Natural Frequency Evaluation of Last Stage Steam Turbine Blade Power Plant Using Finite Element Method

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Abstract. The possibility of a power plant trip can disrupt the stability of the network frequency settings. Load shedding is done at the frequency level. In case of frequency under nominal setting, the load shedding or trip generator decision must pay attention to the generator equipment factor, especially the last stage turbine blade and lacing wire. In this study analysis modal using ANSYS 2019 R2 was carried out to determine natural frequency of the last stage blade LP (low pressure) Steam Turbine Tanjung Awar Awar Indonesia power plant when operating at low frequencies, which is below the nominal operating frequency of 50 HZ or at a rotation of 3000 rpm. The finite element method analysis is applied to the 1 blade and 5 blade systems. The result for a natural frequency of 1 blade system occurs at 2639.6, 2783 and 2915 rpm. For the 5 blade system, natural frequency occurs at 2648 and 2563 rpm, so it can be concluded that the setting under frequency is not in the range of natural frequency last stage LP turbine blade, and lacing wire damage is not triggered by the natural frequency steam turbine blade.

1. Introduction
Electric power systems must be able to provide electricity to customers with practically constant frequency. Frequency deviations from the nominal value must always be within the allowable tolerance range. Active power has a close relationship with the frequency value in the system, while the system load active power and reactive power always changes over time. In this connection there must be an adjustment between the active power generated in the generation system and the active power load. This active power adjustment is done by adjusting the amount of generator drive coupling. According to the comparison of grid frequency settings in various countries, in general power plants are able to continue operating at frequencies of 95% to 105% [1]. Many existing turbine generators must have frequency regulated to avoid mechanical resonance, if rotating machine spins at near one of its resonant modes mechanical vibration damage can occur [2]. In this study, natural frequency verification was carried out at the Last Stage LP (low pressure) steam turbine blade Tanjung awar awar 2 x 350 MW CFPP (coal fire power plant). So that it is expected that under frequency conditions...
according to the network rules there will be no resonance at the last stage blade which impacts on fatigue failure. Figure 1 shows the frequency settings in the Java Bali network in Indonesia. At the frequency of 47.5 Hz, it is expected that the generating unit will at least be able to last 30 minutes connected to the grid. With the development trend toward high parameter and large capacity last stage LP blade become Long and thin [3,4,5,6], the exciting force load on steam turbine blades increase significantly. The blade vibrates intensively and is prone to high cycle fatigue failure [6,7,8,9,10,11,12,13,14,15]. An effective way is to adopt an integral shroud and damping wire in the design of the blade to improve the rigidity and damping of the system. Tanjung Awar Awar 2 x 350 MW CFPP indonesia using long integral shroud blade with damping wire on last stage LP steam turbine blade.

Figure 1. Frequency regulation in Jawa Bali Power grid

On 30 November 2018 found an increase in conductivity from the online analyzer measurement in the condenser that very quickly, at 4600 µs / cm and PH <5.76 the unit was decided to be shutdown and found that the condenser pipe was leaking due to being hit by the impact of a broken lace wire from LP steam turbine blade as shown in Figure 2. Stub lace wire on the blade broke, several tests were carried out to find out the fracture mechanism of the lace wire. There are two installation types of lace wire on the blade, the first is with a tack weld that functions as a stopper between lace wire and stub and the second is by the method of welding directly into the blade. The break of the lace wire was driven by the continued impact of the tack weld lace wire and stub [16], or from the other type While the type of lace wire that is welded directly to the blade, high stress concentration occurs at that point which also triggers crack initiation [17]. Both sources of failure reference conclude that the failure mechanism is from the mechanism of fatigue. In this study will verify the possibility of the last stage LP steam turbine blades operating at their natural frequency, and whether that's the cause of the broken lacing wire.
2. Methodology

In this study, Modal analysis was performed for LP last stage blade turbine on the 1 blade system and 5 blade system. Modal Analysis is an analysis used to determine the vibration potential possessed by the LP turbine structure that is influenced by material and geometry. In Figure 3 shows method in this analysis, 1 blade system represent as free-standing blade and software specifications used to do the analysis are ANSYS 2019 R2, Modal.

