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Natural Frequencies and Vibration Modes of an Aeronautical Axial Compressor Blade and Disk

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***Keywords*— Natural Frequencies; Vibration Modes; Axial Compressor; Aeronautical Gas Turbine; Finite Element Method.**

***Abstract*— This article presents the natural frequencies and vibration modes of the blade and disk of the first stage of an axial compressor from a 5-kN thrust aeronautical gas turbine. The determination of the natural frequencies and vibration modes of these components enables an assessment of their dynamic behavior relative to the operating range, aiming to avoid resonance. There are no analytical solutions for the natural frequencies and vibration modes of structures with such complex geometries; therefore, calculations are performed using numerical computer simulations with Ansys® finite element software. To ensure the reliability of the results, the software is calibrated using simple beams for which analytical solutions exist. After verifying excellent agreement between the analytical solutions and the numerical computational results provided by Ansys® for simple beams, the same methodology is applied to the blade and disk of the first-stage axial compressor, which feature complex geometries. The results obtained for the blade and disk of the aeronautical axial compressor under study show excellent consistency - given the geometry and material used - and allow for an assessment of the components' dynamic behavior to prevent resonance-related issues.*

I. INTRODUCTION

Determining the natural frequencies and vibration modes of a given mechanical component is crucial for ensuring its failure-free operation regarding potential excitation frequencies. If a system continues to vibrate on its own after an initial disturbance - without the action of external forces - the frequency at which it oscillates is known as its natural frequency [1]. Knowing the natural frequency values is important because it allows for the avoidance of phenomena such as resonance. Resonance occurs when a time-varying excitation force coincides with the component's natural frequency. This causes the vibration amplitude to increase, potentially leading to fatigue failure or even catastrophic structural collapse [2].

Finite element analysis simulation software is typically used to calculate natural frequencies and determine vibration modes for complex geometries [3]. In this study, the finite element software Ansys® is used. By discretizing the structure into finite elements, we move away from dealing with a continuous, infinite equation, which is difficult to solve analytically and often lacks a closed-form solution, and instead work with a finite set of variables. This transforms a complex differential equation problem into a numerical matrix problem that can be easily solved computationally.

Since numerical methods are approximation techniques, one way to validate them is to compare the results with simple models for which analytical solutions are already known. Therefore, to ensure confidence in the results presented in this study, the software is calibrated using simple beams that have known analytical solutions [4]. After verifying excellent agreement between the analytical results and the numerical computational solutions provided by Ansys® software for simple beams, the same methodology is applied to the blade and disk of the first stage of an axial compressor for a 5-kN thrust aeronautical gas turbine [5] - components characterized by complex geometries according to Figure 1. The results obtained for the blade and disk of the axial compressor under study show excellent consistency, given the geometries and material used, and enable an assessment of the components' dynamic behavior to prevent resonance-related issues.

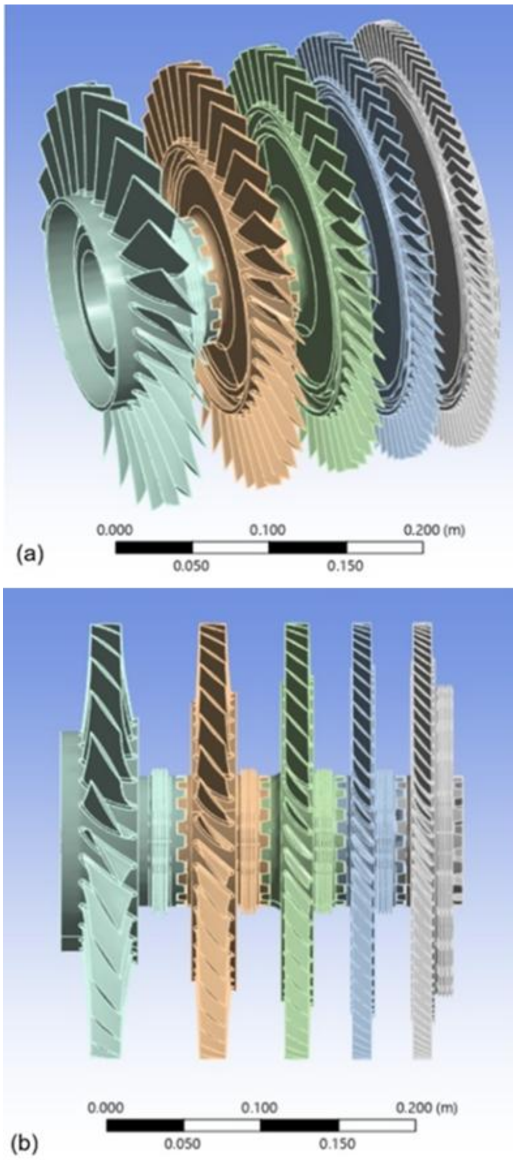

Figure 1. Blades and disks of a 5-stage axial compressor for a 5-kN thrust aeronautical gas turbine: (a) perspective view; (b) side view.

