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Research Article

Experimental and Numerical Modal Analysis of a Bladed Rotor

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ABSTRACT

The behavior of an asymmetric bladed rotor was investigated in this study. The bladed rotor which performs solid-solid, liquid-liquid, liquid-solid separation processes is part of the decanter machine and named as screw conveyor. The purpose of this study is to determine the dynamics of the rotor on the free conditions for future development of the asymmetric and blade assembled rotors. It is important for the designer to determine the natural frequencies of the bladed asymmetric rotor so that some precautions can be taken during the design of the machine and in the operating conditions of the machine. This study consisted of two parts. In the experimental one, the modal test of the bladed rotor was performed and in the numerical part, the modal analysis of the rotor was carried out under free-free boundary conditions using a simulation program based on the finite-element method. The natural frequencies of the blades and the rotor were obtained experimentally and numerically. It was found that the experimental results and numerical results were in good agreement. Besides, it was concluded that the two mode shape was equal to the transverse mode frequency of the rotor and blades and the following mode shapes correspond to the bending and torsion mode frequencies of the blades. Since the shape properties of the rotor and blades are different, it has been observed experimentally and numerically that their natural frequencies also differ.

Introduction

The decanter is used in various fields to separate components of different densities. The machine consists of a bowl rotating at high-speed, a scroll conveyor rotating at a different speed on the same axis with the bowl, a drive group adjusting the speed difference, and a body carrying rotating elements. The present work focused on the dynamic characteristic properties of the screw conveyor which was composed by a rotor with a large number of blades assembled in its circumference. Ewins carried out the pioneering experiments related to the bladed disk and presented the frequency response functions of it in 1969 [1]. Srinivasan presented an in-depth study to improve our understanding of the vibratory behavior of gas turbine blades in 1997 [2]. Rzadkowski *et al*. (2013) investigated free vibration analysis of mistuned aircraft engine bladed disks numerically. They showed mistuned mode shapes of the bladed disk using a simulation program [3]. Kaneko *et al.*(2013) studied the frequency response analysis of a bladed disk for the symmetric and asymmetric vane spacing [4]. Fernandes *et al.* (2016) studied a 3-D static fracture analysis of the gas turbine compressor blade. They determined the natural frequencies and mode shapes for both a single blade and a bladed disk system [5]. Repetckii *et al.* (2018) presented the results of experimental and numerical vibration analysis of the bladed disk. They investigated the effects of various types of mistuning on the free vibrations of the bladed disk [6]. In this paper, this investigation presented the modal properties of the decanter screw conveyor. Many experimental investigations have been carried out and mathematical models have been proposed for the dynamics of the screw conveyor. Jin *et al*. (2004) analyzed the first three modal parameters of the centrifuge and screw conveyor under different support conditions. They presented comparisons between the experimental results and the numerical results [7]. Yang *et al.* (2008) carried out the numerically static analysis and prestress modal analysis for the different diameter, rotational speed, leaf thickness, and other design parameters of the scroll conveyor by a numerical method. They showed the effects of design parameters on the natural frequencies of the conveyor [8]. Wang *et al.*, (2010) analyzed the horizontal decanter centrifuge conveyor using a commercial program. They showed the effects of spiral leaves and conical angles on the dynamics of design [9]. Donohue (2014) investigated the dynamics of decanter's rotors. They presented that the effects of the velocity difference between the bowl and the screw conveyor on the dynamics of the decanter [10]. Hua *et al.* (2015) analyzed the effect of the relative speed and the frictional force on the bearing of the rotor system. The nonlinear effects of bearing stiffness were taken into account [11]. Liu *et al.*(2015) simulated the fluid-solid interaction for decanter using a numerical method. They presented the effects of the centrifuge and hydrostatic pressure on the scroll conveyor and as a result of this effect, critical regions in design were obtained [12]. Tan *et al.* (2015) examined numerically the vibration characteristic properties of the decanter, which was a dual-rotor system. Their target was to simulate decanter working conditions. They performed the harmonic and transient analysis by simulating a ball bearing connection with the commercial program [13]. Jiayi and Luo (2018) carried out experimentally and numerically the dynamic analysis of the turbine disk under free-free conditions. They presented the results of the natural frequencies [14]. There are many experimental and numerical studies examining the vibrations of asymmetric rotors and decanter screws. However, the effects of the blades on the decanter on the response of the system are discussed for the first time in this study.

