

Frequency Evaluation and Design Optimization of Laced HP Gas Turbine Blade

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Abstract: Rotary blade design is an important research area due to its criticality in applications in turbo machinery like steam turbine, aero or land based gas engines. A small mistuning gives rise to stress development through mode localization under high speed rotating conditions. Various elements like shrouds, lacing wire and tie-bolts are used as blade interconnecting elements. In the present work lacing wire is used as blade interconnecting element and the research deals with comparative case studies of frequency evaluation for without lacing and with lacing blade configurations of a HP Gas Turbine rotor. The lacing wire adds required stiffness to obtain greater structural performance as it is more aerodynamic and feasible when compared to without lacing wires free standing blades. The HP blade length of turbine is optimized based on the stress and the frequency checks. The positioning of lacing wire at 3/4th along the blade height of HP turbine blades and wire cross section considered is circular are considered for this research work. A Design of Experiments (DOE) is carried out to optimize the dimensional parameter of the lacing wire. Assuming root of turbine blade is 100% under fixity condition, the study involves critical parametric evaluations involved in achieving mechanical integrity in airfoil design and blade platform design. Mechanical Integrity involves frequency margin checks, stress checks, analysis of Campbell diagram, checks for gross yield stress and so on, for design and off-design conditions for a given stage efficiency of 83% in an ideal HP turbine of a gas turbine engine.

Keywords: Gas engines, lacing wire, HP Gas Turbine rotor, DOE, mechanical integrity, frequency margin, Campbell diagram

1. Introduction

Mechanical Integrity of gas turbine blades to the turbine rotor is the basic and most important criteria for all gas turbines. The blade provides stiffness against external steady and cyclic varying loads and contributes to the mechanical integrity. The prime significant task in rotary machines, such as turbo machinery is to bring about the finest possible structural performance by overcoming the effects due to gas bending and centrifugal forces upon additionally introducing circumferential blade interconnecting elements throughout the turbine stage rotors.

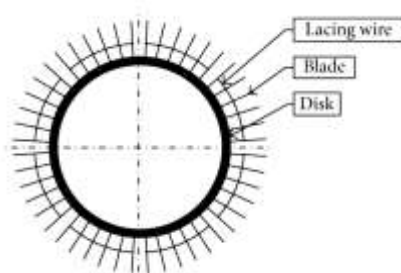


Figure 1: Schematic assembly of blade and disk with lacing wire

Since many decades, rotary engines are being used in applications to produce power with working mediums like water, steam, wind and gas. Incorporating a lacing wire as blade interconnecting element shown in figure 1 decreases the turbine blade dynamic stresses by a factor of 2 to 3, when compared with stresses induced in without lacing wire free standing blade. A stable wire connection around the whole rotor stage is frequently applied to restrain axial turbine blades from harmonic and indeterminate excitations. Lacing

wires during resonance generate considerable friction dissipation due to relative motions between the airfoil and lacing wire vibrations.

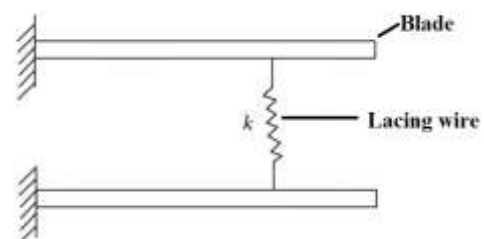


Figure 2: Lacing wire shown as springs

At rotating speeds, the turbine blades are subjected to yielding conditions like Elasto-Plastic and may reach to Fully-Plastic yielding. The centrifugal force induced due to this rotation acts along the blade height from its root hub to tip of the blade, and contributes to an approximately 85% to 90% frequency increase associated with blade torsion and bending modes of vibration, whereas the Coriolis coupling effects leads to other blade vibration modes. Turbine blades are modeled as rotating cantilever beams as shown in figure 2 with airfoil cross-section to study the vibration characteristics related to their dynamic behavior.

The work presented in this paper is to identify the best geometry and dimensional parameter of lacing wires positioned along the 3/4th blade height for a HP stage rotor of a gas engine axial turbine.

2. Problem Definition

The Mechanical Integrity in bladed discs focuses on achieving the best airfoil through aero-structure interaction.