II. METHODOLOGY

Gerhardt and Silveira [6] describe the deductive method as one that starts from principles recognized as true and indisputable, enabling conclusions to be reached in a purely formal manner based on logic. Lakatos and Marconi [7] add that true premises inevitably lead to a true conclusion. On this hand, this work is based on pre-defined classical concepts of solid mechanics, being largely deductive and consistent with established theories available in literature.

Silva and Menezes [8] describe bibliographic research as a process involving the survey and analysis of sources such as books, journals, and standalone publications, among others. Gil [9] argues that the advantage of this type of research strategy is the possibility of exploring a wide variety of phenomena. On this hand, the technical procedure for this work employs bibliographic research, conceptual analytical analysis, and computer simulation as research strategies.

Regarding the problem approach, this research is classified as quantitative. According to Silva and Menezes [8], quantitative research involves translating opinions and information into numbers for classification and analysis. Conceptual analytical analyses and computer simulations provide numerical data that are then subjected to analysis.

In terms of nature, applied research aims to generate knowledge for practical application, directed at solving specific problems. It involves local truths and interests [6]. Based on this definition, this research is classified as applied, as it seeks to generate knowledge with a practical purpose - specifically, analyzing the dynamic behavior of an actual aeronautical gas turbine - thereby contributing to the assessment of its structural integrity and safe operation.

According to Hulley et al. [10], in a transversal-time study, all measurements are taken at a single point in time, without a sequential period. Transversal-time studies are useful when the goal is to describe variables and their distribution patterns. Based on this, this study is classified as transversal-time development, as the research aims to conduct a snapshot analysis of variables related to specific gas turbine blade and disk. Data collection and analysis are performed at a single point in time, without continuous monitoring or observation over time.

Finally, considering its specific objectives, this research is characterized as descriptive; according to Triviños - cited by [6] - this type of objective seeks to provide a detailed portrayal of events and phenomena within a specific reality. Table 1 summarizes the classification of the work presented by this article.

Table 1. Main classifications of the work carried out and presented by this article.

Main classifications of the work presented by this article	
Method	Deductive
Main Aspects	Bibliographic Research, Analytical Analysis, and Computer Simulation
Research Type	Quantitative
Nature Type	Applied
Time Aspect	Transversal-Time Development
Technical Procedure	Objective-Descriptive

III. IMPORTANT THEORETICAL ASPECTS

In general, and simplified terms, the modal behavior of a structure discretized using the Finite Element Method is governed by matrix equation below, [3].

$$[\mathrm{M}]\{\ddot{d}\} + [\mathrm{K}]\{d\} = 0 \qquad (1)$$

This equation represents the free motion of the system - that is, the case in which no external force acts upon it. In the expression, $[\mathrm{M}]$ represents the global mass matrix; $[\mathrm{K}]$ represents the global stiffness matrix; $\{d\}$ represents the nodal displacement vector; and $\{\ddot{d}\}$ represents the nodal acceleration vector. The vibration problem can be solved by assuming that the system oscillates harmonically. To do this, the following solution is adopted

$$d(t) = d'e^{i\omega t} \qquad (2)$$

where d' represents the time-independent mode shape, i represents the standard imaginary unit given by $i = \sqrt{-1}$, and ω is the natural frequency. Differentiating equation (2) twice with respect to time yields

$$\ddot{d}(t) = d'(-\omega^2)e^{i\omega t} \qquad (3)$$

substituting (2) and (3) into (1), we obtain

$$-[\mathrm{M}]\omega^2 d'e^{i\omega t} + [\mathrm{K}]d'e^{i\omega t} = 0 \qquad (4)$$

putting in evidence

$$e^{i\omega t}([\mathrm{K}] - \omega^2[\mathrm{M}])d' = 0 \qquad (5)$$

$e^{i\omega t}$ is not zero, so the equation becomes

$$([\mathrm{K}] - \omega^2[\mathrm{M}])d' = 0 \qquad (6)$$

This is the governing equation of the dynamic system considering free motion. Logan [3] shows that it has a non-trivial solution (i.e., $d' \neq 0$) only when the determinant of the matrix constituting it is zero; thus,