In the present study, the modal parameters of the bladed rotor of the decanter experimentally and numerically were investigated. In the experimental analysis, the modal analysis of the bladed rotor was carried out and the modal parameters such as natural frequencies were presented under free-free boundary conditions. In the numerical analysis, the natural frequencies and the mode shapes of the rotor were presented. The natural frequencies of the rotor and blades obtained numerically and experimentally were compared. The numerical and experimental results obtained were in good agreement.

Experimental Modal Analysis of the Bladed Rotor

The principal equipment of the experimental setup consists of a data collector (Bruel & Kjaer-3050-B-040), an impact hammer (Bruel & Kjaer-type 8202), an accelerometer (Bruel & Kjaer-Type 4533-B), a decanter screw conveyor and a computer is presented in Figure 1. The shape of the freely supported conveyor produced commercially by the private company can be seen in Figure 2. The rotor is constituted of a main pipe and blades bonded to the pipe. The pipe and blades are joined helically by the welding method. The main geometrical parameters of the rotor are listed in Table 1. The bladed rotor is hanged on the frame elastically to provide free-free boundary conditions. The accelerometer is rigidly connected to the rotor surface and the acceleration is measured on the y direction of the rotor. The bladed rotor is excited by means of the impact hammer from a total of 34 points marked in the x direction from each blade surface to a point and three points on the rotor surface between the blades, in the y direction. The hammer is hit five times in the same spot to verify the test. There are different tips for the impact hammer to excite the system, from soft rubber to metal. The appropriate tip should be chosen for the impact hammer so that the accelerometer can measure on the structure. While making this decision, the design and material of the structure should be taken into consideration. Since the rotor considered in this study is made of metal and has thick sections, the metal tip is chosen for the impact hammer. The generated analog data is transferred to the Brüel & Kjaer data collector operating with the Pulse LabShop software, which converts it to a digital signal. The frequency analyzer properties, response signal and excitation signal properties are written in Table 2 and the schematic diagram of the experimental analysis is presented in Figure 3. The frequency-response functions (FRFs) of the measurement points are presented. The modal parameters of the rotor and blades are determined from the FRFs of the points based on "PolyMAX" or polyreference leastsquares complex frequency-domain method. This method provides clean stabilization diagrams from the FRFs [15]. The experimental natural frequencies are presented in the results section.

Table 1. Structure geometrical dimensions for the bladed rotor

Parameter	Value(mm)
Pipe inner radius	89
Pipe outer radius	128
Scroll pitch	60
Blade angle	3~5 degree
Total Length of pipe	566

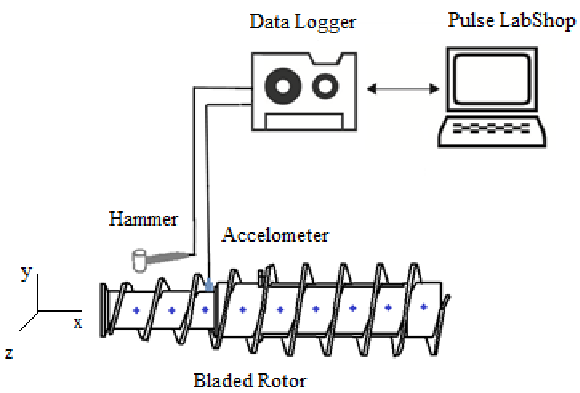


Figure 1. Experimental Setup of the Digital Measurement System

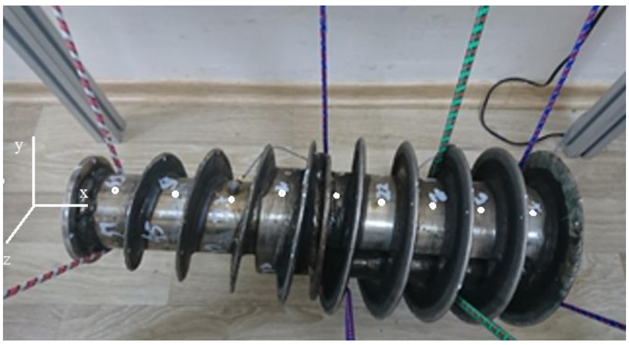


Figure 2. The Asymmetric Bladed Rotor with Free-Free Boundary Conditions

Table 2. The Technical Set-up Properties of Experimental Modal Analysis

Frequency analyzer properties	
Lines:	3200
Span:	3.24kHz
Averaging domain:	Spectrum averaging
Averaging mode:	baseband
Analysis mode:	Linear-5 averages
Time weighting window	uniform