As HP blades are slender blades, subjected to high wake flow excitations, temperature stresses and high alternating stresses in the end stages of the axial turbine of a gas turbine engine, the blades may bend at design and off-design conditions. This asks for the need to introduce structural members interconnecting blades for effective damping to avoid failure due to resonance. Customized methodology is ensured to achieve the best structural performance by introducing circular cross section lacing wires as blade interconnecting element at various locations along the blade.

2.1 Objectives

- To accomplish Mechanical Integration in airfoil design in gas turbines with smooth transition radius to minimize localized stresses.
- To study the importance of lacing wires in damping the blade working frequencies.
- Optimizing the turbine blade length with help of sectional stress and Campbell diagram.
- Optimization of dimensional parameter for the lacing wires.
- Analyze the HP blades for frequency separation margins with Campbell diagrams.
- To bring about a customized methodology for aero-structure interaction in circular cross-section laced blades for best structural performance.

2.2 Design considerations

The 100% operating speed chosen for the engine blade frequency evaluation is 8000rpm. Titanium alloy (Ti-6Al-4V) is chosen as the material for both blade and lacing wire, whose properties are listed in Table.1 below:

Table 1: Shows material property

Density	4.43E-9 Tones/mm ³
Young's modulus	1.138E5 MPa
Poisson's ratio	0.342
Yield strength	880 MPa
Co-efficient of thermal expansion	9.7e-6/°c

Hub thickness is 5mm. Blade transition radius is 2.5mm. Factor of safety assumed is 1.68, for allowable design stress of 522.6537 MPa. Separation margins assumed are left margin 5% and right margin 5% for the chosen operating speed, top 5% and bottom 5% in the Exciter Box. Mean diameter of the rotor is 260 mm, number of blades is 30, pitch of blades in the rotor is 61.8 mm, stage efficiency is 83% and blade axial chord is 24.6mm. For over-speed testing assumed speed is 121% as per American Petroleum Institute (API) standards.

3. Vibration characteristics of HP axial turbines with laced blades

Axial turbines are the most crucial stage in gas turbine as its overall efficiency is what matters to obtain the required thrust and power from the engine. Axial turbine blades are subjected to high pressures and temperatures during the operation envelope. Hence, structural and overall

performances of axial turbine blades play a vital role in a gas engine total performance. In rotating machinery blade design, the determination of blade natural frequencies is an important criterion, since blade failure due to fatigue occurs at resonant frequencies. Hence, study of vibration characteristics plays an important role in safe design of turbo-machines. The HP rotor blade excitations arise from components such as spline joints, blade interconnecting elements, nozzles, coupling bolts, etc. Therefore, engineers have to provide ample blade damping to obtain the required blade life.

In practice the technique used for the minimization of the vibratory stresses in gas turbine blades are to introduce dampers in the form of damping pins, damping wire or lacing wire. A hole is drilled through each blade airfoil and a wire is drawn through these holes which make the blade group stiffer, but there will also be slip between the wire and the blade, providing damping. Frictional forces are then active at the lacing wire outer surface and blade groove lacing contact surfaces and also at the interface along the length of the lacing wire where it talks to each other and hence provides maximum overall frictional damping.

Hence, in the present research lacing wires are chosen as blade interconnecting elements which provide sufficient blade damping. Introducing lacing wires in the blade decreases blade displacement and increases the natural frequencies, and the effect tends to become more predominant as the rotor running speed increases. The contact surface nature of the circular lacing wire will be a line contact in the blade lacing groove.

4. Turbine Blade Modal Analysis and Frequency Separation Margin Evaluation

The free standing blade behaves like a cantilever beam fixed at one end near the blade root and free at the other end near the blade tip. The first 5 modes namely 1st bending, 1st flip, 1st torsion, 1st in-plane and 2nd flip are considered as critical vibration modes for frequency evaluation. The running speed harmonics i.e. $X = (N/60)$, where N denotes rated rpm excitations up to 8X is considered as critical for the wake forces leaving the airfoil for the modal analysis.

