

International Journal of Innovative Research in Science, Engineering and Technology

(An UGC Approved, High Impact Factor, Monthly, Peer Reviewed Journal) | Impact Factor: 7.089|

UGC Approval No: 48027

Vol. 6, Issue 12, December 2017

Design of Hammer Mill Shaft Based on Vibration Analysis Using Dunkerly's and Energy Equations

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ABSTRACT: Understanding the fundamentals of rotor dynamics and critical speed attainment is the aim of this paper. It has been noted that early attainment of the critical speed of the rotor-dynamic shaft is a problem faced by many industries. This vibration may cause noise and shaft failure. This paper presents the results of a FEM and conventional manual calculation used to determine the critical speed analysis of a non-uniform shaft used in a food processor for pulverizing spices. For any rotating machinery, it is not desired for the shaft to reach its critical speed too soon. It has been noted in literature that higher vibration amplitudes at critical speeds are the cause of failure for many rotating machines. An attempt is made to compare the modal analysis in ANSYS software with the theoretical natural frequency and critical speed of the multirotor shaft using the Dunkerly and Energy method. ANSYS software provides an estimate of the most fundamental critical speed at which the system may begin to vibrate at higher amplitude, potentially leading to a breakdown, based on theoretical natural frequency and critical speed calculations. It has been noted that the machine will not reach the critical speed and that no vibration will be produced within the working speed range of 500 rpm to 3000 rpm. To prevent reaching the critical speed for high-speed rotating machine applications, a few substitutes are proposed. Thus, this paper aids in understanding how to prevent rotating machines from reaching their critical speed too soon and aids designers in preventing numerous rotor dynamic phenomena that impair the operation of rotating machinery.

KEYWORDS: Hammer mill, Critical speed, Dunkerly's equation, FEM.

I. INTRODUCTION

Every day, rotor dynamics are commonly experienced without realizing it in pumps used in household appliances, washing machine drums, car engines, computer drives, spindles of machine tools, and airplane engines. A specialist area of applied mechanics that studies the dynamics and stability of rotating machinery is known as rotor dynamics. A rotor, according to ISO definition, is a body that is suspended via a series of cylindrical bearings or hinges that permit it to spin freely around a fixed axis in space. A stator is a non-rotating supporting structure. The vibration amplitude frequently reaches a maximum known as a critical speed as the rotation speed rises. Usually, the unbalance of the rotating structure excites this frequency. The rotational speeds in this instance match the natural frequency. Catastrophic failure may result from vibrations with an excessive amplitude at a critical speed.

Instability is another recurring phenomenon in rotating machinery. At specific velocity ranges, rotor behavior may become unstable. Occasionally, when the centrifugal field is operating, the vibrations' amplitude grows infinitely. The instability fields are the ranges of rotation speeds where self-excitation takes place. Catastrophic failure may arise from these so-called self-excited vibrations[2,3]. For this reason, precise rotor dynamic behavior modeling is essential to the

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design of rotating machinery in order to enhance process efficiency, extend machinery life, and facilitate safe operation in addition to improving product reliability. More sophisticated models are needed for accurate analysis of complicated machinery like gas steam turbines, compressors, pumps, grinders, mills, etc.

More precise and comprehensive rotor dynamic analysis is now feasible thanks to advancements in computer technology and numerical techniques like the Finite Element Method (FEM)[4]. Furthermore, due to the increasingly intricate and multi-scale designs that necessitate miniaturization and safe high-speed operation, it is imperative to investigate the impact of the surrounding medium on the rotor's dynamics.

Improvements in this field are ongoing, mostly as a result of the steady demand from the transportation and power-generating sectors. As a result of changes in materials science and engineering, rotating machinery is needed to operate for extended periods of time and is becoming lighter and faster. Because of all of these elements, fault identification, localization, and analysis are critical to the study of rotor dynamics.

Vibration is one of the main topics of interest in the condition monitoring of rotating machinery nowadays[5]. If a flaw appears and is not discovered, it could, in the best case scenario, be relatively minor and easily fixed; in the worst case scenario, it could cause costly damage and lost productivity, harm, or even fatality. Rotating machinery vibration can be measured and analyzed to identify and pinpoint significant flaws like mass unbalance, shaft alignment, rub, and cracked shaft vibration.

The speed at which the system's natural frequency and the natural frequency of the external force coincide is known as the critical speed. When there is no external load, all rotating shafts will deflect as they rotate. When an object rotates at a certain speed, called the critical speed, its unbalanced mass causes deflection that results in resonant vibration. The opening line of Dunkerley's paper, which was cited by Akpobi, Ovaworie(2008)[1], states, "It is well known that every shaft, however nearly balanced, bends when driven at a particular speed and, unless the amount of bending be limited, could possibly break, although the shaft runs true again at higher speeds." This specific speed, also known as the critical speed, is determined by the way the shaft is supported, its size and elastic modulus, and the sizes, weights, and locations of any pulleys it may be carrying.

