Multiphysics based Modal and Harmonic Analysis of Axial Turbines

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ABSTRACT
The presented work discusses the vibration behavior of an axial turbine, analyzed using the finite element method (FEM) using commercial software ANSYS®. The turbine was modeled as a 2-D plate type of structure discretized with 4-noded shell 63 elements to save computational time. Constraints were applied keeping in view the actual operating conditions. The turbine was modeled with more than one FE meshes. Mesh sensitivity analysis was carried out to ensure the quality and independence of the results. Modal analysis was conducted to calculate a few initial natural frequencies. Results were studied against the operating frequency of the turbine. After carrying out the modal analysis, harmonic analysis was performed to see the response of the turbine under dynamic loading. The nature and cause of the dynamic loading are also discussed in relation to dynamic behavior. It was observed that the turbine is safe in its entire range of operation as far as the phenomenon of resonance is concerned. Also, it was observed that the maximum harmonic response of the turbine on the application of dynamic loading is far lesser than its failure limit within the specified operating range.

1. INTRODUCTION

Vibration deals with the oscillatory motion of dynamic systems and is a combination of matter possessing mass, elasticity, where parts are capable of relative motion [1,2]. Dynamic systems are well-studied and there are a number of examples in aerospace and automotive industry [3-7]. All bodies possessing mass and elasticity potentially can vibrate at specific natural frequencies if excited appropriately. For example, a study involving drop-weight-impact-test of composite clearly demonstrate this phenomenon [8,9]. The resonance phenomenon occurs if the natural frequency of the structural system matches with the frequency of dynamic loading, which may reveal system characteristics [10,11] and if not planned may lead to structure failure [12-14]. Tremendous efforts have been made to study the failure mechanism including use of novel techniques such as thermography [15-17].

Access to computational power has supported the use of numerical methods for various complex multiphysics studies [18,19]. Finite element (FE) methods are being used at large extent for structural analysis [20]. Commercial Software ANSYS® Inc. is well-known for its use for structural analysis [21,22]. A vast number of case studies has been reported where ANSYS® has been employed for complex structural analysis [23-32].

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Efforts has also been made to analyze the structure analytically highlighting key assumptions and limitations [33-35]. It is also well-known that extreme conditions contribute towards structural failures [36-39].

The primary objective of this study is to carry out the vibration analysis of an axial turbine. The axial turbine is part of a turbojet engine, an air-breathing jet engine often used in aircraft. Turbojet engine consists of multiple stages of axial turbine with a narrow propelling nozzle on one end. The engine has an air inlet which includes inlet guide vanes, compressor, combustion chamber, and a turbine that is coupled to drive the compressor. The compressed air from the compressor is heated by burning fuel in the combustion chamber and then expanded through the turbine. The turbine exhaust is then expanded in the propelling nozzle, where it is allowed to accelerate at high speed to obtain the required thrust [40-43]. During the process, turbine blades are subjected to high RPM, which may cause undesirable vibration [44-46]. The objective of the study is to determine its effect on the vibration characteristics and safety of the system under consideration.

In this work, FE method was used for the vibration analysis of turbine using commercially available software ANSYS® [47]. CAD model was developed as per the design specification. Nickel based alloy Inconel 718 material [48] properties were chosen for analysis.

2. METHODOLOGY

For this work, the geometry was built in Pro-E CAD software [49], which can be seen in Figure 1. This geometry was imported into ANSYS® modeling environment by converting CAD file into IGES format. Additional trimming was done in ANSYS® modeling environment. To save computational time, the turbine has been modeled as a 2-D plate type of structure. This was meshed with 4-noded shell 63 element of ANSYS®.

Cyclic symmetry was utilized to decrease the computational requirement for analyses. Complete turbine was divided into 53 cyclic symmetric sectors according to the number of blades in the turbine. Figure 2 shows the finite element mesh of a single sector. For the purpose of the analysis, constraint equations are developed to correctly simulate the turbine disc and blade joints. After that one sector is solved and result can be expanded to the complete turbine.