Response signal	
Sensitivity:	982.7µV/m/s^2
Filters:	7 Hz (High pass)
Max peak input	10V-10.18 km/s^2

Excitation signal	
Time weight:	Uniform (window)
Channel sensitivity:	2.27mV/N
Max peak input:	10V 4.405 kN
Filters:	7 Hz (0.1dB)

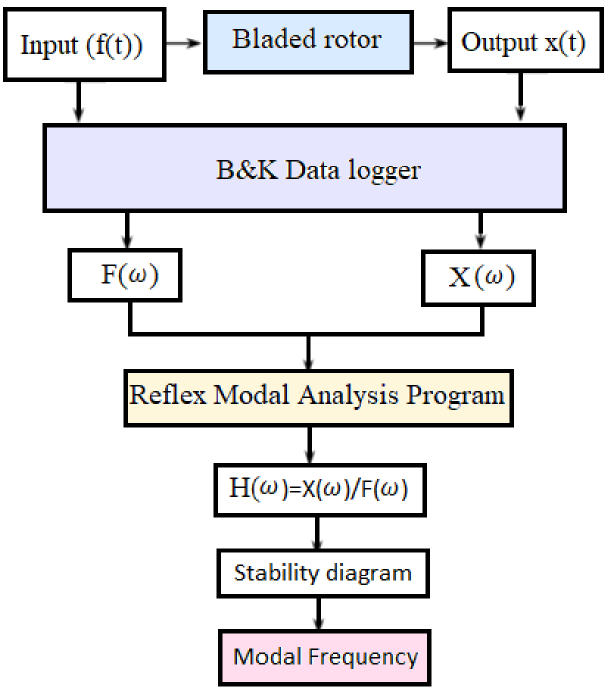


Figure 3. Schematic diagram of Experimental Modal Analysis

Numerical Modal Analysis of the Rotor

The modal analysis of the rotor is performed under free-free conditions using a commercial program (ANSYS). The three-dimensional geometry of the bladed rotor is shown in Figure 4. The mechanical properties of the rotor are given in Table 3. The rotor and blades are meshed by quadratic elements that have 20 nodes and the model is consisted of 770558 nodes and 1203906 elements to achieve grid convergence (Figure 5). Since the screw conveyor under investigation has oval and circular lines, modeling with the quadratic elements represents the structure better than the linear elements. In similar studies in the literature [16-17], in which experimental and numerical analyzes of modal analysis are compared, it is seen that the results are compatible with quadratic element modeling. In the light of references, the conveyor is modeled and analyzed with quadratic elements in this study. Since the minimum thickness of the leaf parts of the structure is 4mm, the element size in the finite element mesh has not been reduced below this size.

Each node has three degrees of freedom in the directions of x, y and z. The bonded contacts of the geometry are modeled by MPC (multi-point constraint) formulation. The Block Lanczos algorithm is used to solve the natural frequency problem. The natural frequencies and mode shapes of the bladed rotor are determined and presented in the results section.

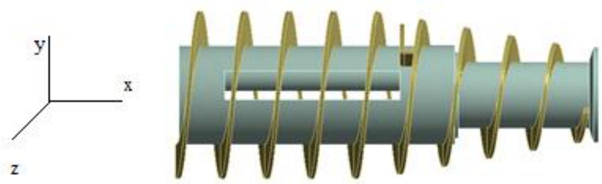


Figure 4. The Geometry of the Bladed Rotor

Figure 5. The Asymmetric Bladed Rotor Meshed Model

Table 3. Physical and Mechanical Properties for the Rotor

Sample	Value
Materials	AISI 304
Young's Modulus	193 GPa
Poisson's Ratio	0.25
Density	8.00 g/cc
Yielding Stress	205 MPa