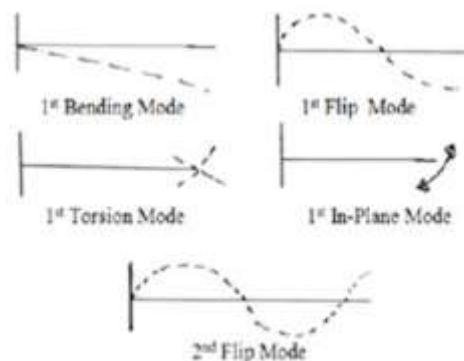


Figure 3: Modes shapes of blade

Figure 3 shows the mode shapes of a free standing turbine blade. Here, the disk mode participation is not considered for the present research. For this purpose Campbell diagram is

drawn at rated 8000 rpm and over-speed conditions. Priority is given for the 1st Mode of turbine blade vibration due to the powerful excitations that come out for the first 6 engine orders. If sufficient separation margins are not achieved, the blade undergoes resonance and is subjected to high cycle fatigue. Hence, tuning of the initial modes with a high margin of safety becomes important. In rotary blades, the coupling between bending and torsion is induced by the lacing wires and the center of flexure is assumed to coincide with center of gravity. There is very less effect of the lacing wire on the blade in 1st torsion and 1st in-plane modes when compared to 1st bending mode which shows considerable effect.

5. Analysis for Optimum Blade Geometry

Case 1: 70 mm, blade model without a lacing wire is considered



Figure 4: A mathematical model of 70mm blade

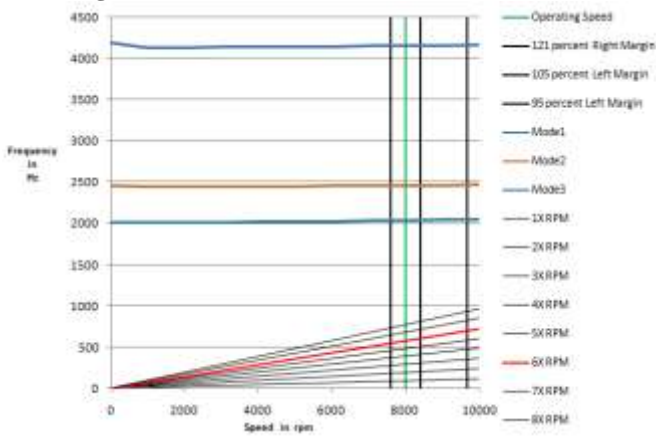


Figure 5: Campbell diagram for blade height of 70 mm

Initially the HP blade considered was of 70mm blade height and based on the Campbell diagram results from Figure. 4; it shows that the first 8 excitations are very far from Mode 1 natural frequency. Hence, calling for a change in blade geometry as the blade is over-designed with higher FOS for design and off-design conditions.

Case 2: Blade model without a lacing wire is considered

By varying length of the blade considering the sectional stress and the frequency 120mm height of the blade is considered as optimum for the present case. The justification of considering 120mm blade height as optimum is given below:

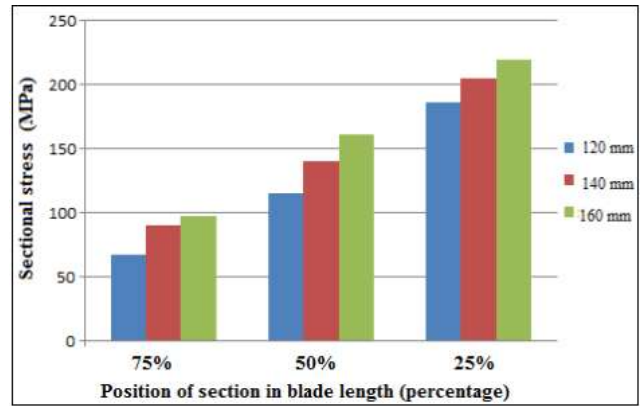


Figure 6: A graph plotted for section stress in blade

Above figure 6 shows a graph plotted for sectional stress for various heights of the blade and with varying length of the sectional plane. At 25% height i.e. bottom of the blade for 160mm height blade has maximum stress and for 120mm has the least when compare to all other heights. Hence, from stress point of view 120mm of blade height is considered.