In order to gain confidence on the results, turbine was modelled with more than one mesh. Mesh sensitivity analysis were then carried out to have a mesh with an optimum number of nodes and elements. An optimized mesh should be capable of giving accurate results with minimum utilization of computational resources.

As far as boundary conditions are concerned, all nodes at turbine shaft-linkage as shown in Figure 3 were constraint in all DOFs (Degree of Freedoms). Node coupling was used to connect nodes at blade-hub Joint (see Figure 4). Both analyses were carried out using linear-elastic material model.

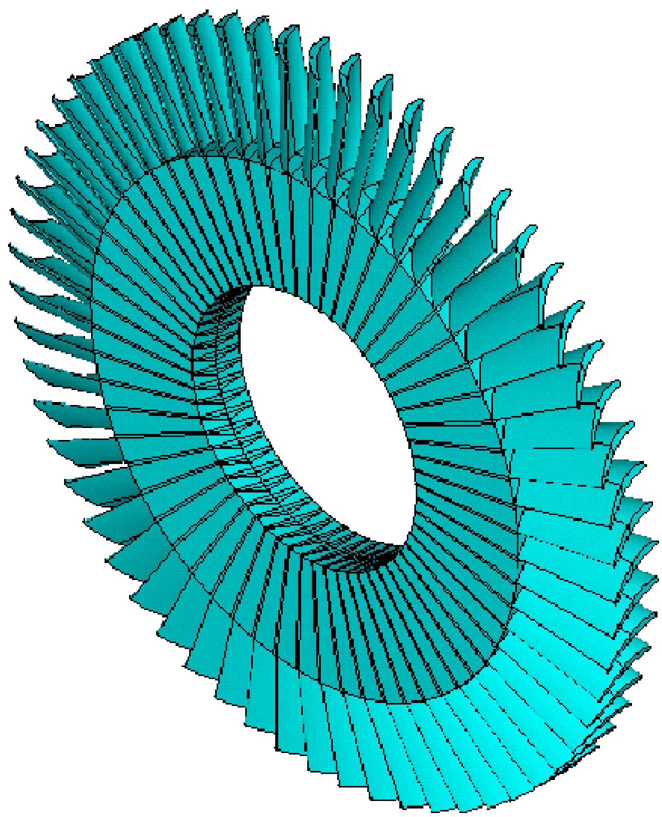


Figure 1. CAD Model of Turbine

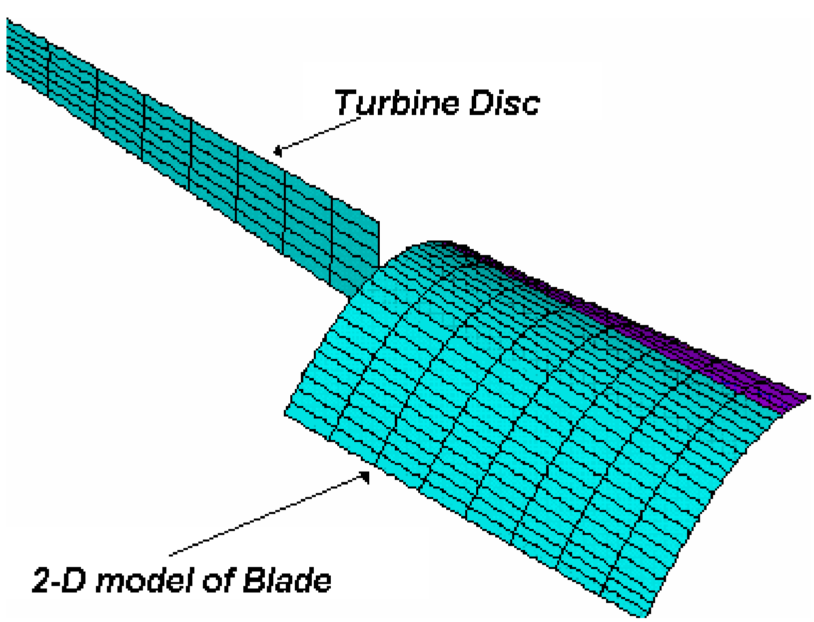


