Subsynchronous Vibration of an Auxiliary Turbine

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Introduction and Background

This paper discusses an auxiliary turbine subsynchronous vibration problem which extended over several years at one of Ontario Hydro's thermal generating station.

The auxiliary turbine is a multi-stage condensing steam type and is used to drive a boiler feed pump via a flexible gear coupling. The turbine rotor is supported by and located in the frame assembly by two cylindrical sleeve bearings, one in the low pressure (L.P.) bearing bracket attached to the frame assembly, and one in the high pressure (H.P.) bearing bracket located in the front standard (Figure 1). Normal operating speed of the unit is at 4250 rpm, and the turbine can run either on main steam or extraction steam.

Since 1984, the auxiliary turbine and the feed pump suffered random subsynchronous vibration problem at high loads, and at speeds approaching normal running speed. The predominant component of these vibrations is "locked in" at 1/2*rpm and the overall vibration levels of up to 150 microns (Pk-Pk) to 200 microns (Pk-Pk) have been observed at H.P. and L.P. bearing pedestals (Figure 2).

In the past, The plant operators were able to reduce the subsynchronous vibration amplitude by dis-engaging and re-engaging the gear coupling. This method appeared to be too inconsistent and the station finally decided to carry out full investigations to resolve the problem. It is important to note that when the subsynchronous vibration was suppressed, the vibration response exhibited peaks at operating speed frequency and its harmonics. In this case, the subsynchronous frequency was present but at much lower amplitude and the operator could run the unit at full load.

During this period, many problems were identified and subsequently rectified, however, the vibration problem still existed. Among those, the most noticeable discoveries were:
Loose interstage diaphragms causing the packing seals to lift and contact the rotor; Boiler Feed Pump/Turbine concrete base was cracked about 4 inches below the top of concrete around the sole plate; Mis-matched gear components; Crack formation on the collar of the flexible gear coupling (near the turbine end); Turbine front standard axial keys were dry and appeared out of position to each other.

Measurements

Detailed noise and vibration measurements were carried out for this unit. In each case, the vibration measurements were taken at the L.P., H.P. and the pump inboard and outboard bearing pedestals. Filtered noise measurements were also taken simultaneously at a distance of 10 inches from the pump and turbine casings at discrete locations extending from the pump inboard bearing to the turbine H.P. bearing (see table 1). Figures 3 and 4 represent a typical noise measurements taken by a real time analyzer. It was noticed in all our measurements that a distinct tone was detectable around the unit at one and two times the rpm when the unit was operating at full load. The noise radiated due to 1*rpm appeared to have maximum amplitude close to the south side of the coupling and the noise radiated at 2*rpm had a maximum level in the close proximity of the third and fourth stage turbine.

Vibration measurements revealed that when the unit was exhibiting the subsynchronous vibration problem, the axial pump inboard and outboard vibration amplitudes were generally low. However, in the case where the subsynchronous vibration was suppressed, the measured pump inboard and outboard axial vibration were excessive with the phase difference of approximately 180 degrees.

Additional phase measurements carried out at full load for the unit using hand held "shaft sticks" or pedestal measurements were not conclusive in identifying the direction of the whirl (either backward or forward precession).

Possible Excitation Phenomena

The subsynchronous vibration problem experienced on auxiliary turbine was generated by one, or more, of the following mechanisms: 1) Hydrodynamic fluid film bearing (Oil Whip); 2) Excessive clearances in the bearings; 3) Incorrect pinch on the bearings; 4) poor bearing foundation tie-down; 5) Defective coupling; 6) Stuck HP front standard axial key; 7) Sub-harmonic whirl instability induced by non-linearity (Mathieu-Hill-Meissner); 8) Aerodynamic induced whirl.

Discussions

The noise and vibration measurements confirmed the existence of coupling unbalance. This occurred due to the mis-match of gear components causing gear mesh position error. In addition, the non-uniform nature of axial vibration and its high magnitude also supported the existence of rotor misalignment. The high noise level detected at close proximity to the turbine casing also indicated the possible misalignment of the pump-turbine rotor.

The subsynchronous vibration was generated above the first critical speed (2600 rpm) and the onset of its formation was too sudden. As the rotor was brought up to speed, we saw no evidence of subsynchronous peak. Only when the shaft speed was near 3900 rpm did this subsynchronous peak occurred. As well, we found no evidence (based on the critical speed data) to support that the critical speed of the system was reduced significantly rotor speed approached 3900 rpm. therefore, it was assumed that the oil whip was not the cause of this instability.

