Analysis of Muara Tawar CCPP Block 1 Steam Turbine Vibration

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Abstract. Condition monitoring of steam turbine enables early detection of faults and avoid unexpected breakdowns. Vibration analysis has gained much attention in the field of condition monitoring because of its accuracy in detecting faults and its ability for proper diagnosis of the faults. Muara Tawar CCPP block 1 has a steam turbine generator with a capacity of 215 MW. Steam turbine roll after inspection shows high vibration occurs on bearing 1. This study aims to find the cause of this high vibration and provide recommendations to avoid such symptoms. The vibration data used in the analysis are obtained from vibration analyzer tools associated with vibration monitoring systems. It is found that the amplitude of vibration at rotational speed frequency (1X) increased and phase angle changes at a constant speed. The vibration level at the coast down is higher than the observed vibration level at run-up. There is a shift in the position of critical speed which indicates a change of rotor stiffness. It is concluded that the high vibration is likely by a thermal bow on the rotor. The thermal rotor bow is caused by a light partial rubbing between rotor and labyrinth seal on the stator.

Keywords: steam turbine vibration, thermal bow, rubbing, vibration analysis.

1. Introduction

The steam turbine is one kind of rotating machinery and the main equipment of the combined cycle power plant (CCPP). Steam turbine vibrates when it is operating. The forces within the steam turbine cause vibration at the rotor and bearing. The forces are the result of rotational and frictional forces. It generates a vibration signal which contains the working state information of mechanical equipment. The vibration of the steam turbines has many disadvantages, which result in serious consequences. Condition monitoring enables early detection of faults and avoidance of unexpected breakdowns. Vibration analysis has gained much attention in the field of condition monitoring because of its accuracy in detecting faults and its ability for proper diagnosis of the faults [1].
Condition monitoring on rotating machines using vibration sensors in the shaft gives better results than the sensors in bearings [2]. Nowadays the sensor used for the identification of vibrations is permanently installed by the manufacturer [3]. The source of vibration can be identified by analyzing the vibration signal using several methods. The main symptom of unbalance is the emergence of 1X rotational speed frequency (1X) vibration in the spectrum [4]. Misalignment is identified by looking at the spectrum, orbit, and shaft centerline. On the spectrum appears vibration at 1X, 2X, orbit shaped like "banana" or "number eight", and its abnormal centerline shaft [5]. The majority of the misalignment forces induce higher harmonics, the 3X frequency component is more prominent. The misalignment (parallel and angular both) in the rotor system reveals strong levels of backward whirling (negative) frequencies in the full spectrum plot [6].

Rotor bow is identified by the emergence of 1X rotational speed frequency vibration and an indication of the movement of a circular phase angle or resembling a spiral [4]. When rotor rub occurs, there is a change in the vibration of 1X in the polar plot[4]. Rubbing causes changes in orbit, elliptical orbits will enlarge or even change shape and change in vibration 1X, 2X [7]. Rubbing also can be identified using the bode plot, at a constant speed there is an increase in vibration amplitude and phase change [8]. Rubbing causes the thermal bow on the rotor can be identified by the appearance of unstable spiral vibration on a polar plot [9]. Using bode plot rubbing can be identified by an increase in the vibration amplitude and phase change and greater amplitude at coast down than at run-up [10].

Muara Tawar CCPP block 1 has a steam turbine generator with a capacity of 215 MW. Steam turbine roll after inspection shows high vibration on bearing 1. Several times of start-ups show the same indication. Vibration increases at a constant speed cause the unit needs to be shut-down while vibration before the overhaul tends to be below the allowable limit. This study aims to find the cause of high vibration bearing 1 and provides recommendations to avoid such symptoms.

2. Fault Identification

The trend in operator station (OS) only shows the overall vibration trend so that the diagnostic system needs to be obtained more informative characteristics. Each of the steam turbine bearings is monitored by a displacement transducer mounted on the bearing case in the (XY) plane perpendicular to the steam turbine rotor axis to observe the axial radial movement. The XY pair of non-contact displacement probes are mounted on a 45-degree left (X-probe) and 45-degree right (Y-probe). The steam turbine bearing layout is shown in Figure 1. The rotor rotates clockwise when viewed from driver to driven.

![Figure 1. Steam turbine bearing layout and transducers mounting.](image)

The vibration trends are grouped into four types of start-up conditions as shown in Figure 2. This figure gives information that time ranges from start up to steam turbine shutdown become longer, this indicates the cause of the vibration instability is reduced.