Figure 2. Finite Element of Turbine Sector

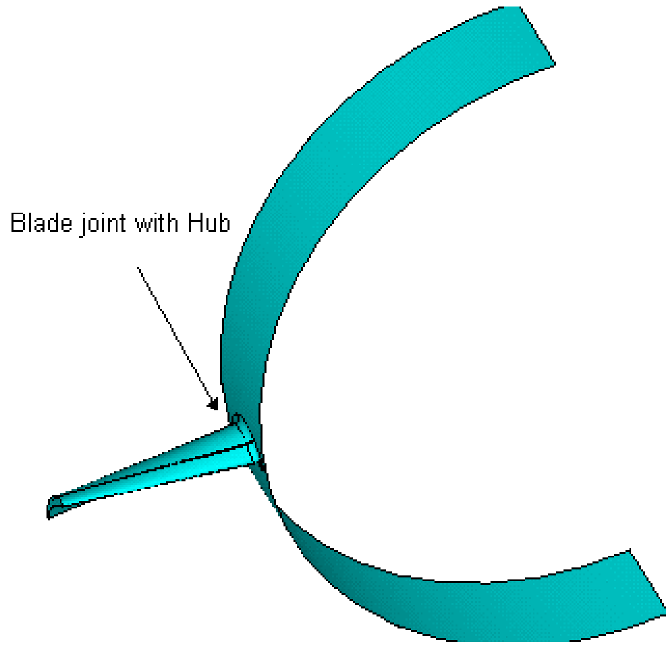


Figure 3. Illustration of Blade Hub Joint (Single Blade)

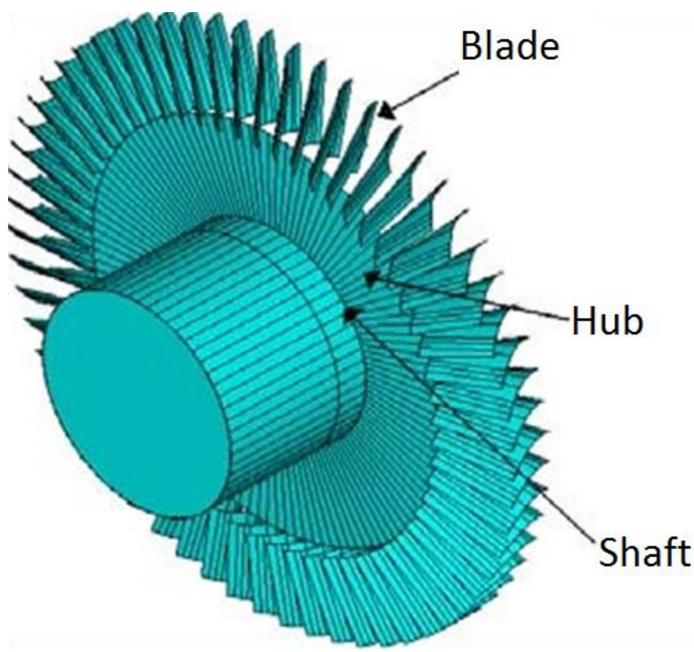


Figure 4. Illustration of Turbine Shaft Linkage

2.1. MODAL ANALYSIS

Modal analysis is used to determine the natural frequencies of a structure [50,51]. The natural frequencies and mode shapes are important parameters in the design for dynamic loading conditions [52-55]. Modal analyses can be performed on a pre-stressed structure, such as a spinning turbine blade. For the modal analysis of a turbine, loads and constraints were applied keeping in view the actual situation. Loads and operating conditions of turbine as per the aero-propulsion experts are specified in Table 1.