![Figure 2. Lacing wire fracture in LP turbine (a) condenser pipe hit by broken lacing wire, (b) position of lace wire fracture, (c) stub rupture hit tack weld, (d) broken wire lace position on the LP turbine stage blade](image)

2.1. Modal Analysis

The Modal analysis was performed to determine the shape of the vibration mode, frequency, and the influence of the turbine rotation on that mode and frequency purely because of the material and geometry of the part. The characteristics of the material used are shown in Table 1.

![Figure 3. Modal Analysis method for 1 blade and 5 blade system](image)
2.2. Model

The model used is the last stage blade which has been fully scanned when overhaul 2019 includes fir tree root, shroud, damping wire hole, and blade surface profile. As shown in Table 2 and Figure 4, 3D scanning and manual measurement are then compared.

Table 1. Material characteristic of LP steam turbine Tanjung Awar Awar CFPP

| No | Parameter (Operating, Dimensional, Or Geometrical) | Unit | Value |
|----|--------------------------------------------------|------|-------|
|    | **Turbine Specification**                        |      |       |
| 1  | Type                                             |      | Sub-critical, singe-reheat, tandem compound, multi-cylinder type steam turbine. |
| 2  | Speed                                            | rpm  | 3000  |
| 3  | Number of extractions                            |      | 7 (3+1+4) |
| 4  | Length of last stage blade                       | mm   | 900   |
| 5  | Annulus area of last blade                       | m²   | 7.35  |

|    | **Material**                                    |      |       |
|----|------------------------------------------------|------|-------|
| 1  | Blade                                           | 17-4 PH | C = 0.07, Mn = 1, Si = 1, Cr = 15.5–17.5, Ni = 3-5, P = 0.04, S = 0.03, Cu = 3-5. Tensile yield Strength = 1275 MPa Tensile ultimate strength = 1379 MPa |
| 2  | Damping Wire                                    | 15-5 PH | C = 0.07, Mn = 1, Si = 1, Cr = 14–15.5, Ni = 3.5–5.5, P = 0.04, S = 0.03, Cu = 2.5-4.5. Tensile yield Strength = 793 MPa Tensile ultimate strength = 1103 MPa |

Table 2. 3D modelling and manual measurement comparison of last stage blade LP turbine

| No | Item         | 3D modelling measurement result (mm) | Manual measurement result (mm) |
|----|--------------|-------------------------------------|--------------------------------|
|    |              | a                                   | 164.56                         | 164.20 |
|    |              | b                                   | 899.55                         | 899.75 |
|    |              | c                                   | 108.00                         | 107.60 |
|    |              | d                                   | 5.01                           | 4.85   |
|    |              | e                                   | 125.58                         | 125.40 |
|    |              | f                                   | 58.74                          | 58.60  |
|    |              | g                                   | 164.56                         | 164.20 |
|    |              | h                                   | 103.54                         | 103.10 |
2.3. Mesh

The mesh settings used are the Hex Dominant method applied to all blade bodies and Sweep applied to the lace wire body. Body Sizing of 8 mm and Face Sizing of suction and pressure side of 8 mm are also applied to the blade. Table 3 summarizes the mesh system 1 blade and 5 blade, and Figure 5. (a), (b) shows mesh 1 blade system and 5 blade system after generate. Detailed damping wire on blade showed in Figure 5. (c)

| Mesh parameter | 1 blade | 5 blade |
|---------------|---------|---------|
| Element       | 18422   | 93543   |
| Nodal         | 49064   | 251079  |

Figure 4. (a) last stage blade manual measurement, (b) 3d scanning and (c) final geometry.
### Method

| Sizing | Blade: Hex Dominant | Lace wire: Sweep |
|--------|---------------------|-----------------|
|        | Blade: 8 mm         | Blade body: 8 mm |
|        | Face: 8 mm (on both suction and pressure side of blade) | Blade face: (on both suction and pressure side of blade) |
|        |                     | Lace wire body: 5 mm |

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**Figure 5.** (a) 1 blade system mesh after generate, (b) 5 blade system after mesh generation and (c) detail damping wire.

