Numerical and Experimental Vibration Analysis of a Steam Turbine Rotor Blade

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Abstract: Damage to the rotor blade of a steam turbine is a relatively common problem and is one of the leading causes of sudden and unplanned shutdowns of a steam turbine. Therefore, the high reliability of the rotor blades is very important for the safe and economical operation of the steam turbine. To ensure high reliability, it is necessary to perform a vibration analysis of the rotor blades experimentally and in a computer environment. In this paper, a modal analysis was performed on the twisted blade of the last stage of the turbine in the Ansys software. The results of the modal analysis of the stationary rotor blade were compared with the results obtained by the bump test, which confirmed the numerical model of the blade. A modal analysis of a rotating rotor blade was performed on the same numerical model, and Campbell diagrams were plotted to determine the critical speeds.

Keywords: Campbell diagram; critical speeds; modal analysis; natural frequencies; rotor blades; vibration modes

1 INTRODUCTION

Steam turbines are rotary machines that are used extensively in thermal power plants and the process industry. The pursuit of higher power output leads to increasing turbine operating speeds. However, higher operating speeds pose a greater challenge to the turbine designers regarding the dynamic machine behavior. Also, the turbine rotor should be balanced in a higher class according to the ISO 1940 standard [1], which ensures an acceptably low level of vibrations caused by the rotor unbalance.

Rotor blades of steam turbines operate in relatively difficult conditions. During operation, they are subjected to high tensile stresses, but resonant vibrations are also common. They are the cause of bending stresses of the blade. In addition to bending and tensile stresses, in high pressure areas the blades are exposed to high working medium temperatures. In low pressure areas blades are subjected to moist steam which causes their erosion. Fatigue caused by alternating stresses or a single stress greater than the allowable one can cause the blade to failure. According to the appearance of the fracture surface, it can be concluded which type of fracture it is (due to fatigue or due to a stress greater than allowed). The main cause of failure of steam turbine blade is high cycle fatigue. [2] An example of rotor blade fracture at the root as a result of high-cycle fatigue is shown in Fig. 1. [3]

Rotor blade fractures caused by static stresses rarely occur in practice. The most common cause of steam turbine failures are fractures of rotor blades due to dynamic stresses. Dynamic stresses are the result of vibrations of the rotor blades, which after a certain number of cycles can lead to fatigue fracture of the blades. Stress risers and corrosion significantly reduce the fatigue strength of the blades, resulting in initial cracks that propagate due to dynamic stresses. When the crack extends so deeply that the stresses, in the reduced cross-section of the blade, exceed the strength of the material, a final violent blade fracture occurs. It is especially dangerous when the rotor blades of the last stages of the energy steam turbine, which are long and have a large mass, failure. Due to the action of centrifugal force, the fracture of such blades causes them to fly out and break

through the thick-walled turbine casings, which leads to catastrophic damage. [4]

Figure 1 Example of rotor blade fracture [3]

A vibration is any movement of the body that is repeated. Avery body that possesses its own mass and elasticity has the ability of oscillatory motion. The vibrations of the turbine rotor blades can be free and forced. The rotor blades vibrate in bending and torsion modes. Bending vibrations can be in tangential (in the plane of the rotor bladed disc) and axial (perpendicular to the plane of the rotor bladed disc) directions. The torsional vibrations of the blades occur around their longitudinal axes.

Turbine blades vibrate due to several of reasons. The most common cause is a periodic force that excites vibrations. Periodic forces, which excite the vibrations of the rotor blades, can have an aerodynamic and mechanical character. Purely aerodynamic causes of vibration are due to the flow of steam through the stator channels and the local flow of steam through the join of the upper and lower halves of the diaphragm. The frequencies of the periodic excitation force generated by the flow of steam through the stator channels are zn and $2zn$ (z is the number of stator blades, and n is the rotor speed in revolutions per second). [5] The mechanical sources of blade vibrations are due to the

vibration of the turbine rotor and the vibration of the discs carrying the rotor blades. The frequencies of the periodic forces in this case are the first and higher harmonics of the rotor speed. In some cases, the so-called self-excited vibrations may also occur. But self-excited vibrations in turbine rotor blades are rare. When the frequency of the excitation force coincides with the natural frequency of the rotor blades, resonance occurs, which is manifested by an increase in the amplitude of vibrations and consequently by an increase in dynamic stresses. Unfortunately, such a phenomenon can ultimately lead to failure of the rotor blades and catastrophic damage of the turbine.

