**Vibration of a Steam Turbo-Generator (TG) Set during Shutdown Period**

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**Abstract:** The steam turbo-generator (TG) sets in power plants during the shutdown period often operate at a very low speed with the aid of barring gear. This low speed is known as the barring speed and it is of the order upto 100RPM. The purpose is to float the rotor in the fluid bearings so that the rotor heavy self-weight should not cause any damage to the bearings. The barring speed also maintains uniformity of temperature across the complete rotor when cooling down from operating condition to normal condition. Operating at the barring speed is often considered as the machine is close to the stand still condition and may not be subjected to any harsh vibration problem. However the in-situ vibration measurement on a typical steam TG set during machine shutdown operating at the barring speed reveals that the machine is subjected to vibration which may have potential to propagate the existing damage further. The paper presents a typical industrial case study and observations.

**Keywords:** Turbo-generator (TG) set; Shutdown; Barring speed; Vibration Analysis; Fatigue tests

**1.0 Introduction**

Modern power station practise incorporates two shifting practises, meaning that the turbo-generator (TG) sets used in the power plants are shut down for 8-12 hours every day depending upon the power requirements and demands on day-to-day basis. It is also observed that the steam TG sets used in the power plants are generally not completely brought to stand still (Zero RPM) condition unless some repair or maintenance works need to be carried out. Instead the steam TG sets are allowed to operate at a very low speed with the aid of barring gear during the machine off-load period. This low speed is known as the barring speed and it is of the order upto 100RPM. The purpose is to float the rotor in the fluid bearings so that the rotor heavy self-weight should not cause any damage to the white metal bearings. Jacking Oil Pump (JOP) is also used to circulate the oil at sufficiently high pressure in the fluid bearings to counter the rotor self-weight during this shutdown period. The barring speed also maintains uniformity of temperature in both upper and lower part of turbine blades and shaft when the machine is cooling down from operating condition to normal condition.

The general perception is that the machine during such shutdown may not be subjected to any significant vibration except the background vibration like a machine at the stand still condition. However the in-situ vibration measurement on a typical steam TG set during machine shutdown operating at the barring speed reveals that the machine is subjected to vibration which may have potential to propagate the exiting damage further. This typical steam TG installed in the EDF Power plant, UK which presented a case study evidence of a few cracks in the last stage blades of the LP turbines. There are a number of suspected reasons for the blade cracking [1-6], however the further in-situ vibration measurement during the machine shutdown also revealed that the complete machine and the LP turbines last stage blades are subjected to the resonance throughout the shutdown period which could further contribute to the propagate the exiting crack/damage. A simple vibration fatigue tests on a couple of cantilever beams [7] (simulating turbine blades) with a notch near the clamp end also support the postulates that the propagation of the damage is possible even when subjected to low level vibration excitation. The paper presents the observations and results from the in-situ vibration measurement during machine shutdown on a typical steam TG set and the fatigue tests on the notched cantilever beams.

**2.0 Steam Turbo-Generator (TG) Set**

West Burton Power Plant, UK owned by EDF Energy consists of 4 steam turbo-generator (TG) units for the power generation. Each TG unit consists of a high pressure (HP) turbine, an intermediate pressure (IP) turbine and three low pressure (LP) turbines together with a generator and an exciter. A simple schematic layout of TG sets is typically shown in Figure 1. B1 to B14 are the number of fluid bearings in each TG unit. These units were installed and commissioned between 1967 and 1969 and have since operated smoothly without any major problems up to 2007. In 1995 and 1996, two turbo-generator sets were retrofitted with the rotors with new rotor and blade design for LP turbines and in 2005, retrofitting of new HP rotor for all four turbo-generator units was commenced. The retrofitting was done without changing the foundation, but only with the aim to enhance efficiency and the power output by 20MW. Figure 2 shows the photograph of TG unit where in-situ vibration measurements on the bearing housings and on a few the LP turbine last stage blades are carried out. Cracking of the last stage blades of LP1 and LP2 turbine, steam-end blades has been observed in the 2 TG units fitted with new design LP last stage blades. A typical cracked LP last stage blade is shown in Figure 3.

**3.0 Machine Critical Speeds and LP Turbine Blades Natural Frequencies [1]**

Earlier study [1] and vibration measurements during machine transient operations (rundown and run-up) suggests that the machine critical speeds are 27.34Hz, 33Hz, 40.50Hz and 46.88Hz upto machine 3000RPM. The mode shape at the critical speed of 33Hz is shown in Figure 4. Similarly in-situ modal tests conducted on a few LP1 last stage blades [1-2] are also conducted using the impulse-response method [8]. Modal tests are conducted when machine was completely at stand still condition (zero machine RPM). The mounting of an accelerometer on a blade used during modal tests is typically shown in Figure 5. Typical measured frequency response functions (FRF) plot both amplitude and phase for a LP1 last stage blade is also shown in Figure 6. The experimentally identified first 3 natural frequencies are in the frequency range of 62.87-64.70Hz, 116.6-119.0Hz and 247.8-250.9Hz respectively. The frequency range at each natural frequency of the blades indicates the small scatter in natural frequencies from blade-to-blade. However it is clear that the first natural frequency of the blade is around 64Hz. The 1st and 2nd mode shapes of the LP1 last stage blade from the modal experiments are shown in Figure 7. These data are important for the present study.