It is necessary to analyze the vibration plots during the constant speed conditions at points 1, 2, 3, 4, 5, 6, and 7 shown in Figure 2. This analysis is used to know the cause of vibration instability.
Figure 3 shows the spectrum plot at constant speed in three different conditions. Figure 3a is the spectrum at 1500 rpm, Figure 3b is the spectrum at 3000 rpm and Figure 3c is the spectrum under load conditions. The three spectrum plots shown in Figure 3, all show high amplitude at 1X despite being taken under different conditions.

![Figure 3. Spectrum comparison at constant speed in three different conditions.](image)

Figure 4 shows a polar plot at operation speed of 1500 rpm and 3000 rpm. It can be seen from this figure that the phase angle shifts in the radial and circular direction, but dominant in the radial direction. Vibration instability that occurs at a constant speed of 1500 rpm, only a slight phase shift occurs as shown in Figure 4a. The phase angle shift from point 1 to 2 is about 10°. At constant speed 3000 rpm as shown in Figure 4b, there is a reversal in the polar direction, but then move back out as the amplitude of 1X increases. The polar plot analysis of loaded conditions is distinguished between the conditions in
which the vibrations are unstable as shown in Figure 5a and the conditions in which the vibrations are stable as shown in Figure 5b. The location of the phase angle in both conditions is in the same quadrant. There is still movement of the dominant phase angle towards the radial while for the stable loaded conditions, phase angles tend to remain constant. Phase at constant speed 1500 rpm and 3000 rpm, when compared with loads conditions have the opposite direction.

![Figure 4. Polar plot at 1500 rpm and 3000 rpm.](image)

![Figure 5. Polar plot at constant speed and load conditions.](image)
Figure 6. Bode plot start up condition up to 1500 rpm and 3000 rpm

![Bode plot 0 rpm - load](image)

Figure 7. Bode plot start up condition up to loaded.

The amplitude change and phase angle to speed of the turbine (rpm) during run up, idle up to coast down is shown with bode plot of Figure 6. Bode diagram up to 1500 rpm as shown in Figure a, 3000 rpm as shown in Figure b and load as shown in figure 7. At coast down at the vibration level exceeds the vibration level at run up on all startup conditions. At coast down from start up 1500 rpm appears peak and accompanied by phase change in range 800-1000 rpm and 1200 - 1400 rpm. Coast down from start up 3000 rpm appears peak at range 1500 - 2000 rpm accompanied with phase change. There are two adjacent peaks and tend to be one peak. Coast down from load conditions, vibration go down first though, then increase in two peak critical speed. The increase of vibration amplitudes at coast down is not as high as vibration amplitude in bode plot 0 - 3000 rpm.

3. Data Analysis

Spectrum analysis confirms that a 1X rotational speed frequency increase when vibration trends increase. This happens in all startups conditions. An increase in the 1X rotational speed frequency implies that either the centrifugal force increased from the time of the event or the dynamic stiffness was reduced, or that both happened at the same time. Vibration at 1X rotational speed frequency is increasing at constant speed suggesting that the source of vibration is not unbalanced. This situation suggests that the source of high vibration is not unbalanced. There are other factors causing the 1X rotational speed frequency component. The polar plot analysis of all start-up conditions indicates a slight phase angle change. This phase angle change indicates that there is a change in heavy spot around the rotor. These symptoms confirm that it is not a pure unbalance that causes high vibrations in the rotor.

Generally, faults can cause changes in unbalance condition and heavy spot around the rotor is due to the occurrence of rotor bow. The cause of the rotor bow is the uneven distribution of heat. Changes in bow positions caused by heat distribution changes. The cause of thermal rotor bows either from rubbing between rotor and labyrinth seal to stator causing uneven heat around the rotor or the emergence of hot spots occurring within the journal bearing due to the difference in the viscous shearing of the lubricant in the journal bearing. This second possibility is usually caused by rotor overhung, bearing geometry and lubricant viscosity changed, but no conditions like this in a steam turbine.

Bode plot shows the condition where the vibration level at coast down is higher than at run up, either at start up to 1500 rpm and start up to 3000 rpm. There is influence of critical speed which marked with emergence of two peaks along with phase change during coast down, but amplitude is much higher than at start up. Change in critical speed at coast down confirms that thermal bow occurs in the rotor caused
by rubbing between rotor and labyrinth seal on stator. This rubbing will increase the vibration amplitude at coastdown and change the stiffness of the shaft so the critical speed shifted. For loaded conditions, coasting down vibration drops first though, then increases in two peak critical speeds. The increase of vibration amplitude during coast down is not as high as start up to 1500 rpm and 3000 rpm. The vibration stability is getting better. For every start up conditions indicate that rubbing between the rotor and labyrinth seal on stator causes thermal bow is reduced.