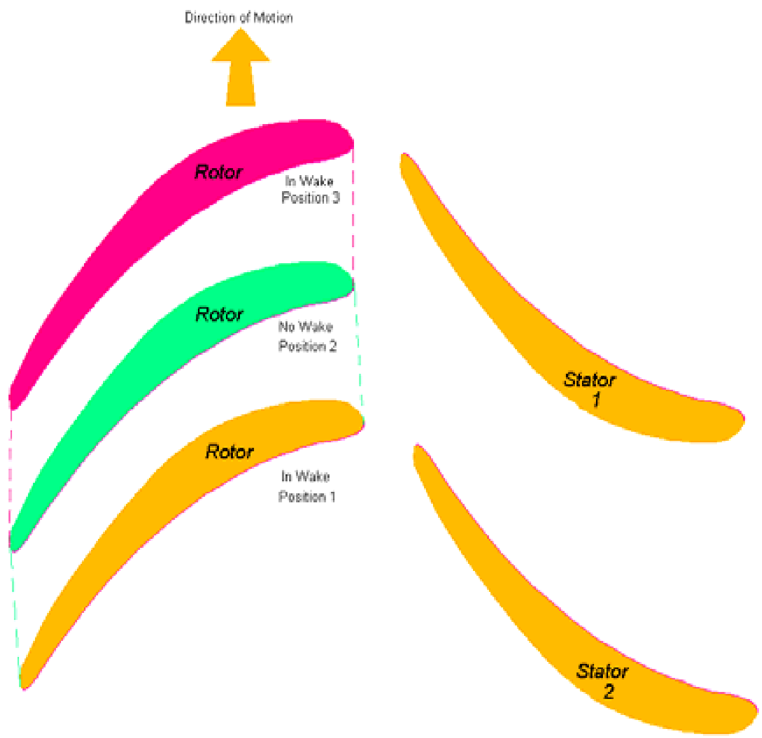
Table 1. Operating Conditions of Turbine

Maximum Pressure	0.383 MPa
Maximum Temperature	944.44 K
Operating RPM (Inertia)	32200 RPM

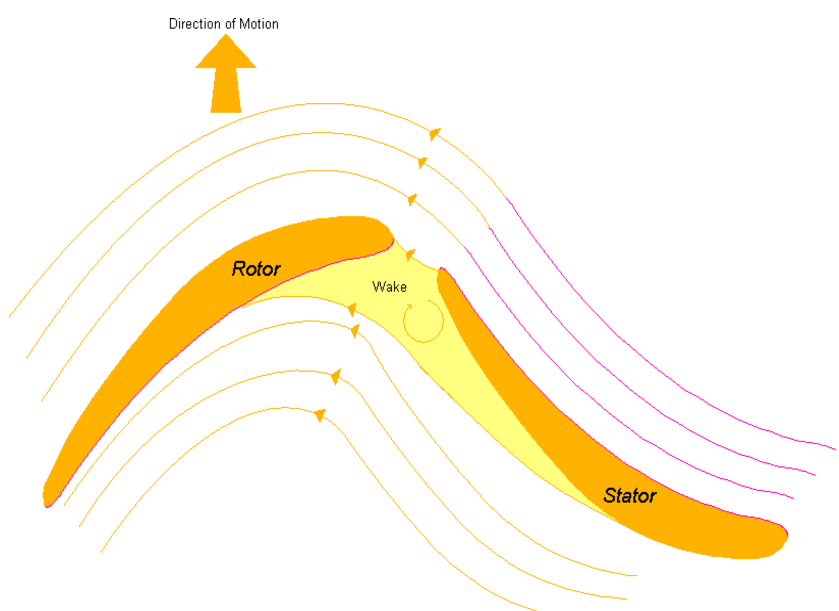
2.2. HARMONIC ANALYSIS

Any sustained cyclic load will produce a sustained cyclic response (a harmonic response) in a structural system. Harmonic response analysis is used to predict the sustained dynamic behavior of the structure, thus verifying whether or not structure will successfully overcome resonance, fatigue, and other harmful effects of forced vibration.

Harmonic analyses require cyclic load data for the analysis. This load data was gathered as per the aerodynamic analyses of turbine carried out earlier by aero-propulsion experts. It was found out that pressure load is the only load which is cyclic in nature under steady state operation. Reason for this variation is the existence of wake due to the presence of nozzle guide vanes (stators) before turbine rotors. Whenever the rotor passes by a stator it has to pass in low pressure region (wake). The repetition of this low-pressure region is equal to the number of stators before the turbine. Frequency of this dynamic loading is equal to product of number of stators and turbine RPM. This phenomenon is explained in Figures 5(a) and 5(b). This analysis was carried out for normal and over run operation (32200 & 38000) X 31 RPM or 16636 to 19633 Hz. The number of stators before turbine rotors is 31.