#### 2.4. Setup

To model the 1 blades system when LP Turbine operates, the boundary conditions on the fir tree root, blades base, and shroud are set as shown in Figure 6 (a). The root is set as fixed support, and zero displacement is given on the blade base. In shroud, the last stage blade construction is assumed not to under deflection when operating, this approach is carried out to simplify the analysis process, the shroud is modeled as having contacted the blade perfectly. For the 5 blade system when LP Turbine operates, the boundary conditions in the fir tree root, blade base, and shroud are set as shown in Figure 6 (b). Root is set as fixed support, and zero displacement is given on the blade base. Lacing wire is modeled with a bonded connection on the surface. In addition, the shroud connection at the ends of the system is modeled in contact with the same 5-blade system but side by side, the 5-blade system that is
opposite to the above system is not covered in the 3D model but is only modeled as a boundary condition. The connection between lace wire and blade is modeled as no separation contact because the loose-frictional model will cause non-linear calculations and cause the lace wire and blade to penetrate each other on the simulation results.

Figure 6. Modal analysis setup on (a) 1 blade system and (b) 5 blade system.

3. Result and Discussion
From the analysis of normal mode by modeling the vibration as undamped vibration and without the effect of rotation, the first six (6) vibration modes are calculated. A turning point or peak vibration image for each mode is presented, summarized in the Figure 7. From the analysis of damped vibration with the influence of rotation, Campbell's diagram was compiled. Campbell's diagram is a representation of the whole area of vibration excitation that might appear in component operations. This diagram is plotted from the shaft rotation data on the X-axis and the natural frequency of the system on the Y-axis. In contrast to the normal analysis mode in this analysis, the vibrational mode frequency will be affected by the rotational speed as presented in the Table 4. In the Campbell Diagram also plotted linear lines of excitation frequency. When the vibratory mode curve intersects with the linear lines of this excitation frequency, a large resonance is likely to occur. This results in even small excitation that can cause very large vibrations. For this analysis plotted excitation lines up to 6 times ratio. The critical speed value calculated is intersection between linear lines of the excitation frequency and the vibratory mode curve at 2500-3000 rpm operating cycle. This is to take into account the dominant operation area of LP Turbine and numerical simulation errors that may result from frequency calculations in the table in the previous section.
| Speed | Natural Frequency (Hz) |
|-------|-----------------------|
| rad/sec | Hz | rpm | 1   | 2    | 3   | 4   | 5   | 6   |
| 0      | 0   | 0   | 190.34 | 256.22 | 393.08 | 402.19 | 566.53 | 608.39 |
| 62.83  | 10  | 600 | 136.88 | 213.07 | 363.75 | 519.18 | 624.2 | 772.94 |
| 125.66 | 20  | 1200| 87.205 | 172.73 | 328.52 | 551.71 | 849.29 | 1146.1 |
| 188.5  | 30  | 1800| 61.941 | 138.42 | 306.46 | 575.14 | 1103.9 | 1592.9 |
| 251.33 | 40  | 2400| 47.612 | 113.01 | 288.66 | 602.83 | 1375.6 | 2063.1 |
| 314.16 | 50  | 3000| 38.546 | 94.568 | 272.43 | 634.54 | 1658.2 | 2453.2 |

Figure 7. Natural frequency for 1 blade system (a) 190.34 Hz 1st mode, (b) 256.22 Hz 2nd mode, (c) 393.08 Hz 3rd mode, (d) 402.19 Hz 4th mode, (e) 566.53 Hz 5th mode and (f) 608.39 Hz 6th mode.