**4.0 In-situ Vibration Measurement during Shutdown Period**

The operating period of the TG units is quite flexible at EDF West Burton Power plant during summer period. Often 2 or 3 TG units are usually shutdown every night for 8-12hrs depending on the power requirements and demands. During this shutdown period, the rotor usually rotates at barring speed and the jacking oil pump (JOP) operates at speed of 980RPM (16.3Hz) for circulation of oil in the fluid bearings and to avoid the direct contact between the rotor and bearing within the fluid bearings which prevents the damage of the bearings. The barring speed is just rolling the rotor at low speed with the aid of barring gear which is equipped with the steam TG set. The steam is also not present in the turbines during this shutdown period. Hence it is postulated that the JOP RPM may be a potential source of vibration excitation to the TG set during the shutdown period. The in-situ vibration measurements are carried during this shutdown period when the turbine barring speed was around 60RPM. The measurements are done at the bearing housings for bearings in the vertical direction. Accelerometers of sensitivity 100mV/g with linear frequency measurement range up to 10 kHz are used. The vibration data are then collected and stored in the PC through 16 channels 16-bit data acquisition analog to digital device for further signal processing.

The acceleration amplitude spectra at the bearings B5 to B10 in the vertical direction across the LP1 to LP3 turbines are shown in Figure 8. The high amplitude at low frequency in all spectra indicates the rotor speed 60 RPM related vibration. The frequency peak around 16.5Hz in the spectra relates to the JOP RPM. However the presence of higher harmonic frequency peak near 33Hz with relatively higher amplitude than peak at 16.5Hz in the spectra shown in Figure 8 indicates the possibility of the rotor subjected to resonance at 33Hz even during the machine shutdown condition. The amplitude distribution at 33Hz at the bearings B5 to B10 is also found to be similar to mode shape shown in Figure 4 and hence this confirms the suspicion of the rotor resonance at 33Hz.

The frequency peak near 66Hz seen in most of the spectra shown in Figure 8 is also likely to excite the blade resonance at the LP last stage blade natural frequency of 64Hz. The LP stage blade is also expected to be under resonance condition during the machine shutdown period due to JOP operation. The additional tests were conducted to confirm the suspicion.

**5.0 Additional Tests**

The vibration responses are also measured at the tip of a few number of LP1 last stage (steam end) blades using accelerometers when the rotor was not rotating but excited by the JOP pump operation. The vibration response at the bearing B5 housing in the vertical direction is also measured simultaneously with the blades tip vibration. The schematic of this measurement is shown in Figure 9. This is nothing but a close simulation of machine running at barring speed (i.e., shutdown condition). This experiment is conducted to understand the dynamics of the blades due to the JOP RPM excitation.

Typical measured acceleration responses on a LP1 last stage blade (steam end) and the bearing (B5) pedestal in the vertical direction are also shown in Figure 10. It is obvious from Figure 10 that the peak to peak vibration amplitude of the blade tip is nearly 5 times higher than that of the bearing pedestal. The spectra of the measured vibration responses on the blade and bearing B5 are also shown in Figure 11. The appearance of the frequency peak at 33Hz (2 times of JOP RPM) simply indicates that the rotor of the TG unit is always at resonance at the critical speed of 33Hz even during the shut-down period. Similarly, the LP last stage blades are also always subjected to resonance at much higher vibration amplitude during entire 8-12 hours of machine shutdown nearly every day during summer period. It is because the 4th harmonics of the JOP RPM equals to the blade first natural frequency at around 64Hz. Here blade vibration amplitude at its first natural frequency is nearly 100 times compared to the amplitude of vibration observed at the bearing B5 housing. This observation also confirms that the blades are subject to higher vibration compared to the vibration level observed at the bearing pedestals due to the rotor vibration.

**6.0 Laboratory Fatigue Tests**

If there is a small notch produced due to erosion in the high stress region of bending deflection of the blade then low level of excitation at the blade resonance during the TG shut down state may potentially lead to the crack propagation. Hence the simple fatigue test experiments were conducted on a number of steel beams of size 500mm (length) x 50mm (width) x 3mm (thickness). The beam endurance limit at 106 cycles and the elastic modulus are 210MPa and 208GPa respectively. The beam was clamped as the cantilever (see Figure 12) with the effective cantilever beam length of 478mm (22mm used in the clamping). A small triangular notch of approximately 1mm was also made in the clamped location as pointed in the zoomed view of the beam with the clamp in Figure 13. This experimental arrangement is a simplified setup of a blade with a small notch.