(a) Rotor positioning according to the stators



(b) Rotor in the wake of stator location

Figure 5. Illustration of pressure variation on rotor blades due to the wake of stator blades

3. RESULTS AND DISCUSSION
3.1. MODAL ANALYSIS RESULTS

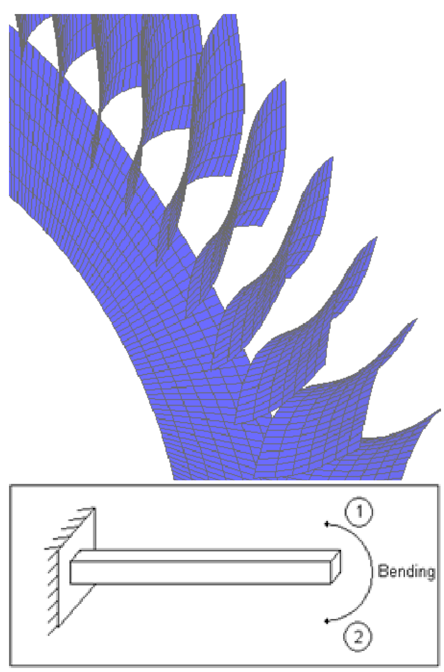
First five natural frequencies were calculated for both the cases i.e., with stress stiffening on and with stress stiffening off. Results of modal analysis have not been affected much because of applied loads. The comparison of both the results is given in Table 2. We found that there is not much difference in both the results. We also observed that operating RPM of turbine is far lesser in value than first modal frequency as shown in comparison Table 3. Mode shapes obtained for turbine are provided in the Figure 6.

Table 2. Modal Analysis Results

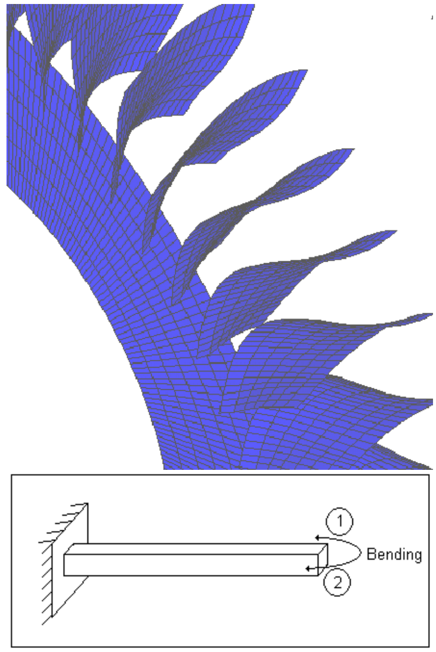
Mode No.	Natural Frequencies with Stress-Stiffening Effect (Hz)	Natural Frequencies without Stress-Stiffening Effect (Hz)	Mode Shape Nomenclature
1	2250	2248	1st Bending
2	6080	6075	2nd Bending
3	8185	8176	1st Twisting
4	10102	10095	3rd Bending
5	14886	14876	4th Bending

Table 3. Comparison of minimum modal frequency and operating RPM

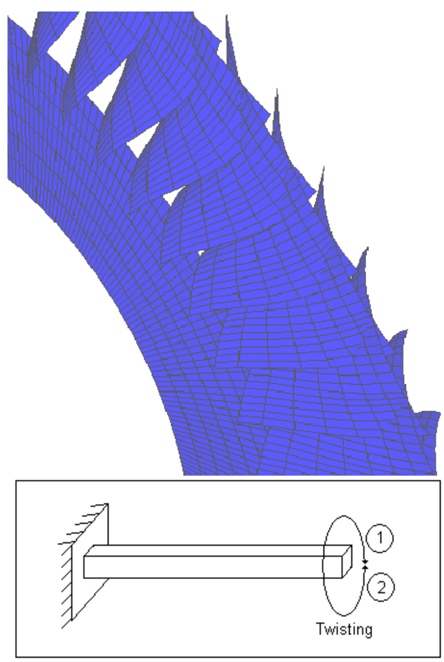
Operating RPM(Hz)	First Modal RPM(Hz)
32200 (537)	135000 (2250)