Table 4. Natural Frequency of 1 blade system with rotational speed.
From the Figure 8, we get the critical speed value in the operating range between 2500-3000 rpm, the excitation frequency ratio 1-6, and the first six vibrating modes as shown in Table 5. This analysis shows that for 1 blade system, there are 3 critical speed values between the operating range of 2500 to 3000 rpm originating from vibration mode 1 at 2639.6 rpm or 43.99 Hz, vibration mode 2 at 2915.2 rpm or 48.59 Hz, and vibration mode 3 at 2783 rpm or 46.38 Hz as shown in Figure 8 with the intersection between vibration mode lines and excitation frequency ratio. So if there are too many looseness on the shroud and damping wire so that the blade operates as if it is a free-standing blade, then it is very dangerous when grid system frequency is low, shown by intersection of 2nd modes at 2915.2 rpm.

**Table 5 Critical Speed 1 system blade between 2500-3000 rpm**

| No | Mode | Critical speed (rpm) | Excitation frequency ratio |
|----|------|----------------------|---------------------------|
| 1  | 1    | 2639.6               | 1x                        |
| 2  | 2    | 2915.2               | 2x                        |
| 3  | 3    | 2783                 | 6x                        |

From the analysis of normal mode by modeling the vibration as undamped vibration and without the effect of rotation, the first six (6) vibration modes are calculated. For the system of 5 blades shown at Figure 9. Through the same damped vibration analysis process, the Campbell Diagram was prepared. The frequency of the vibrate mode to rotate speed is presented in table 6. Through the same process excitation lines up to 6 times ratio was plotted. The critical speed value calculated is the intersection of the linear lines of the excitation frequency to the vibratory mode curve at 2500-3000 rpm operating cycle shown in Figure. 9, and From Figure 9, we get the critical speed value in the operating range between 2500-3000 rpm, the excitation frequency ratio 1-6, and the first six vibrating modes shown in Table 7.
Table 6. Natural Frequency of 5 blades system with rotational speed

| Speed | Natural Frequency (Hz) |
|-------|------------------------|
| rad/sec | Hz | rpm | 1 | 2 | 3 | 4 | 5 | 6 |
| 0 | 0 | 0 | 167.61 | 215.29 | 267.6 | 301.57 | 332.11 | 346.56 |
| 62.83 | 10 | 600 | 159.64 | 178.34 | 257.43 | 303.82 | 358.81 | 419.47 |
| 125.66 | 20 | 1200 | 123.99 | 159.59 | 238.44 | 306.72 | 408.48 | 566.97 |
| 188.5 | 30 | 1800 | 95.15 | 145.49 | 224.75 | 310.22 | 468.43 | 741.31 |
| 251.33 | 40 | 2400 | 75.98 | 130.81 | 215.48 | 314.63 | 535.34 | 929.11 |
| 314.16 | 50 | 3000 | 62.806 | 117.11 | 208.53 | 319.92 | 607.46 | 1124.4 |

Figure 9. Natural frequency of 5 blade system (a) 167.61 Hz 1st mode, (b) 215.29 Hz 2nd mode, (c) 267.6 Hz 3rd mode, (d) 301.57 Hz 4th mode, (e) 332.11 Hz 5th mode and (f) 346.56 Hz 6th mode.
Figure 10 Campbell diagram of 5 blade system.

Table 7 Critical Speed 5 blade system between 2500-3000 rpm

| No | Mode | Critical speed (RPM) | Excitation frequency ratio |
|----|------|----------------------|----------------------------|
| 1  | 2    | 2548.4               | 3x                         |
| 2  | 3    | 2563.1               | 5x                         |

As shown in table 7, This analysis shows that for 5 blade system and lace wire system, there are 2 critical speed values between the operating range of 2500 to 3000 rpm originating from vibration mode 2 at 2548.4 rpm or 42.47 Hz according to frequency system reference at 50 Hz and vibration mode 3 at 2563.1 rpm or 42.71 Hz. so that the integral shroud blade and damping wire increase stiffness and cause it to operate more safely when the frequency of the grid drops. At the frequency of 47.5 Hz, it is expected that the generating unit will at least be able to last 30 minutes connected to the grid.