$$|[\mathrm{K}] - \omega^2[\mathrm{M}]| = 0 \qquad (7)$$

This is, therefore, an eigenvalue and eigenvector problem. From this equation, the values of ω - representing the system's natural frequencies - are obtained; for each frequency found, there is an associated vector d' corresponding to the system's vibration mode. In systems with complex geometries, it is common to use computational tools to find approximate solutions. As stated before, Ansys® software is used to discretize geometry, apply boundary conditions, and assemble the matrix formulation of the global problem. Thus, solving the eigenvalue and eigenvector problem becomes much simpler and faster, allowing the researcher to focus on correctly defining the model, its discretization, material properties, boundary conditions, and the critical interpretation of the results.

IV. RESULTS AND DISCUSSIONS

Figure 2 shows the modeling of the blade and disk analyses in Ansys®. Table 2 presents the main material properties used in the computational simulations.

The engine's operating range lies between 22,520 rpm and 28,150 rpm, serving as the reference for evaluating the natural frequencies obtained from modal analyses. Subjecting the blade of the first stage to modal analysis yield the vibration modes and their respective natural frequencies, as presented by Figure 3.

Similarly, by subjecting the first disk stage to modal analysis, the vibration modes and corresponding natural frequencies were obtained, according to Figure 4.

To avoid operation near resonance, a minimum separation is required between the natural frequencies and the most significant excitation frequency - namely, the rotational speed of the rotating assembly itself. A widely used criterion dictates that the natural frequency must remain outside the range of 0.8 to 1.2 times the excitation frequency. For every case, there is no single number defining this margin; rather, it is a concept of separation. In some rotating mechanical systems, for instance, the adopted margin ranges from 20% to 30%. In this study, a 30% separation margin is adopted to mitigate potential variations associated with numerical modeling, boundary conditions, and operational changes in the system.

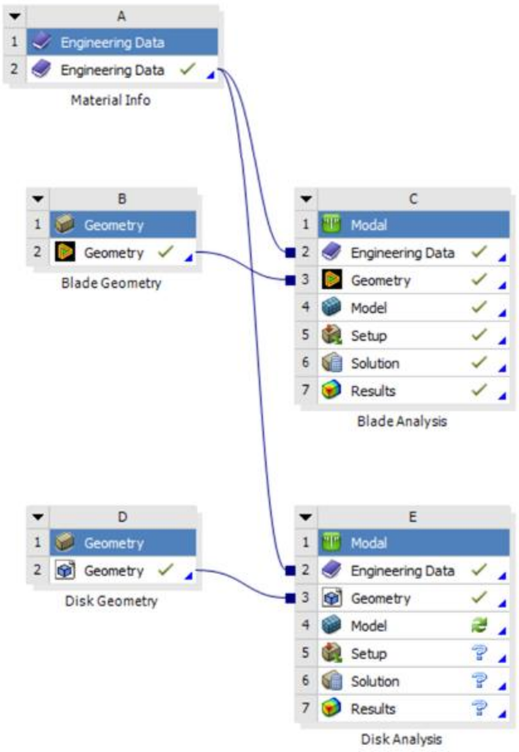

Fig.2. Modeling of the blade and disk analyses in Ansys® Workbench.

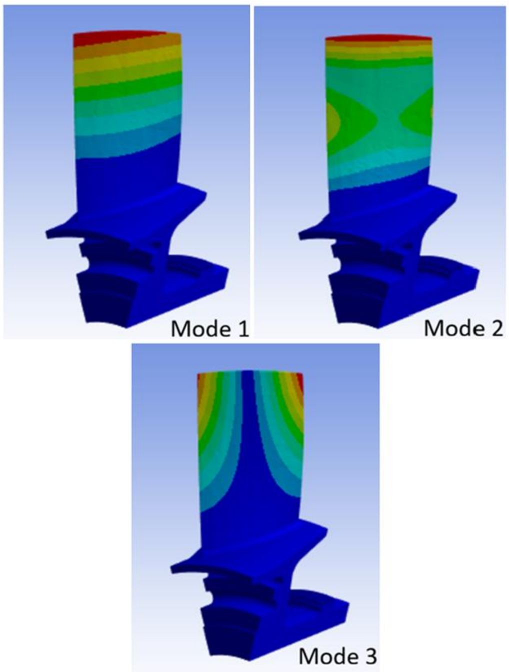

Fig.3. Blade modal analysis: Mode 1 @ 718 Hz; Mode 2 @ 2,946 Hz; and Mode 3 @ 2,958 Hz.

