Simulation and Analysis of Dual Unbalanced Rotor Effects on Natural Frequency in a Digital Twin Shaft Model

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Abstract

Blades are one of the basic components of a gas turbine and its main function is to rotate the shaft associated with the generator motor. Gas turbine model MS9001E used power plants at south Baghdad station, the blades are subjected to harsh working conditions such as high vibration, temperatures and pressures, thus highlighting the importance of studying the materials used in Manufacture of blades that work under harsh operating conditions. In this research, stress, strain and deformation produced by the centrifugal force that the blade is subjected to be studied, as well as studying the natural frequencies of the blades. Three-dimensional was created through the program solidwork 2018 and then exported to the program ansys 2019 for analyzing. Two alloys of materials (GTD-111) and (IN-738) were analyzed and compared between them, and the results showed that alloy (GTD-11) is the best and is suitable for use in the manufacture of blades.

Keywords

Rotors, Natural frequency, Deformation, vibrations frequency, Modal analysis

1. Introduction

The dynamic rotor plays a crucial role in the behavior of rotary machines, ranging from large-scale systems like power plant rotors and turbo-generators to smaller systems such as tooth drills, pumps, and air compressors [1, 2, 3]. Understanding the history of rotor dynamics is essential as it highlights the fundamental challenges in developing and implementing rolling bearings for various applications [4, 5, 6, 7, 8] when stability and quality must be assured [9]. The study of dynamic behavior in rotating machinery began in the early years of the 19th century when the industrial revolution increased the demand for analyzing rotational motion also in the robotic industry [10, 11, 12, 13]. Since the fifties, numerous researchers have conducted studies on crack propagation in shafts, and some of these findings have been extended to real-world rotors, providing valuable insights for designers [14, 15, 16]. Typically, rotors operate under cyclic pressure, making them susceptible to various operational issues, such as fatigue cracks. These cracks tend to occur when the rotors' natural frequencies and critical speeds increase as the shaft length decreases and the cross-sectional area increases [17]. It is crucial to identify the vibration characteristics of cracked shafts to develop a control system that can detect operational errors and early-stage cracks [18, 19] and prevent sudden accidents. Numerous researchers have focused on investigating the impact of cracks on the efficiency of rotating shafts. Some have conducted analytical analyses to study these issues, while others have approached them approximately. One key factor in minimizing undesired vibrations is effectively controlling the rotor's geometric imbalance [20, 21, 22]. By employing calculations and understanding the mass of unbalance during rotation, it is possible to measure the vibration response of any system. Multiple researchers have conducted studies on the impact of externally applied axial force and torque on the lateral vibration of shafts. Alaa et al. [23] derived the equation of motion for a flexible rotating shaft subjected to a constant compressive axial load by incorporating gyroscopic moments consistently. In the study [24] examined the stability of a rotating cantilever shaft carrying a rigid disk at its free end, considering follower axial force and torque loads. Chen and Sheu [25] analytically investigated the stability behavior of a rotating Timoshenko shaft with an intermediate attached disk under longitudinal force, providing frequency equations for various boundary conditions and numerically determining critical axial and follower forces. Chen et al. [26] The influence of inertial forces on shafts and beams can result in axial stresses. Researchers have also examined the rotation of beams around an axis perpendicular to their beam axis, where centrifugal force directly induces axial stress in the beam. In this study [27] utilized the dynamic stiffness matrix for an Euler-Bernoulli beam subjected to axial force to analyze the vibration of rotating uniform and tapered beams. In the investigation reported [28, 29] derived the governing equation for the linear vibration of a rotating Timoshenko beam, considering the coupling between extensional and flexural deformation by linearizing the fully geometrically nonlinear beam theory. They

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proposed a power series solution method to determine the natural frequency of the rotating Timoshenko beam. This study introduces a new phenomenon that can influence the performance of rotating shafts. The change in natural frequency is attributed to the position of rotors and the midspan between double rotors. Unlike previous studies, the disks located equal spaces between fixtures and rotors. To accomplish this objective, the lateral deformation and natural frequency need to be calculated, then evaluating the impact of this distance on the lateral natural frequency of the shaft. The paper presents a numerical solution to evaluate the systems.

2. Materials and Method

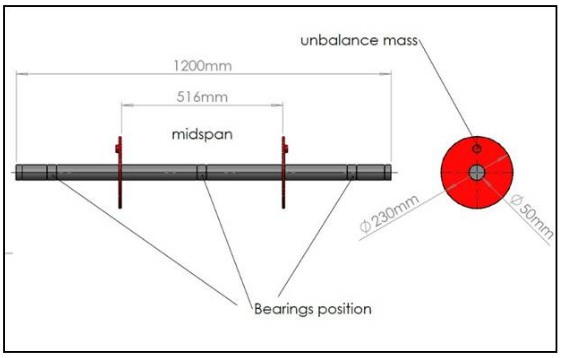
A modal analysis was conducted on a structural steel shaft, and commercially available rotors were used. The material properties of the structural steel are provided in Table 1. Finite element analysis (FEA) was chosen as it offers more comprehensive results compared to experimental studies, with the added benefits of speed and cost-effectiveness [30, 31, 32] also in term of missing data reconstruction or imputation [33, 34]. The FEA employed a finite element discretization approach to solve complex structural equations by dividing the structure into specific finite elements[31, 32, 35, 36]. The unbalanced rotors were designed using FEA, utilizing a mesh system composed of interconnected nodes. The model, created in SOLIDWORKS 2020, was exported to ANSYS 2019 software. The ANSYS model then meshed, and boundary conditions were applied. The software solved the system equations to determine the model's natural frequencies.

