-D software on the market.

“AFER MAJOR FAILURE, DUE TO THE LOSS OF LUBRICATION OIL, IDENTICAL REPLACEMENT TURBINE WAS LOCATED AT AFTERMARKET SUPPLIER WITH ALL SPHERE PARTS”
After initial contact and teleconference with the customer, the following steps were suggested and accepted:

1. Review all records since original Turbine failure
2. Supply **Cascade MVS** technical drawings on:
   2.1 Steam turbine rotor
   2.2 Steam turbine bearings
   2.3 Coupling
   2.4 Gearbox

From the above information **Cascade MVS** was able to build a rotor-dynamic model consisting of the steam turbine rotor, coupling, and gearbox pinion. Appropriate bearing damping and stiffness were calculated and applied for each bearing location.

**Steps taken:**

1. Engineering drawings of new replacement bearing were made
2. Measurements and obtained clearances were used to build rotor model

<table>
<thead>
<tr>
<th>Description</th>
<th>Rotor Weight – Lbm</th>
<th>Polar Inertia –Lbm-In^2</th>
<th>ST Bearing span - in</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotor Assembly</td>
<td>1129.00</td>
<td>33609.1</td>
<td>≈ 35.70</td>
</tr>
</tbody>
</table>

Figure 1 – Pinion – Coupling – Steam Turbine rotor
Modeling goal:

a. Determine lateral dynamic behavior of the combined rotor with existing bearings.

b. Design optimized pressure dam bearing to ensure minimum vibration, maximum possible stability, and sufficient damping, and move any critical out of the rotor running speed range.

Shaft Journal Dimensions:

<table>
<thead>
<tr>
<th></th>
<th>Steam End</th>
<th>Exhaust End</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bearing Bore</td>
<td>2.620 +0.000/0.001</td>
<td>2.624 +0.001/0.00</td>
</tr>
<tr>
<td></td>
<td>-3.245 +0.000/0.001</td>
<td>3.254 +0.001/0.00</td>
</tr>
</tbody>
</table>

Bearing Running Clearances:

<table>
<thead>
<tr>
<th></th>
<th>Steam End</th>
<th>Exhaust End</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>-0.004” to 0.006”</td>
<td>0.009” to 0.011”</td>
</tr>
</tbody>
</table>

Oil in use – ISO Grade 100 @ 120°F inlet temperature

The bearings were of “Oil Dam” design. After the shop measurements confirmed the instruction book sizes, diameters, and clearances, bearing stability calculations were applied to the rotor model.

Typical calculated and produced plot presentations

The bearing modeling and calculations indicated possible instability.
The resulting rotor model with existing bearing clearances and geometry were as follows:

![Critical Speed Mode Shape](image)

The first bending model was calculated to 4210 RPM, matching customer described “critical speed.”

The next step in rotor modeling was to attempt to stabilize the bearings as well as to achieve unit operation through full operational speed range up to 5500 RPM.

**Step 1.** Optimize both steam turbine bearings by either changing the geometry, type, and or clearances. Per customer instructions, the type of oil used and inlet temperature could not be changed. We opted, as step one, to use existing the “Oil Dam” bearing geometry with the following optimized measurements and characteristics:

**Steam End Bearing**

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shaft Diameter</td>
<td>Ø2.620”</td>
</tr>
<tr>
<td>Calculated Dia clearance</td>
<td>0.0045” (0.00168”/Shaft Ø)</td>
</tr>
<tr>
<td>Pocket arc/depth</td>
<td>145° / 0.007”</td>
</tr>
<tr>
<td>Pocket width</td>
<td>1.00”</td>
</tr>
<tr>
<td>Bearing Length (L)</td>
<td>2.00”</td>
</tr>
</tbody>
</table>
The resulting rotor model with optimized bearing clearances and geometry were as follows:

Shaft Diameter = Ø3.245"
Calculated Dia clearance = 0.0055" (0.00168"/Shaft Ø)
Pocket arc/depth = 145° / 0.008"
Pocket width = 1.248"
Bearing Length (L) = 2.50"

Optimized “Oil Dam” bearings satisfied needed stability producing adequate oil film pressure, i.e. stiffness and damping.

At the same time oil film pressure, shaft film thickness, and operating temperatures were inside acceptable levels.

The result was that the First Critical speed was moved above the minimum required 5500 RPM and allowed full speed range operation as requested by the customer.