Experimental Results

The results of the FRF analysis obtained from 34 measurement points on the rotor are presented in Figure 6. This function presents the acceleration values corresponding to the frequency values of the conveyor.
The stability diagram is obtained to determine physically mode frequencies. The PolyMAX estimator is one of the modal parameter estimators and this estimator yields very clear stabilization diagrams [18]. The estimator is applied to determine modal parameters from stability diagram provided by using Reflex Modal Software, and parameter estimation results are shown in Figure 7, where the symbol '∇' stands for frequency and eigenvector stable, '×' for frequency stable, '*' for frequency and damping stable, and '◇' for all stable for frequency estimation under the stability criteria [19].
The experimental natural frequencies and damping ratio of the bladed rotor are determined from this diagram. The first nine natural frequencies of the bladed rotor obtained experimentally by the PolyMax method are written in Table 4. The damping ratio of the rotor for the first mode determined by the PolyMax method is 0.088. The first two peaks marked in Figure 7 (a) represents the first two-mode frequencies of the rotor, while the following peaks marked in Figure 7 (b) represent the mode frequencies of the blades. Further, it is observed that the vibration peaks of the blades have high amplitudes and they are close to each other in Figure 7 (b). These peaks are caused by differences of blades' properties which are physical properties and boundary conditions. In the literature, the problem of feature differences in bladed structures is commonly called mistuning behavior [20]. This behavior has an important effect on the vibration behavior of the system. It can cause the spatial location of vibration energy around a limited number of blades and the system's amplitude and stress increase.