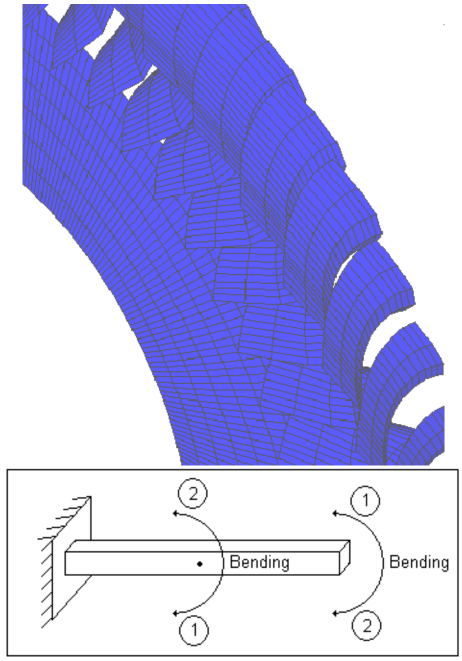
(a) 1st Mode Shape (1st Bending)



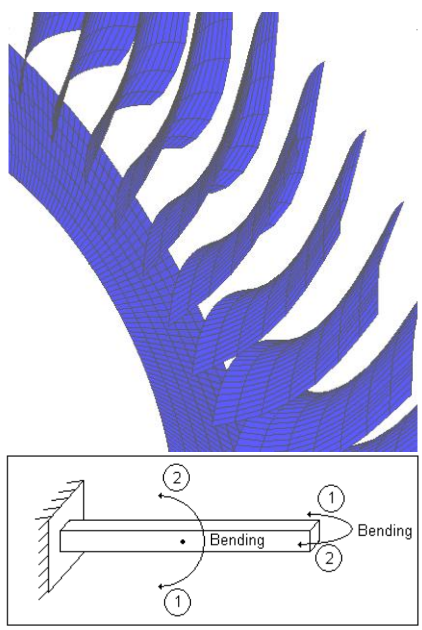
(b) 2nd Mode Shape (2nd Bending)



(c) 3rd Mode Shape (1st Torsion)



(d) 4th Mode Shape (3rd Bending)



(e) 5th Mode Shape (4th Bending)

Figure 6. Turbine Mode Shapes with Description

3.2. HARMONIC ANALYSIS RESULTS

Harmonic analysis results are in the form of a graph commonly known as FRF (frequency response function). This FRF is specific to the node at the trailing edge of the tip of the turbine blade, which showed the most aggressive behavior to cyclic loading. Other nodes were also analyzed, as they showed similar response, but value of displacement was lesser compared to selected node, so shown FRF is associated to tip trailing edge node. Location of specific node is shown in Figure 7. FRF for this analysis is shown in Figure 8.

Value of the maximum displacement indicated by FRF is very small as compared to the value of maximum displacement calculated using static analyses of turbine, under which turbine blade is safe for operation. The comparison of displacements at maximum RPM, obtained through analyses is shown in Table 5.

Tab Table 5. Comparison of value of maximum displacements at 19633Hz

Max. Displacement (Harmonic Analysis)	Max. Displacement (Static Analysis)
0.0001 mm	0.40 mm