Figure 7: A mathematical model of 120mm blade

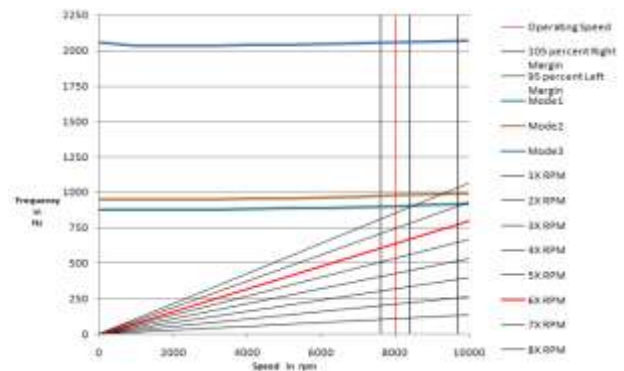


Figure 8: Campbell diagram for blade height of 120 mm

Based on case 1 results, an approach was done to tune the blade frequency to a lesser FOS by incrementally increasing the blade height from 70mm to 120 mm, and the design was stopped at this blade height. The acceptability is well reflected in the Figure.8 for 120 mm, calling for identification of best dimensional parameter of circular cross-section lacing wires, with wire at 3/4th position along the HP turbine blade length.

6. DOE (Design of Experiments)

Gas turbine rotors are often subjected to high transient and fluctuating speed loads. Many key parameters from design, material strength, geometry and behavior play a vital role in reliability and robustness during in-service condition. Sensitivity analysis and DOE become very essential to arrive at an optimum geometry. The design parameters and behavior constraints considered for the present work are to arrive at optimum dimensional geometry of the lacing wire through DOE which are given below:

- DOE is performed by maintain the height of the blade as 120mm.
- The stress of the blade should be within the allowable design limits.
- Allowable radial growth \leq specified tolerance of 1 mm.
- Allowable axial growth \leq specified tolerance of 1 mm.

A three best diameter of the circular cross-section is derived by following above method with specified constraints which is shown in the table below,