Table 2. Main material properties used in computational simulations.

Material Properties	
Alloy	Al 7178-T6
Young's Modulus	71.7 GPa
Density	2,770 kg/m3
Poisson's Ratio	0.33

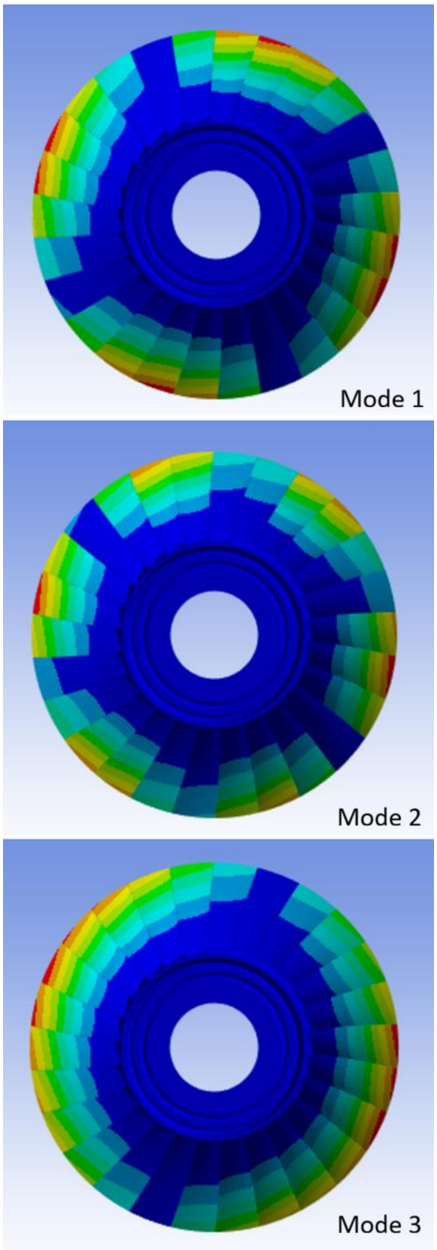

Fig.4. Disk modal analysis: Mode 1 @ 694 Hz; Mode 2 @ 703 Hz; and Mode 3 @ 708 Hz.

As stated before, the defined operating range for the aeronautical gas turbine is 22,520 to 28,150 rpm. Tables 3 and 4 present the natural frequencies obtained for the blade and disk of the first stage axial compressor, respectively, along with the percentage separation margins values calculated based on the most critical condition within the operating range - that is, the highest synchronous excitation frequency corresponding to 28,150 rpm.

Table 3. Natural frequencies obtained for the blade along with separation margin.

Natural Frequency	Separation Margin
(1) 718 Hz	53%
(2) 2,946 Hz	528%
(3) 2,958 Hz	531%

Table 4. Natural frequencies obtained for the disk along with separation margin.

Natural Frequency	Separation Margin
(1) 694 Hz	48%
(2) 703 Hz	50%
(3) 708 Hz	51%

It can be observed that, for all natural frequencies analyzed, the percentage separation margins remain above 30%, indicating that the natural frequencies are sufficiently far from the fundamental synchronous excitation. The other axial compressor stages, considering blades and disks, can be analyzed in the same manner.

V. CONCLUSIONS

The overall objective of this work was achieved through the calculation and analyses of the natural frequencies and vibration modes of the blade and disk of the first stage of an axial compressor for a 5-kN thrust aeronautical gas turbine. The results obtained demonstrate that the methodology adopted is consistent and technically adequate. To ensure the reliability of the results, the software was calibrated using simple beams for which analytical solutions exist. After verifying excellent agreement between the analytical solutions and the numerical computational results provided by Ansys® for simple beams, the same methodology was applied to the application under study, which features complex geometries. The results obtained for the blade and disk of the axial compressor first stage indicated that the calculated natural frequencies are sufficiently far from the fundamental synchronous excitation frequency. It is also expected that the separation margins found will be even greater when the gas turbine is in operation, as there will be an increase in the stiffness of the blades and disks associated with the arising of centrifugal stresses. The vibration modes obtained, in turn, exhibit characteristics compatible with the expected behavior, indicating physical coherence and consistency of the results. As a contribution, this work reinforces the importance of modal analysis as a fundamental step in defining the design and evaluating the structural integrity of aeronautical turbines, especially about preventing failures associated with vibratory phenomena.

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