**Figure 8.** Hot Spot formation illustration causes rotor bow.

Rubbing between the rotor and labyrinth seal to the stator can produce hot spots and rotor bow illustrated in figure 8. Friction due to rubbing at the same location every shaft revolution and friction in the contact area will cause local heating and hot spots. The rotor will then bow toward the hot spot, creating a new imbalance in the rotor system. This new unbalance will add vector with original unbalance and if the total unbalance is increased, then the vibration also increases in its 1X rotational speed frequency.

**Figure 9.** Rubbing findings on the Labyrinth strip gland seal.
Inspection on the NDE and DE side of the Turbine Casing HP Shows that the Labyrinth strips Gland Seal had indeed sustained a rub, illustrated in figure. 9. Rubbing founded in the axial direction. It occurs between the rotor seal strip and the seal strip on the gland seal. This happens when the HP Differential Expansion exceeds the axial clearance between the Rotor seal strip and the Gland seal strip. The Steam Turbine run-up, steam enters the turbine so that the Rotor and HP Casing will warm up. The effect of heating by steam, both the rotor and casing will experience expansion. During the initial run-up of the steam turbine, the rotor expansion was greater than the HP Casing expansion, so it needed to be monitored with the HP Differential Expansion parameter.

![Figure 10. The direction of rotor expansion.](image)

The direction of rotor expansion illustrated in figure 10. Rubbing occurs if the HP Differential Expansion exceeds the clearance between the rotor seal strip and the gland seal. Rubbing between the seal strip rotor and the seal strip gland causes a thermal bow on the rotor. The thermal bow on the rotor causes vibration to increase. This rubbing causes the strip seal to erode. The more frequent rubbing occurs, the strip seal will thin out and eventually run out. During the inspection, it was found that the condition of the seal strip gland had all been eroded and had run-out. When the seal strips erode all and run out, the vibration is stable. The Steam Turbine managed to run up to 3000 rpm.

4. Conclusions

From the various symptoms that emerged it can be concluded that the cause of vibration instability in the Muaratlaw CCPP steam turbine is due to the rubbing between rotors with labyrinth seal on the stator resulting in a thermal bow on the rotor. At the rubbing section, it is necessary to observe the visual condition of the rotor and stator seal during steam turbine overhaul. Inspection on the HP Steam Turbine Casing shows that the Labyrinth strips Gland Seal had indeed sustained a rub. Rubbing founded in the axial direction. The occurrence of rubbing is not necessarily due to lack of clearance between the rotor and stator seal, which can also be caused by extreme shaft position against the casing. Expansion in the rotor that exceeds the clearance between the seal strips causing rubbing, so to avoid similar things, it is necessary to monitor the HP Differential Expansion during rolling up the steam turbine.

References

[1] A. Sen, M. C. Majumder, S. Mukhopadhyay, and R. K. Biswas, "Condition Monitoring of Rotating Equipment Considering the Cause and Effects of Vibration: A Brief Review" International Journal of Modern Engineering Research, vol. 7, pp. 36 - 49 2017.

[2] M. E. Elnady, J. K. Sinha, and S. O. Oyadiji, "Condition monitoring of rotating machines using on-shaft vibration measurement," presented at the 10th International Conference on Vibrations in Rotating Machinery, LONDON, 2012.

[3] Y. Kaneko, H. Kanki, and R. Kawashita, "Steam turbine rotor design and rotor dynamics analysis," in Advance in Steam Turbine for Modern Power Plants, T. Tanuma, Ed., ed Duxford: Woodhead Publishing, 2017, pp. 127-150.
[4] D. E. Bently and C. T. Hatch, *Fundamentals of Rotating Machinery Diagnostics*. Minden, NV: Bently Pressurized Bearing Press, 2002.

[5] A. Muszynska, "Vibrational Diagnostics of Rotating Machinery," *International Journal of Rotating Machinery*, vol. 1, pp. 237-266, 1995.

[6] T. H. Patel and A. K. Darpe, "Vibration response of misaligned rotors," *Journal of Sound and Vibration*, pp. 609–628, 2009.

[7] N. Bachschmid, E. Tanzi, M. B. Santos, and L. F. Sexto, "Some Experimental Results in Rubbing Phenomena," in *DINAME 2007*, Ilhabela, 2007.

[8] N. Peton, "Balancing With the Presence of a Rub," *Procedia Engineering*, pp. 182 – 191, 2012.

[9] N. Bachschmid, P. Pennacchi, and A. Vania, "Thermally induced vibrations due to rub in real rotors," *Journal of Sound and Vibration*, pp. 683–719, 2007.

[10] P. Pennacchi and A. Vania, "Analysis of Rotor-to-Stator Rub in a Large Steam Turbogenerator," *International Journal of Rotating Machinery*, vol. 2007, p. 8 pages, 2007.