Before the analysis of the rotor blade, in this paper, the modal analysis of the cantilever beam has been performed, for which the natural frequencies and the corresponding vibration modes have been determined. In order to verify the numerical model, the analysis has been done analytically and numerically for the non-rotating (stationary) and rotating cantilever beam. The results of the analytical analysis have been compared with the numerical results obtained using the Ansys software package. [6]

After the numerical and analytical analysis of the cantilever beam, the numerical modal analysis of the rotor blade of the last turbine stage has been performed. The rotor blade is twisted, and the shape of the blade profile changes with the height of the blade. Due to the complexity of the blade shape, the model will be created in the SolidWorks software package. [7] Numerical modal analysis was performed separately for non-rotating (stationary) and rotating blade.

For additional verification of the numerical model, the results of the numerical analysis for non-rotating blade were compared with the experimental results obtained by the bump test. Then the results of the numerical modal analysis of the non-rotating and rotating blades was compared to determine the influence of turbine rotor speed on the blade natural frequencies. Finally, Campbell diagram was done to determine the critical speeds. [8]

2 NUMERICAL MODEL VERIFICATION
2.1 Modal Analysis of the Cantilever Beam

A numerical modal analysis of the cantilever beam shown in Fig. 2 was performed. The dimensions and properties of the beam are given in Tab. 1.
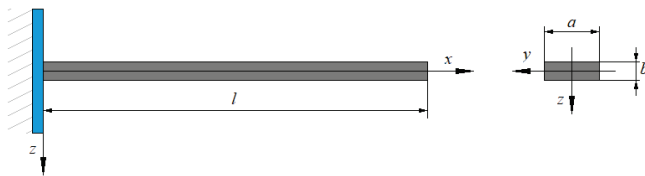

Figure 2 Cantilever beam

Table 1 Cantilever beam properties

Modulus of elasticity	E	200 GPa
Density	ρ	7850 kg/m³
Poisson's ratio	v	0,3
Cantilever beam length	l	0,6 m
Cantilever beam height	b	0,01 m
Cantilever beam width	a	0,024 m

The numerical modal analysis of the non-rotating beam was performed first. The analysis was performed in the Ansys software package. The results of the numerical modal analysis were compared with the analytical results obtained using the expression from [9] (Tab. 2). As can be seen from Tab. 2, the largest deviations are below 1%, which confirms the validity of the numerical model.

Table 2 Comparison of analytical and numerical analysis results of the cantilever beam

Mode	Natural frequency, Hz		Deviation %
	Analytical	Numerical	
Transverse mode y-axis	22,65	22,69	0,18
	141,94	142,00	0,04
	397,46	396,80	0,17
	774,65	775,34	0,09
Transverse mode z-axis	54,35	54,35	0,00
	340,65	338,09	0,75
Torsional mode	868,88	866,45	0,28

A numerical modal analysis of the rotating cantilever beam was then performed. A prestressed modal analysis is performed that involves the influence of centrifugal force due to rotation. This is achieved by performing a static structural analysis first on the rotating cantilever beam, followed by the modal analysis. The result of static analysis are stress and strain fields due to inertial loading. The inertial load is due to the rotation of the beam about the y-axis at an angular speed Ω. The calculation was performed for rotational speeds of 25π rad/s and 50π rad/s. The obtained stress fields are further used as input to the modal analysis. The results of the numerical modal analysis were compared with the semi-analytical results obtained using the following expression [10]:

$$\omega_i = \sqrt{\omega_{in}^2 + \Omega^2 \cdot (c_i - \sin^2\psi)}, \tag{1}$$

where ω_{in} is the ith natural frequency of the non-rotating cantilever beam; ψ is angle of the beam inclination in relation to the axis of rotation; c_i is a constant that depends on the vibration mode. A comparison of these results is shown in Tab. 3. The largest deviations are below 1% which validates the numerical model. In this way, confidence in the numerical model of the cantilever beam is achieved.