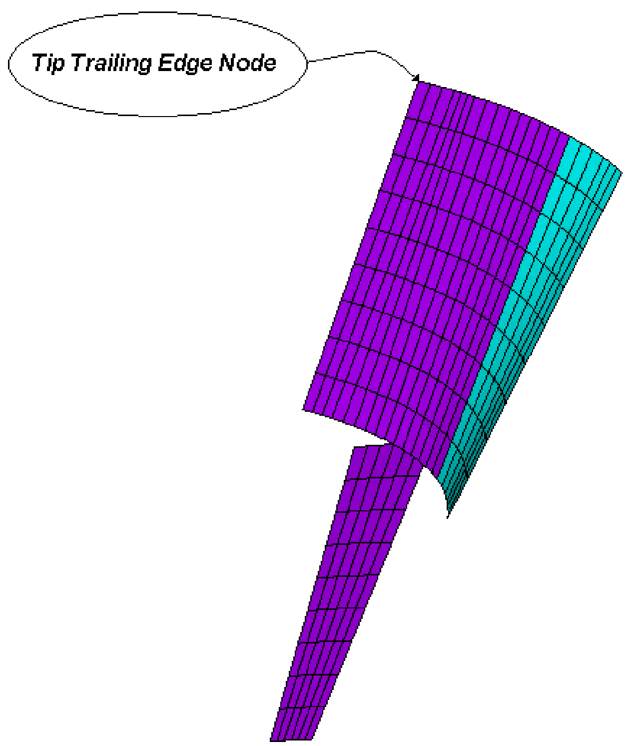


Figure 7. Node at Trailing Edge of the Tip of the Turbine Blade

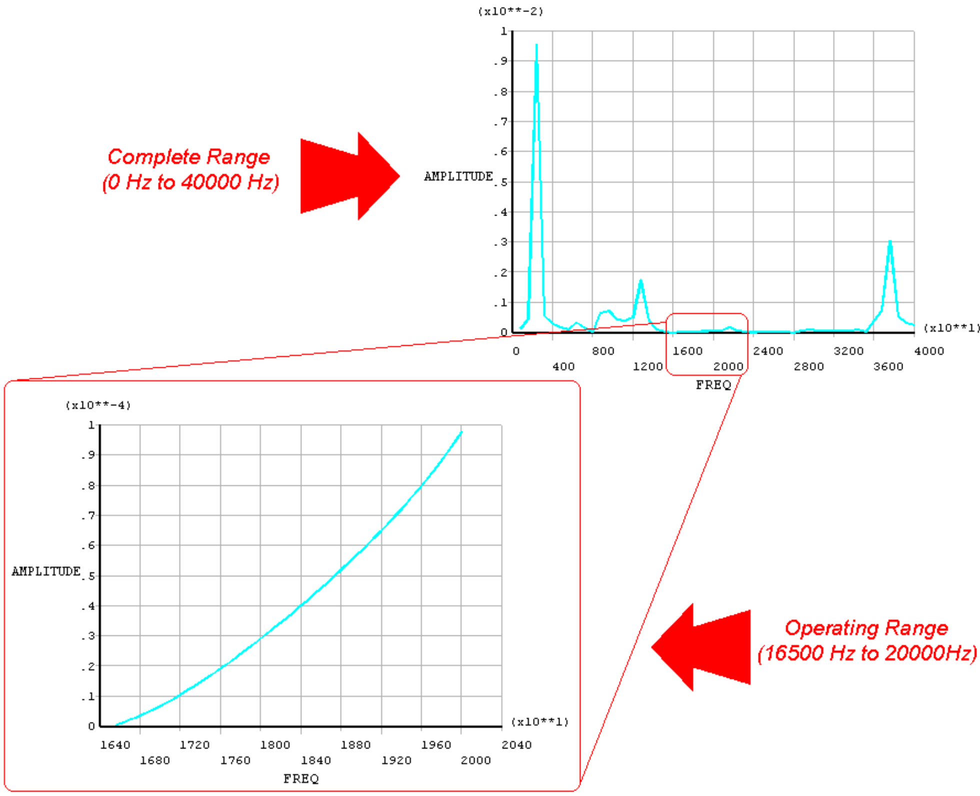


Figure 8. FRF (Frequency Response Function)

4. CONCLUSION

Natural frequencies as per the modal analysis are shown Table 2. This turbine has to operate at 32200 RPM or 536 Hz as per the design specification. The evaluation shows that the first natural frequency is far higher than maximum operating frequency, this gives clear indication that turbine is safe against resonance phenomenon. Also, harmonic response within specified range is also acceptable, as maximum value of displacement is far lesser than static displacement. Also, FRF indicates no chances of the occurrence of resonance within this range.

ACKNOWLEDGEMENT

The publication charges for this article have been funded by a grant from the publication fund of UiT-The Arctic University of Norway.

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