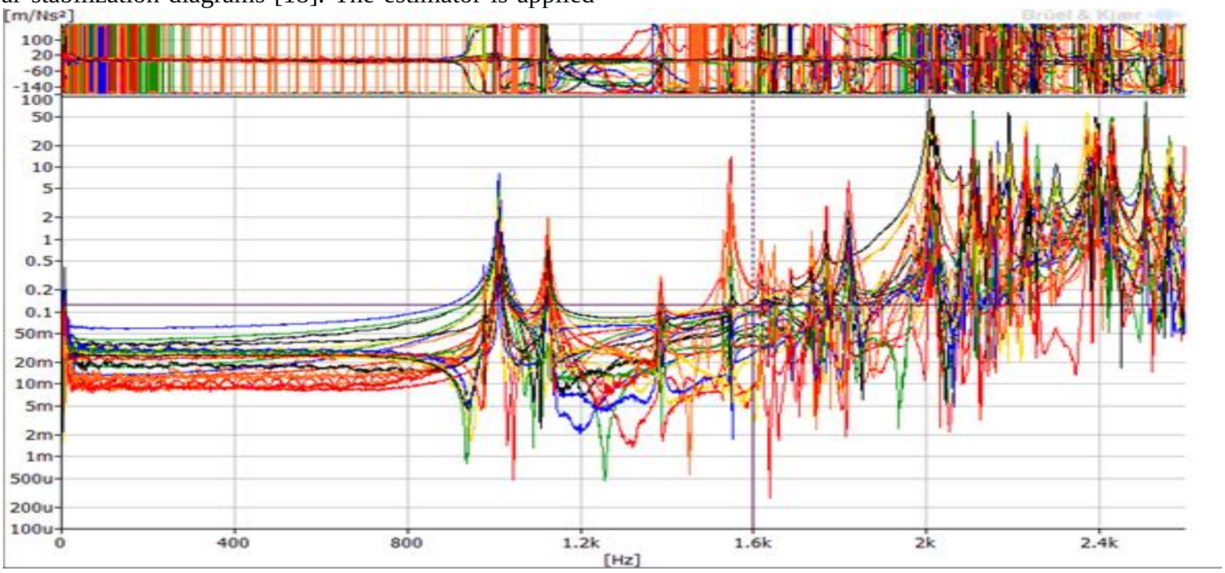


Figure 6. Frequency Response Functions for the Bladed Rotor

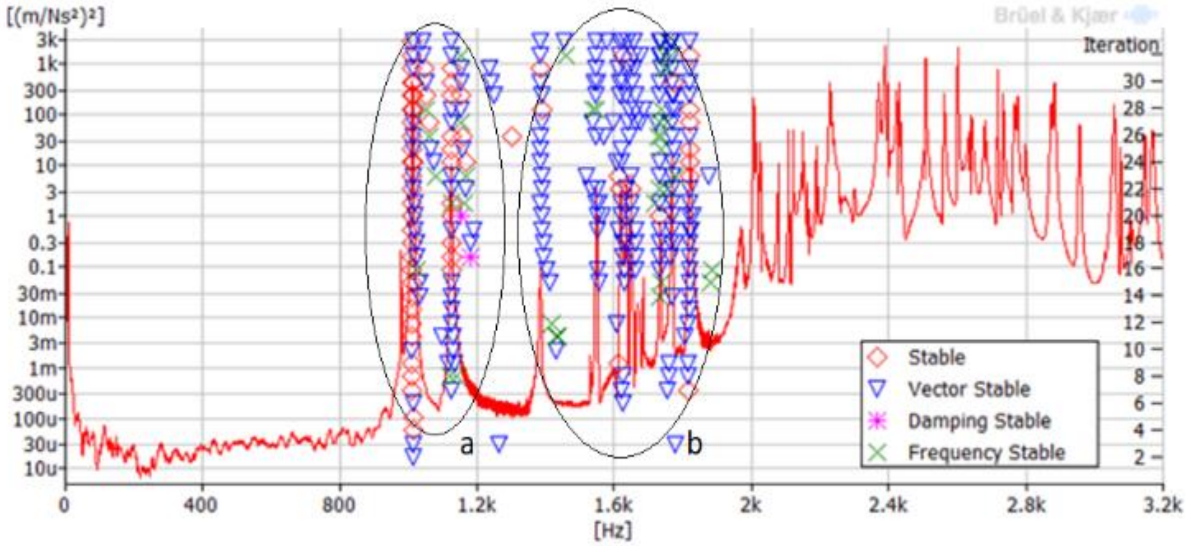


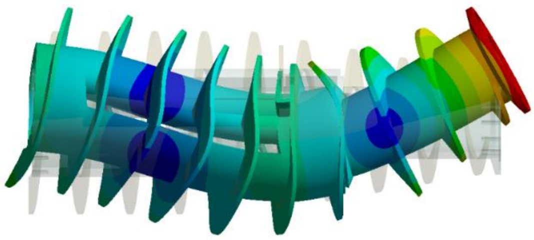
Figure 7. Stability Diagram of the Bladed Rotor
(a)The Natural Frequencies of the Rotor, (b) The Natural Frequencies of the Blades

Table 4. The Experimental Natural Frequencies of the Bladed Rotor

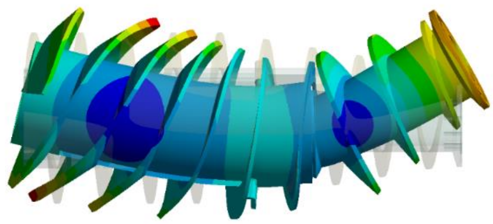
Mode Number	Experimental Natural Frequencies (Hz)
1	1012
2	1124
3	1385
4	1548
5	1648
6	1686
7	1740
8	1761
9	2006

Numerical Results

The numerical modal analysis is presented, including the natural frequencies and mode shapes of the bladed rotor. The mode shapes of natural frequencies of the bladed rotor are given in Figures 8-9. The behavior of the rotor is considered between 0 and 2100 Hz. In Figure 8 (a) and (b), the bending mode shape and natural frequency of the rotor are shown. The blades do not vibrate at 1018.6 Hz (Figure 8 (a)), while, both the rotor and the blades vibrate at 1127.7 Hz (Figure 8 (b)). In Figure 9, the mode shapes and natural frequencies of the bladed rotor are shown. When the mechanical properties of the blades are slightly different from each other, the resonance frequencies are also slightly different from each other. This phenomenon is referred to as the mistuning problem and there are theoretical and experimental studies on the mistuning problem of bladed systems in the literature [1, 3-6]. In Figure 9 (a) and (b), the first two blades vibrate only. In Figure 9 (c-g) the transverse mode shape and natural frequencies of the first five blades are shown. Since each of the blades has slightly different physical properties, each of the blades has slightly different amplitudes. It is observed that the blades vibrate at higher frequencies, as the widths of the blades at the right end of the rotor are smaller. The designer needs to determine the nature of the bladed asymmetric rotor so that some precautions can be taken. As seen from the numerical and experimental results, many blade structures that resonate over a wide frequency range. The designer must determine the rotational frequency of the machine taking into account the resonance frequencies of the system. Moreover, during the design phase of the blades, optimization studies can be carried out to shift the resonance frequencies. Thus, safe working areas where the machine can work can be provided.