Table 3 Comparison of analytical and numerical analysis results of the rotating cantilever beam

Mode	Natural frequency, Hz			
	$\Omega = 25\pi$ rad/s		$\Omega = 50\pi$ rad/s	
	Analytical	Numerical	Analytical	Numerical
Transverse mode y-axis	25,17	25,03	31,56	31,36
	153,52	153,57	183,96	183,93
	410,51	409,79	447,35	446,25
	788,66	789,23	829,28	829,32
Transverse mode z-axis	60,88	60,79	77,05	76,83
	346,54	343,97	363,64	361,01

2.2 Modal Analysis of the Non-Rotating Twisted Rotor Blade

The calculation of natural frequencies and vibration modes was performed on a twisted rotor blade of the last

stage of a steam turbine with a maximum power of 20 MW. The turbine is condensing controled extraction, and it drives an electric generator. The rotation speed of the turbine rotor is 3000 rpm. The number of rotor blades of the last turbine stage is 90, and the number of belonged nozzles is 74. The geometry of the blade model was prepared in the SolidWorks 2018 software package (Fig. 3). [11]
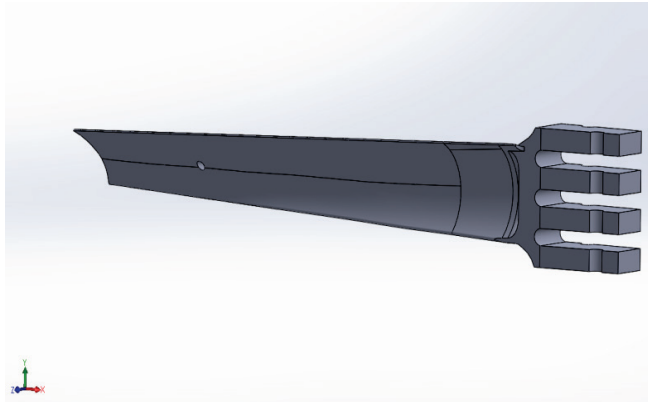

Figure 3 Rotor blade geometry [11]

The above model was imported in ANSYS 17.2 for modal analysis. The material of the blade is high temperature steel X22CrMoV12-1 whose properties are given in Tab. 5. [11]

Table 4 Rotor blade material properties [11]

Steel density	7700 kg/m³
Modulus of elasticity	216 GPa
Yield stress	675 MPa
Ultimate tensile strength	880 MPa
Poisson's ratio	0,28

The element mesh was generated using a three-dimensional 10-node finite element SOLID187 (Fig. 4). The mesh consists of 529418 finite elements and 768457 nodes. [11] All degrees of freedom on the multi fork root surfaces of the blade are constrained because they are firmly attached to the rotor disk. This finite element model was used to obtain the mode shapes and natural frequencies for the non-rotating turbine blade.
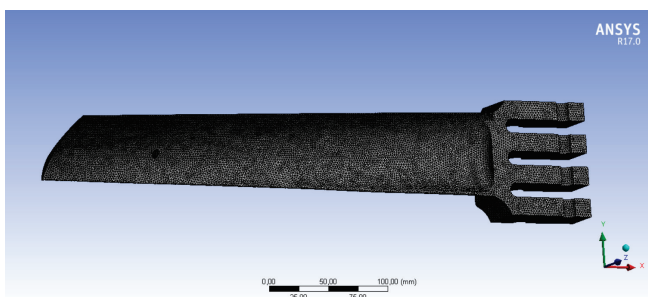

Figure 4 Generated finite element mesh [11]

The natural frequencies of the actual rotor blade were measured using a bump test (Fig. 5). The hits of a special hammer on the object generate impulse excitations in a wide range of frequencies, which excite a large number of natural

modes of vibration. Prüeftechnick type VIBEXPERT II FFT analyzer and Prüeftechnick piezoelectric accelerometer type VIB 6.142 R were used to measure the frequency response. [11]

Figure 5 Measurement of natural frequencies of the rotor blade

A comparison of numerically calculated and experimentally measured natural frequencies of rotor blades is given in Tab. 5. [11] The deviations of the calculated and measured frequencies are acceptably low, which gives full confidence in the numerical model of the blade.