Table 1
Material Properties of Rotating shaft and Rotors

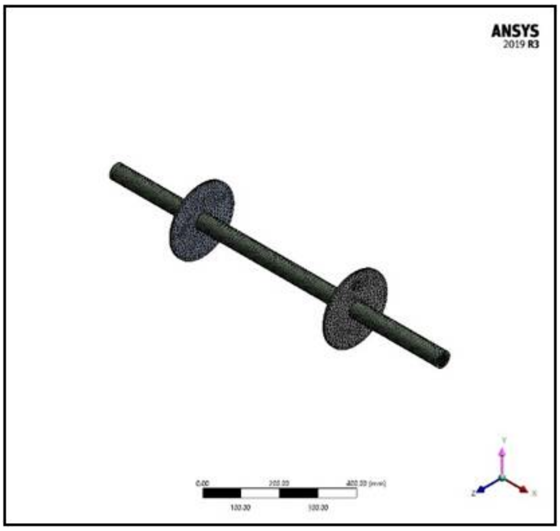
PROPERTIES	VALUE
Specific Heat (J/Kg K)	485
Young's Modulus (GPa)	210
Density (kg/m3)	7850
Poison's ratio	0.30
Thermal conductivity (W/mk)	60
Thermal expansion (oC)	14×10^{-6}
Yield strength (Mpa)	450
Shear Modulus (GPa)	80

3. Modelling and Analysis of Rotating shaft and Dual Rotors

The model developed aims to simulate rotating machinery with unbalanced rotating components. The model consists of two main components to simplify the system: the rotors and the shaft. The shaft has a length of 1200mm, while the rotors have diameters of 230mm, as shown in Figure (1a). The bearings are positioned at three locations, with the midspan position being particularly interesting for identifying the optimal position. The system's geometry was created using SolidWorks 2020 and then exported to ANSYS 2019 software. The rotor system geometry was discretized in the initial stage using tetrahedral elements. The model comprises 160,022 nodes and 40,196 elements, as depicted in Figure (1b).



(a)



(b)

Figure 1: a. The geometry of rotor system in SOLIDWORKS, b. Meshed model using ANSYS

The geometry of the rotor is affected by the unbalanced mass and the rotating velocity of 200 RPM in ANSYS Modal. The boundary conditions shown in Figure (2), the two ends of the shaft are fixed support and the position of bearings are BEARING boundary conditions in ANSYS Model. The geometry of the rotor and shaft are meshed with ELEMENT 186 to adept to different shapes.

ANSYS Modal analysis uses the following equations

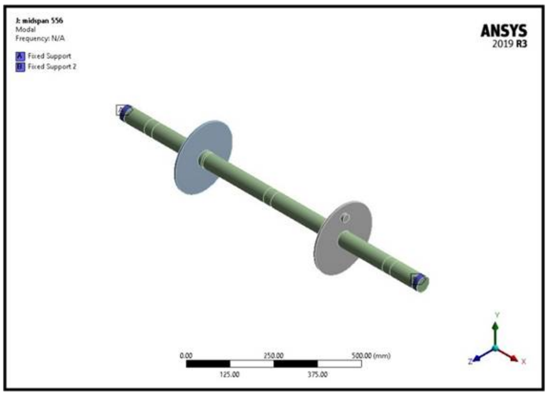


Figure 2: Boundary conditions in ANSYS.

to solve vibration problems:

1. Mass Matrix Equation: $[M]\psi + [K]\psi = 0$

2. In this equation: [M] is the mass matrix, which represents the distribution of masses in the system. ψ is the vector of mode shapes or modal displacements. [K] is the stiffness matrix, which represents the stiffness of the system.

3. Eigenvalue Equation: $[K]\psi = \lambda[M]\psi$

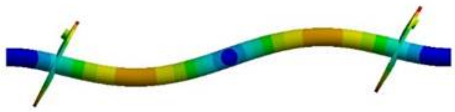
4. In this equation: λ represents the eigenvalues, which determine the natural frequencies of the system. [K] is the stiffness matrix. ψ is the vector of mode shapes.

By solving the above equations, ANSYS Modal analysis calculates the natural frequencies (eigenvalues) and corresponding mode shapes (modal displacements) of the system.