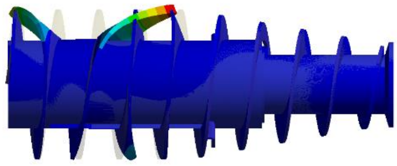


(a)1018.6 Hz

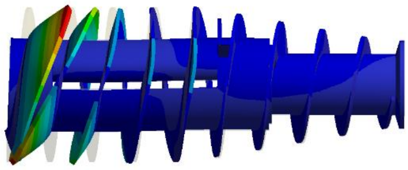


(b) 1127.7 Hz

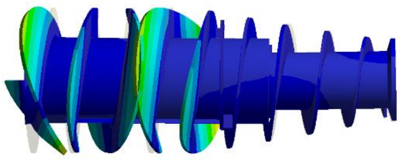
Figure 8. The Mode Shapes of the Bladed Rotor



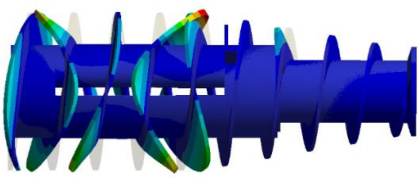
(a) 1388.7 Hz



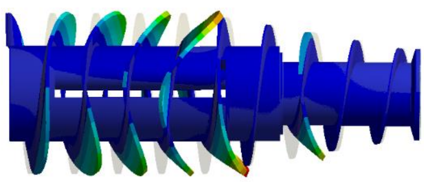
(b)1578.4 Hz



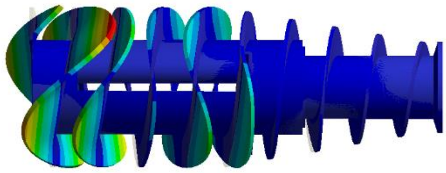
(c) 1654.1 Hz



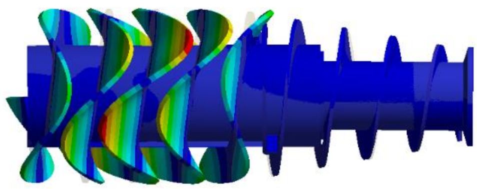
(d) 1687.8 Hz



(e) 1756.1 Hz



(f) 1774.1 Hz



(g) 2010.7 Hz

Figure 9. Mode Shapes of a Mistuned Blades

A Comparison between Numerical and Experimental Results

The experimental and numerical rotor models which have the same physical properties were analyzed under the same boundary conditions. The results obtained from the experimental modal analysis are compared to the numerical ones (Table 5). Relative difference represents the ratio of the difference of the natural frequencies between the experimental results and the numerical ones. The differences presented in Table 5 are less than 2 percent. These results prove that mesh optimization in numerical analysis is done correctly. From this point of view, it can be said that they are in good agreement. It is observed that the natural frequencies of the blades are close to each other such as 1124, 1385, 1548, 1648, 1689, and 1740 Hz. These results confirmed that the bladed rotor represents mistuning behavior. This effect is stated by the results of experimental and numerical analysis that may occur in the screw conveyor considered as bladed rotors.

Table 5. Comparison of Experimental and Simulation Results

Mode Number	Experimental Natural Frequencies (Hz)	Numerical Natural Frequencies (Hz)	Rate of Difference (%)	Mode Shapes
1	1012	1018.6	0.65	Rotor and Blade
2	1124	1127.7	0.33	Rotor and Blade
3	1385	1388.7	0.27	Blade
4	1548	1578.4	1.93	Blade
5	1648	1654.1	0.37	Blade
6	1686	1687.8	0.11	Blade
7	1740	1756.1	0.92	Blade
8	1761	1774.1	0.74	Blade
9	2006	2010.7	0.23	Blade

Conclusions

In summary, the dynamics of the bladed rotor under free-free boundary conditions were studied experimentally and numerically. In the experimental analysis, the FRFs of the bladed rotor were presented using the impact hammer test. In the numerical analysis, the mode shapes and the natural frequencies of the bladed rotor were obtained using the simulation program based on the finite element method. Various observations and findings have been obtained from the experimental and numerical analysis, as follows:

1. It can be considered as a screw conveyor consisting of a combination of a main pipe and blades.

2. It has been observed that the natural frequencies of the blades, which have slightly different mechanical properties from each other, are also close to each other.

3. There is a difference of 100 Hz between the first two bending modes due to the asymmetrical nature of the system consisting of blades and rotor.

4. It is observed that the mistuning effects on the response of the bladed rotor are realized and these effects increased as the width of the blade increased.

5. It is observed that participation in the mode shape increased as the width of the blade increased.

6. A FEA model has been developed for a decanter screw to calculate all nine modal frequencies and mode shapes.

7. The results of experimental modal frequencies obtained are within 2% error compared to the FEA prediction.

The present study presents the response of the screw conveyor considered as a bladed rotor experimentally and numerically.

Conflict of Interest- All authors declare no financial/commercial conflict of interest regarding the study.

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