Table 5 Comparison of the results of numerical analysis and measurements on the non-rotating twisted blade [11]

Mode	Natural frequency, Hz		Deviation, %
	Numerical solution	Bump test	
1	190,96	191	0,02
2	362,75	363	0,07
3	789,93	789	0,12
4	953,66	1004	5,01
5	1311,50	1277	2,70

3 MODAL ANALSYS OF THE ROTATING TWISTED ROTOR BLADE

After the above verification of the numerical model, a modal analysis of the rotating blade could be performed. Similar to the case of the rotating beam, a static structural analysis of the blade rotating around an axis coinciding with the longitudinal axis of the rotor shaft was first performed. The blade rotation radius was 533 mm (radius of the rotor disc), and the rotational speed was $\Omega = 100\pi$ rad/s. The obtained stress fields are further used as input to the pre-stressed modal analysis. The results of the numerical modal analysis are given in Tab. 6. [11]

Table 6 Results of numerical analysis of the rotating twistd rotor blade [11]

Mode	Natural frequency, Hz
1	207,82
2	375,75
3	810,35
4	957,86
5	1329,00

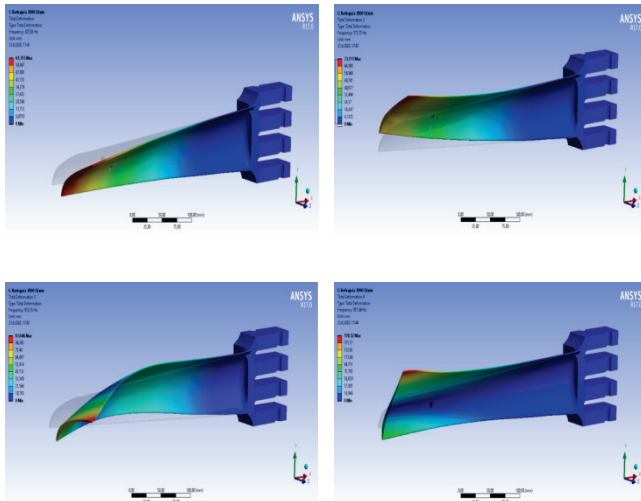Fig. 6 shows the first four modes of the turbine rotating blade. [11]

Figure 6 First four vibration modes of the rotating blade [11]

4 RESULTS AND DISCUSSION

Tab. 7 gives a comparison of the natural frequencies obtained by the numerical calculation for the non-rotating and rotating rotor blades. [11]

Table 7 Comparison of numerically calculated natural frequencies of non-rotating and rotating blades [11]

Mode	Natural frequency Hz	
	Non-rotating blade	Rotating blade
1	190,96	207,82
2	362,75	375,75
3	789,93	810,35
4	953,66	957,86
5	1311,50	1329,00

As can be seen from Tab. 7 the corresponding natural frequencies for the rotating blade are higher than for the non-rotating one. This is physically understandable because the centrifugal force, which results from the rotation, increases the stiffness of the blade. The centrifugal force has the greatest influence on the increase of the first natural frequency, to which the bending mode of vibration belongs (Fig. 6). The centrifugal force has the least effect on increasing the natural frequency of the 4th natural frequency, which is in fact a torsional vibration mode (Fig. 6). The stiffening effect depends on the rotational speed. This means that the natural frequencies depend on the rotational speed. This dependence is represented in the form of a Campbell diagram. The Campbell diagram is used to determine the so-called critical speeds. Critical speeds are the speeds at which the excitation frequencies coincide with the natural frequencies of the rotating structure (resonance phenomenon). The excitation frequency can be synchronous to the rotation speed (1X) or multiples of the rotation speed (2X, 3X, …). The excitation frequencies are plotted as straight lines in the Campbell diagram. The critical speed is the intersection of this straight line with the natural frequency curve.