Table 2 Lacing wire Specification

First diameter	2.57 mm
Second diameter	2.63 mm
Third diameter	2.71 mm

Case 3: 120mm blade with lacing wire of diameter 2.57mm

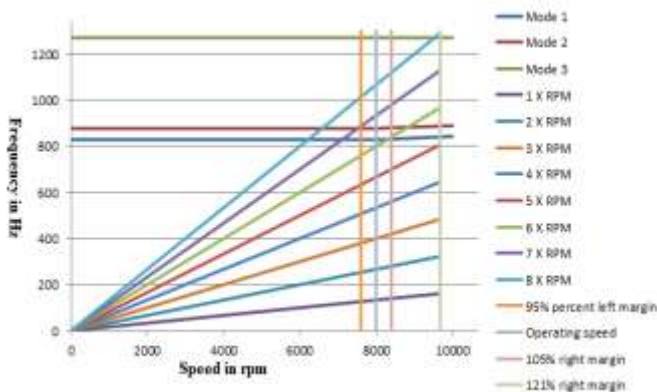


Figure 9: Campbell diagram for lacing wire of diameter 2.57mm

Case 4: 120mm blade with lacing wire of diameter 2.63mm

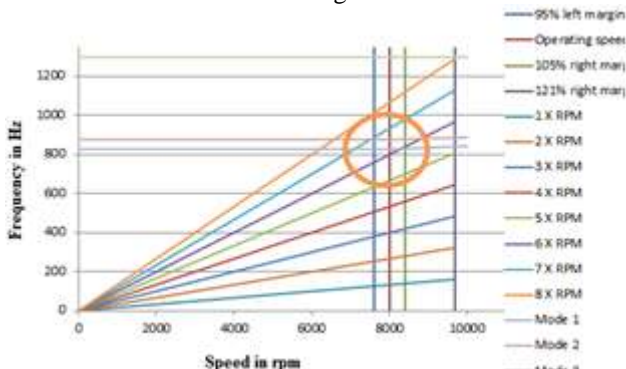


Figure 10: Campbell diagram for lacing wire of diameter 2.61mm

Case 5: 120mm blade with lacing wire of diameter 2.71mm

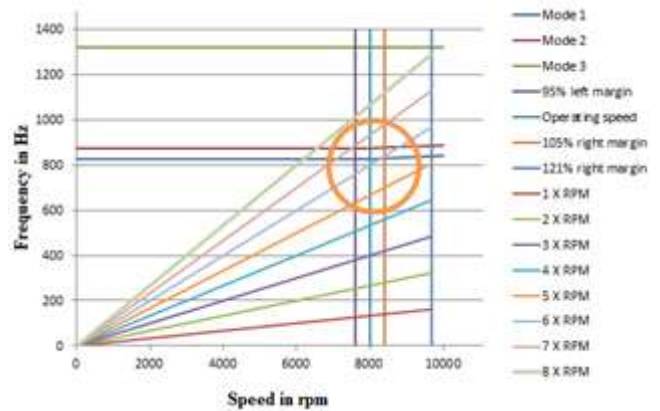


Figure 11: Campbell diagram for lacing wire of diameter 2.71mm

Comparing Campbell Diagram (CD) results from case 3 to case 5, i.e. Figures 9 to 11, it clearly shows that all of the first 6 gas engine powerful excitations are safely passing with enough bandwidth without any resonance for the considered operating speed, separation margins and 121 % over-speed condition. Another important observation to be made is that the case 4 and case 5 shows close proximity to each other results and are the desired test cases when compared to other test cases. The highlighted area in figure.9 and figure .10 called as the Exciter Box justifies for the similarities in vibration characteristics of case 4 and case 5. Hence, we arrive at the conclusion that the case 5, i.e. blade with lacing wire at 3/4th with circular diameter of 2.71 mm is the best optimum design configuration obtained out of the above 03 test cases considered as it is single laced with optimum desired vibration characteristics. The Exciter box is the area of prime importance which shows the frequency separation margins and mode excitations passing through it, which can be best analyzed in detail from figure12.

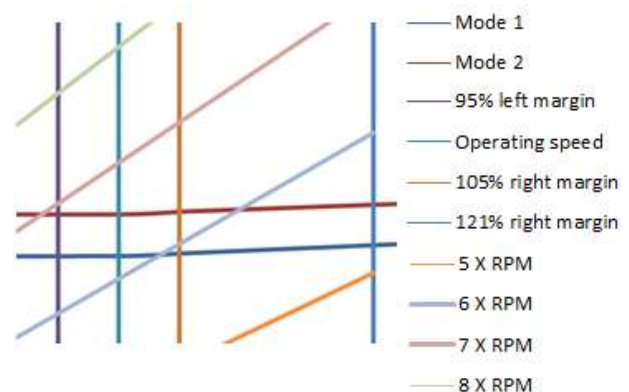


Figure 12: Exciter box for lacing wire of diameter 2.71mm

Figure. 12 is a detailed representation of the 5X, 6X, 7X and 8X frequency excitations passing over the modal frequencies in the considered operating speed range from 95% to 105%, derived from Campbell diagram of Figure. 10. The above excitations are passing safely without any resonance in the considered separation margin regions for an operating speed of 8000 rpm for both Mode 1 and Mode 2 frequencies. Usually in engineering practice, we check for 6X excitations resonance conditions; which is very much satisfied even in the 121% over-speed condition as per requirement of API

standards for blade with lacing wire at 3/4th with circular diameter of 2.71 mm case and is as shown in above Figure. 11. Thus, we have achieved an operating margin of 5% at left, 3% at right, 6% at top and 9% at bottom, as seen in the Exciter Box for Mode 1 for 8X excitation which is very much acceptable and confirms that blade with lacing wire at 3/4th with circular diameter of 2.71 mm has no resonance.

7. Conclusion

- 1) Mechanical integration in airfoil design in gas turbines was achieved with a smooth transition radius of 2.5 mm to minimize localized stresses to 340.12 MPa which is well within the gross yield strength.
- 2) By the means of Campbell diagram and the sectional stress checks the turbine blade model was optimized to 120mm.
- 3) The lacing wire plays a vital role in blade frequency damping, and it is found that elliptical lacing wires provide better damping due to surface contact than circular lacing wires, where line contact exists in the airfoil cross-section along the blade height.
- 4) The best dimension for 120mm blade model with lacing wire at 3/4th position is 2.71 mm.
- 5) From the Exciter Box, it concludes that the % variations of frequencies for the considered test cases are less than 1% for zero speed, less than 2% for 50% load (4000 rpm), less than 3.9% for 100% load (8000 rpm) and less than 5.8% for 121% load (9680 rpm) for an HP turbine stage blade.