4. Conclusion

Based on the modal analysis evaluation using finite elements, for the 1 blade system, natural frequency mode 1 is at 2639.6 rpm or 43.99 Hz, mode 2 at 2915.2 rpm or 48.59 Hz, and mode 3 at 2783 rpm or 46.38 Hz. If there are too much looseness on the shroud and damping wire so that the blade operates as if it is a free-standing blade, then it is very dangerous causing by intersection 2nd modes at 2915.2 rpm when grid system frequency is low. for 5 blade and lace wire systems, there are 2 critical speed values, vibration mode 2 at 2548.4 rpm or 42.47 Hz vibration mode 3 at 2563.1 rpm or 42.71 Hz. so that the integral shroud blade and damping wire increase stiffness and cause it to operate more safely when the frequency of the grid drops. It can be concluded that the setting under frequency is not in the range of natural frequency last stage LP turbine blade, and lacing wire damage is not triggered by the natural frequency steam turbine blade.
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References
[1] Mchado I and Arias I 2006 Grid Code Comparison (Gotenberg: Chalmers University of Technology) pp 37-69
[2] Kirby BJ, Dyer J and Martinez C 2002 Frequency Control Concerns in The North American Electric Power System (Tenesse: Oak Ridge National Laboratory) p 3
[3] Zhao W, Liang L and Zhang D 2016 Study on Vibration Characteristics of Damping Blade With Snubber and Shroud Based on Fractal Theory (Xi’an: Jiaotong University) pp 1
[4] Misek T and Kubin ZJ 2008 Static and Dynamic Analysis of I 220 mm Steel Last Stage Blade for Steam Turbine (Plzen: University of West Bohemia) pp133-139
[5] Singh, Murari P and Lucas GM 2011 Blade Design and Analysis for Steam Turbines (McGraw Hill)
[6] Pandhare, Aditya R, Dushant GW and Bhyar DC 2016 Design and Analysis of Last Stage Blades of Low-Pressure Turbine (IJARIIE-ISSN(O)-2395-4396) Vol-2 Issue-6
[7] Szwedowicz J 2008 Balded Disk: Non-linear Dynamics (Basel) pp 1-17
[8] Szwedowicz J 2006 High Cycle Fatigue (Basel) pp 1-54
[9] Wang WZ, Xuan FZ and Tu ST 2006 Failure Analysis of The Final Stage Blade in Steam Turbine (Shanghaai: Elsevier)
[10] Mazur Z, Illescas RG, Romano JA and Rodriguez NP 2006 Steam Turbine Blade Failure Analysis (Morelos, Elsevier)
[11] Kubiak S, Urquiza B, Garcia C and Sierra E 2007 Failure Analysis of Steam Turbine Last Stage Blade Tenon and Shroud (Morelos: Elsevier)
[12] Kubiak S, Segura JA, Gonzales G, Garcia JC, Sierra E, Nebradt G and Rodriguez JA 2008 Failure Analysis of The 350 MW Steam Turbine Blade Root (Morelos: Elsevier)
[13] Sanjeev S, Pandey JP, Ranjit SS, Gupta GK and Modi OP 2015 Coupled Mechanical, Metallurgical and FEM Based Failure Investigation of Steam Turbine Blade (Bhopal, Elsevier)
[14] Wensheng Z, Yanhui L, Meixin X, Pengfei W and Jin J 2018 Vibration Analysis for Failure Detection in Low Pressure Steam Turbine Blades in Nuclear Power Plant (Wuhan:Elsevier)
[15] Segura JA, Castro L, Rosales I, Rodriguez JA, Urquiza G and Rodriguez JM 2017 Diagnostic and Failure Analysis in Blades of a 300 MW Steam Turbine (Morelos, Elsevier)
[16] Qiao L and Feng K 2019 Failure Analysis of The Low-Pressure Blade Lacing Wire in Steam Turbine (Hangzhou: IOP Conference Series: material science and engineering) pp 1-7
[17] Poursaeidi E and Mohammadi Arhani 2010 Failure Investigation of an Auxiliary Steam Turbine, (Zanjan University)