The concept of support stiffness asymmetry was reviewed after the report of pump pedestal cracking (see above). For this phenomenon to occur, at least two distinctly different critical speeds should have been present reflecting the different stiffness in two different directions. Such a response was not observed and consequently this argument was not supported as the cause of this instability.

The sharp increase in the half frequency component could have also been generated due to aerodynamic steam forces. This was however not supported due to the fact that the subsynchronous vibration was phase-locked and as well the frequency was exactly at 1/2 of the running speed.

Based on our analyses, we concluded that the auxiliary turbine was suffering from two types of instabilities. The first type was the subsynchronous vibration and was generated as a result of "Free Mathieu Effect" and the second type was generated as a result of "Gyroscopic Induced Whirl". The two types of whirl had similar responses in many aspects. The factors that influenced the type of instability were the change of radial load created by turbine-pump rotor alignment and the initial degree of rotor misalignment. The subsynchronous vibration was caused by non-linearity effects. In this case the rotor was displaced sufficiently to hit the stationary parts. This generated a sufficient energy for the rotor to sustain its continuous motion. The rotor displacement was as a result of: a) dynamic shaft bending; b) excessive misalignment of turbine-pump rotor; c) inadequate bearing stiffness.

Second type of vibration normally occurred after the unit has been shut down and re-started and the coupling has been dis-engaged and re-engaged. Because the gear coupling comprised unmatched gear components and the turbine-pump rotor had some misalignment, it was quite possible depending on the relative radial positions of the coupling gears upon re-engagement, the required coupling flexibility might not work. In the event of a locked engagement, an additional radial load was generated which was added to the bearing stiffnesses. In this case, the rotor was constrained to be bowed and the Gyroscopic effect would result in an instability. In this mode, light internal rub did take place and many resonances were excited.

To alleviate the vibration problem, an appropriate course of actions were recommended to the station which will be highlighted during this presentation.

Acknowledgement

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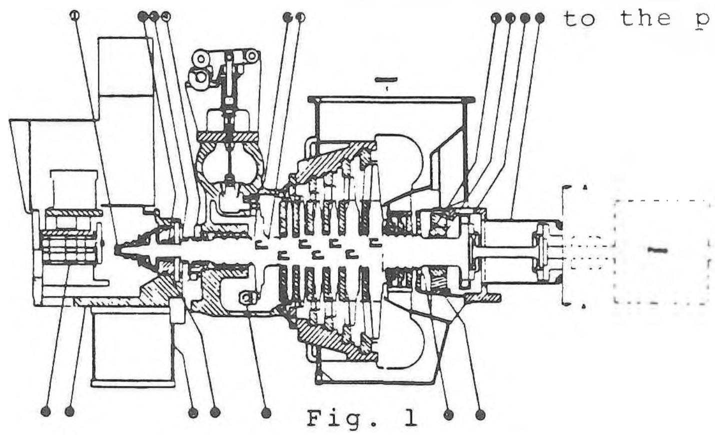

Fig. 1

Fig. 2

Position	Approximate Location of Measurement
A	Pump Inboard Bearing
B	Coupling
C	LP Bearing (Perpendicular to Axis of Rotation)
D	LP Bearing (Parallel to Axis of Rotation)
E	Turbine Casing (Close to Coupling)
F	Turbine 7th Stage
G	Turbine 6th Stage
H	Turbine 5th Stage
I	Turbine 4th Stage
J	Turbine 3rd Stage
K	Turbine 2nd Stage
L	Turbine 1st Stage
M	HP Bearing

TABLE 1

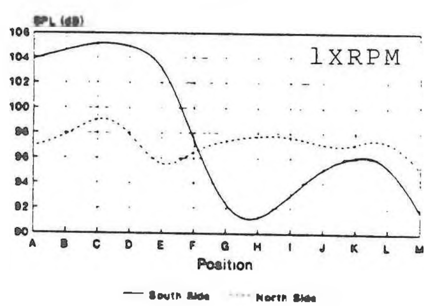

Fig. 3

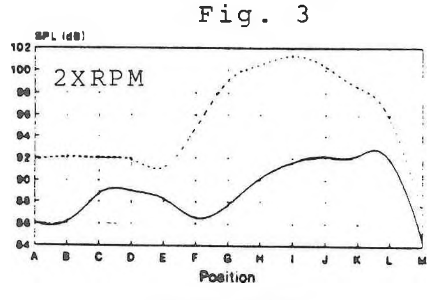

Fig. 4