References

- [1] Tulsidas.D and M. Shantharaja, 2015. Effect of Taper and Twisted Blade in Steam Turbines. International Journal of Science Technology and Management, Vol.4(1), pp. 2394-1537.
- [2] B.T. Lebele-Alawa, H.I. Hart, S.O.T.Ogaji and S.D. Probert, 2008. Rotor Blades profile influence on a gas turbine's compressor effectiveness. Journal of Applied Energy, Vol.85, pp.494-505.
- [3] G.M.Chapman, E.Swain, X.Wang and M.Yang, 1992. Vibration Reduction of blading using Interconnecting Elements. Rotordynamics-1992, Proceedings of the International Conference on Rotating Machine Dynamics, Venice, Springer-Verlag London Limited, pp.270-277.
- [4] A.L.Salama, M.Petyt, 1978. Dynamic Response of Packets of Blades by the Finite Element Method, Journal of Mechanical Design. Transactions of ASME, Vol.100, pp. 660-666.
- [5] D.J.Ewins, 2010. Control of vibration and resonance in Aero engines and rotating machinery—An overview. International Journal of Pressure Vessels and Piping, Vol.87, pp.504 -510.
- [6] V.A. Bazhenov, A.I. Gulyar, S.O. Piskunov and V.P. Andrievskii, 2013. Design Life assessment of the blade root of a Gas Turbine unit under thermo mechanical loading. Journal of Strength of Materials, Vol.45(3), pp. 329-339.

- [7] S. Narasimha, G. Venkata Rao and S. Ramakrishna, 2009. Stress and Vibration Analysis of a Gas Turbine with a Cottage-Roof Friction Damper using Finite Element Method. 14th National Conference on Machines and Mechanisms (NaCoMM-09), NIT Durgapur, India, Paper ID: DVAMSN8, pp.316-324.
- [8] Mahesh Shankar, K. Kumar and S.L. Ajit Prasad, 2010. T-root blades in a steam turbine rotor: A case study. Journal of Engineering Failure Analysis, Vol.17, pp.1205-1212.
- [9] Wen-hu Huang, 1981. Free and Forced Vibration of Closely Coupled Turbomachinery Blades. AIAA Journal, Vol.19(7), pp.918-924.
- [10] G.M.Chapman and X.Wang, 1988. Interpretation of Experimental and Theoretical Data for Prediction of Mode Shapes of Vibrating Turbocharger Blades. Journal of Vibration, Acoustics, Stress, and Reliability in Design, Transactions of ASME, Vol.110, pp.53-58.
- [11] N.N: Magazine of Modern Power Systems. 1995, pp.49.
- [12] M.M.Bhat, V.Ramamurti and C.Sujatha, 1996. Studies on the Determination of Natural Frequencies of Industrial Turbine Blades. Journal of Sound and Vibration, Vol. 196(5), pp.681-703.
- [13] B.L.Jaiswal and S.K. Bhawe, 1994. Experimental Evaluation of Damping in a bladed disk model. Journal of Sound and Vibration, Vol.177(1), pp.111-120.
- [14] A.L.Deak and R.D. Baird, 1963. A Procedure for Calculating the Packet Frequencies of Steam Turbine Exhaust Blades. Journal of Engineering for Power, Transactions of ASME, pp.324-329.
- [15] Animesh Chatterjee, Mangesh.S.Kotambkar, 2015. Modal characteristics of turbine blade packets under lacing wire damage induced mistuning. Journal of Sound and Vibration, Vol. 343, pp.49-70.
- [16] E.Poursaeidi, M.R.Mohammadi Arhani, 2010. Failure investigation of an auxiliary steam turbine. Engineering Failure Analysis, Vol.17, pp.1328-1336.