Fig. 7 shows a Campbell diagram on which the straight lines of the first eight excitation harmonics are plotted. [11] The diagram (Fig. 7) shows that in the whole range of

rotational speeds no natural frequency curve intersects the straight line 1X, 2X or 3X. The curve of the 1st natural frequency intersects the straight line 4X, so the critical speed of rotation is 330 rad/s. This critical speed is higher than the operating speed (314.16 rad/s) by 5%, which means that at the operating speed there will be no resonance of the blade caused by the 4th excitation harmonic. The straight lines 5X, 6X, 7X, and 8X intersect the 1st natural frequency curve at critical speeds that are lower than the operating speed of rotation. During the start or stop of the turbine, this speed range should be passed smoothly, without holding. The straight lines 7X and 8X intersect the curve of the 2nd natural frequency, so the critical speeds are 339.2 rad/s and 293.9 rad/s. The first critical speed is 8% higher than the operating speed, while the second critical speed is 7% lower than the operating speed. It follows that the resonance of the blade caused by the 7th or 8th excitation harmonic will not occur at the operating speed.
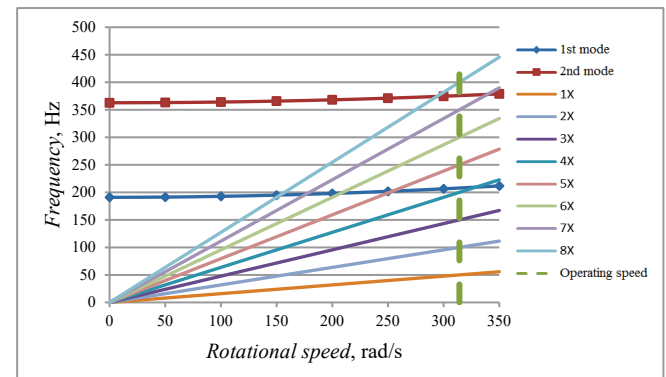

Figure 7 Campbell diagram for the first eight excitation harmonics [11]
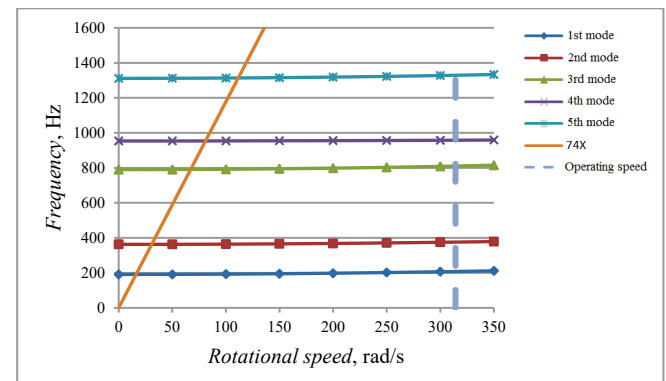

Figure 8 Campbell diagram for 74th excitation harmonic [11]

Excitation of rotor blade vibrations can be caused by steam flow from the nozzle channels. The flow of steam causes pressure oscillations behind each stationary blade. These oscillations are called nozzle wakes. Passing through the nozzle wakes the rotor blades experience a series of impulses. The frequency of these impulses is a product of the number of nozzles and the frequency of rotation. This is called nozzle passing frequency (NPF). Since the number of nozzles is 74, the excitation frequency is then 74X. Fig. 8 shows a Campbell diagram with a straight line 74X plotted. [11] It can be seen in the diagram that all the natural

frequency curves intersect the 74X straight line at different rotational speeds, which represent the critical speeds. All critical speeds are significantly lower than the operating speed. During the start or stop of the turbine, this speed range should be passed smoothly, without holding.

5 CONCLUSIONS

Determining the natural frequencies and the corresponding vibration modes of the steam turbine rotor blades is an important step in dynamic analysis. By knowing the natural frequencies, the occurrence of blade resonance, which could lead to blade fracture and turbine catastrophic failure, can be avoided. This paper shows that numerical modeling in the software package Ansys can successfully calculate the natural frequencies and vibration modes of the non-rotating and rotating blades. Confidence in the numerical model was confirmed by the results of measuring the natural frequencies of non-rotating blades by bump test. The confirmed numerical model could thus be applied to the analysis of a rotating blade, thus calculating its natural frequency values. Due to the influence of centrifugal force on the increase of blade stiffness, the natural frequencies of the rotating blade are higher than the corresponding natural frequencies of the non-rotating blade. Analysis of the Campbell diagram found that blade resonance would not occur at the turbine operating speed. Namely, all critical speeds are at speeds lower or higher than the operating speed.

6 REFERENCES

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