Modal test [8] was then conducted using the instrumented hammer and the accelerometer to find out the natural frequencies and mode shapes. The Frequency of 11.25Hz was identified as the first bending mode of the cantilever beam [7]. The first mode shape of the cantilever beam is generally the deflection pattern of the beam with a vertical point load at the beam free end tip. Hence the maximum possible deflection ($δ\_{max}$) at the tip due to the endurance stress of 210MPa for the beam is calculated as

 $δ\_{max}=\frac{12σ\_{en}L^{2}}{3Et} $ (1)

where $E$= modulus of elasticity,$ t$=thickness of beam, $L$=length of beam and $σ\_{e}$ = Endurance limit. The maximum deflection ($δ\_{max}$) is estimated to be 51.2mm. Hence the 5% of the endurance stress will results in approximately 2.56mm deflection at the beam free end tip. At this stress level the beam is unlikely for fail for virtually infinite number of stress cycles. Hence the beam was excited at first mode of 11.25Hz using a small modal shaker as per the experimental setup shown in Figure 12. Hence the period of 1 cycle was equal to $\frac{1}{11.25} $ = 0.0889s. The excitation level was adjusted such that the tip maximum deflection nearly equal to +/-2.56mm to ensure that the alternating stress equals to just 5% of the endurance stress. The beam free end tip deflection was measured by the accelerometer as shown in Figure 12. The measured acceleration at the excitation frequency of 11.25Hz and the equivalent deflection at the beginning of the experiment are listed in Table 1. The measured deflection was close to 5% hence the experiment was continued for several hours at this setting and the acceleration vibration responses were monitored continuously. If the crack initiates at the notch location then it will propagate further with time. Since the input is the cyclic loading so that the breathing (closing and opening) of the crack during the crack propagation will increase the amplitude of vibration at the higher harmonics (2x, 3x, etc.) of the excitation frequency at 11.25Hz (1x) with time. Typical amplitude acceleration spectra at 121,500 cycles and 536,625 cycles are shown in Figure 14. The amplitudes at 1x, 2x, … are increased at 536,625 cycles compared to 121,500 cycles. This indicates the propagation of crack with time. The trend of amplitude ratios of 2x/1x, 3x/1x and 4x/1x is also shown in Figure 15 with number of cycles. It is very obvious from this plot that the ratios of amplitudes remain constant up to 380,000 cycles which indicates that there is no crack initiation up to this cycle. However, these ratios start increasing with further cycles of excitation means that the crack is initiated at 380,000 cycles which is further propagating with time. Similar observation was made when this experiment was repeated again on another sample of the identical beam. The experiments were further carried out on the identical samples but with excitation level equivalent to 10% and 15% of the endurance limits. The observations are listed in Table 2. The present experiments confirms that even small level of excitation can leads to crack initiation if there is a small notch in high stress area which can then propagates further. Hence the blade vibration during the shutdown period for the case considered here is not healthy.

Table 1 Target Deflection and measured deflection

|  |  |
| --- | --- |
| Target Deflection | Measured Data |
| $$\% δ\_{max}$$ | Deflection,$ mm$ | Measured Acceleration, *m*$/s^{2}$ | Displacement, $mm$ | $$\% δ\_{max}$$ |
|  5 |  2.56 | 12.36 | 2.50 | 4.88 |

Table 2 Summary of observations from all Experiments

|  |  |  |  |
| --- | --- | --- | --- |
| Experiment | TargetDeflection($mm$) | % ofEnduranceLimit | Number of cyclesto initiate CrackPropagation($cycles$) |
|
|
| 1 | 2.56 | 5% | 360,000 |
| 2 | 5.12 | 10% | 220,000 |
| 3 | 7.68 | 15% | 200,000 |

**7.0 Concluding Remarks**

The vibration behaviour of a TG set operation in a Power Plant during the shutdown condition when operating at the barring speed is presented here. Number of additional *in-situ* vibration tests conducted on the LP last stage blades was also conducted together with the laboratory scaled fatigue experiments on a simple experimental setup simulating a blade with a notch due to erosion. It is clear from the study that the operation of JOP always causing resonance of the LP last stage blades even during shutdown condition which has potential to initiate the crack in the exiting notch that can further propagate with time. This machine has already seen the LP last stage blade failure. Therefore the contribution to the blade failure even during the TG shutdown condition cannot be ignored. Hence it is important to conduct the vibration measurement and analysis of such TG sets even during shutdown condition when operating at the barring speed to eliminate the likely problems.

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