4. Results and Discussion

The outcomes derived from ANSYS 2019R3 depend on various elements such as the system's shape, length of the shaft, elastic characteristics, rotor spacing, and boundary conditions. These factors impact the inherent frequency of the system. A higher inherent frequency signifies an improved design by reducing vibration amplitude and lowering the likelihood of component malfunction [37]. The modal analysis offers a valuable understanding of these outcomes, aiding in assessing and enhancing the system's performance. Figure (3) shows the mode shapes for the first five modes for the 916mm midspan. Deformation (11.69 mm) at a frequency (162.43Hz) 1st mode, and deformation (12.22 mm) at a frequency (418.48Hz) 2nd mode. The following three modes show an increase in frequency, and deformation will reach the peak at 4th mode for the 3rd ,4th, and 5th modes; the frequency and deformation are 569.65Hz, 35.22mm; 601.04Hz, 35.74mm; and 648.08Hz, 29.22mm respectively.

(a) 1^{st} mode 162.43Hz



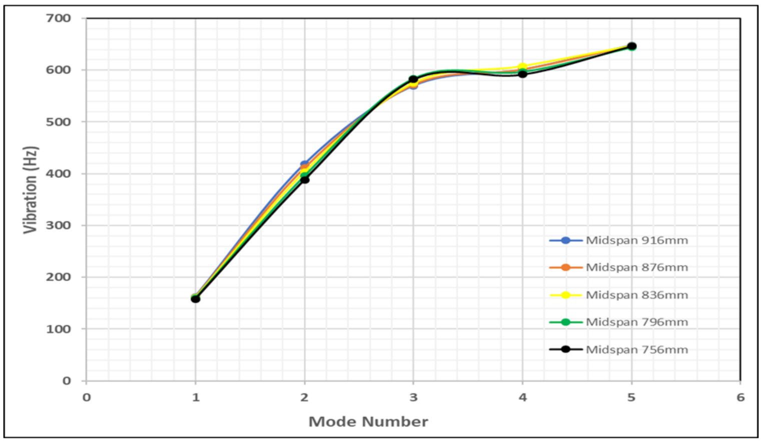
(b) 2^{nd} mode 418.48Hz

(c) 3^{rd} mode 569.65Hz

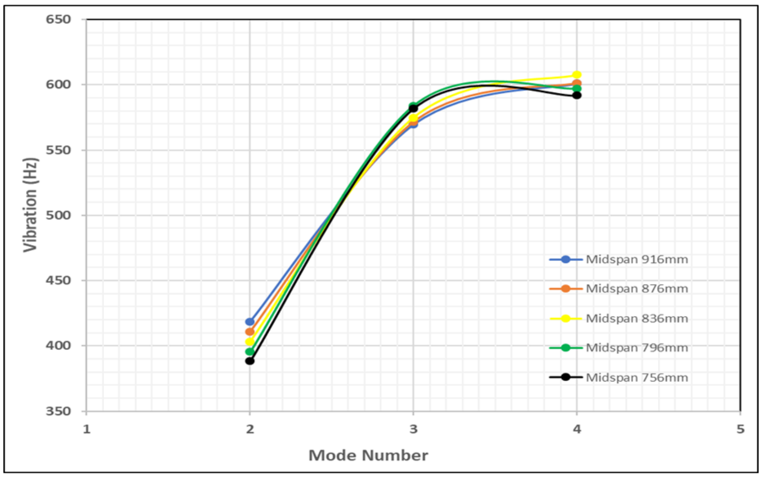
(d) 4^{th} mode 601.04Hz

(e) 5^{th} mode 648.08Hz

Figure 3: The first five mode shapes of rotor midspan 916mm which are extracted by ANSYS19, are labeled (a) to (e) respectively



(a)



(b)

Figure 4: Shape modes and the Natural frequency in (Hz).

Figure (4a) shows the change in natural frequency of the unbalanced rotating shaft with the mode shape number. It is noticed that the 1st and 5th modes are having same value, the change in natural frequency occur in the other three modes. Figure (4b) gives a lock at the three modes 2nd,3rd, and 4th, for midspan 916mm gives highest frequency in the second mode while lowest frequency for the midspan 756mm for the same mode. The third mode midspan 796mm and 756mm approximately have the same frequency about 570 Hz and the 796mm midspan has same value with 756mm midspan. 836mm midspan has a value between them. 4th mode midspan 756mm has the lowest frequency 591 Hz and the mid span 836mm has the highest value 607Hz.

5. Conclusions

The following conclusions can be derived from the findings of this study:

1. Using midspan at the highest value increase the value of natural frequency and reduces the effect of whirling.
2. The position of the rotors does affect much on the rotating shaft's natural frequency because of the deformation in the rotating disk, which has the same dimensions.
3. The second mode is affected by the position of rotors which give high natural frequency at a high midspan value.
4. As well as the distance between two rotating discs is higher, the system balance increases too.

In future works, new optimization methods based on neural approaches will be explored [38, 39].

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