Resonance may be present if the frequency of any harmonic component of a periodic forcing phenomenon equals or closely resembles the natural frequency of any vibration mode of the rotor. A speed is referred to as a critical speed if resonance occurs at a finite speed (API 613, 1995, 4th Edition). According to Wilson, Vibration, Griffin (London) 1959, pp. 155–160, "a speed at which vibrations occur of such character and magnitude as to be of real significance in relation to the life or function of the machine. "The system operates in a state of resonance when the synchronous rotor frequency equals the frequency of a rotor natural frequency. If the resonance is not critically damped, the rotor's response is amplified (API 684, August 20, 2005).

The critical speed definitions mentioned above aid in understanding that while resonance may occur within the working range, critical speed can be suppressed and exceeded with appropriate design. This aids in determining the fundamental critical speed approximation as well. Critical speed can be computed using two primary techniques: the Rayleigh-Ritz method and Dunkerley's method. The first natural frequency of vibration, which is estimated by both, is thought to be almost equal to the critical rotational speed. The natural frequency is generally overestimated by the Rayleigh-Ritz equation and underestimated by the Dunkerley equation. The equation Rayleigh-Ritz

$$N_c = \frac{30}{\pi} X \left(\frac{g}{\delta_{st}} \right)^{0.5}$$

Where,

Nc =Critical Speed

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g = gravitational acceleration ($\approx 9.81 \text{ m/s}^2$)
 δ_{st} = total maximum static deflection

The magnitude and location of the shaft unbalance, the shaft's diameter, length, and type of bearing support all affect critical speed. According to best practices, the maximum operating speed shouldn't be higher than 75% of the critical speed. What are the fundamental questions to comprehend vibration in rotating machinery, such as how many "critical speeds" can be determined? The rotating machine can function satisfactorily and suppress a few critical speeds. Resonance causes vibration amplitude to increase, which can have unintended consequences such as part loss and increased load on the bearings, shortening their lifespan. The terms "rotordynamic phenomena" are as follows: The gyroscopic effect causes the shaft and the components mounted on it to revolve around their axis.

Three analytical methods—the Rayleigh method, the Transfer Matrix method (TMM), and Dunkerley's method—for figuring out the lateral natural frequencies of a shaft rotor system were analyzed by P. H. Mathuria[2]. The natural frequency is evaluated and these methods are applied to a single hypothetical case. Program development is done via the Transfer Matrix Method. The natural frequency of a fiberize for the sugar industry or a crusher for the cement industry is carried out with the help of TMM in order to prevent shafting failures. Standard procedures in rotor design include calculating the bending critical speeds, evaluating the out-of-balance response, and choosing a suitable balancing procedure.

II. TECHNIQUES FOR DETERMINING A SHAFT'S CRITICAL SPEED FOR MULTIPLE LOADS

The estimation of the fundamental frequency is effectively illustrated by the Rayleigh method and the Dunkerley method

- The fundamental frequency is assumed by Dunkerley's equation to be substantially lower than the harmonics. The estimated fundamental frequency is always less than the real value since the harmonics are ignored. The natural frequency of a structure's transverse vibrations can be determined using this technique. The system's load is dispersed equally.
- The vibration at the natural frequency is assumed to have a mode shape by the Rayleigh method. The technique is typically employed to determine the fundamental frequency since estimating the harmonics' mode shapes is more challenging. It is the same as adding more constraints to the system if the assumed mode shape is not exact. As a result, the estimated frequency frequently exceeds the actual value. This technique is used to determine the system's natural frequency when transverse point loads are applied to the shaft beam. It is predicated on the idea of energy conservation. To find the natural frequency, the maximum kinetic energy and maximum potential energy of the system are equalized.
- Holzers Method: In the Holzer method, if the boundary conditions for a multirotor system are met, the solution is obtained assuming a trial frequency.

III. DISCRPTION OF HAMMER MILL'S BLADE

Hammer Mill: The hammer mill depicted in Figure 1 is used to break down materials that are fed in at a 100 mm screen passing and are then crushed dry or wet. Between swinging hammers, a half-serrated liner, and a bottom screen with a diameter ranging from 3 mm to 0.5 mm, pulverization occurs. The swinging hammers reduce the size between the screen and liner. Material can be fed from the side of the center and discharged from the bottom. Application: For wet dry fibrous and non-fibrous capacities ranging from 10 kg/hr to 10 tons/hr, coarse and medium fine grinding is needed. There are several sizes available with 3 HP to 75 HP.

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Fig:1 Rotor assembly of Hammer Mill

1. Design data for the hammer mill: The following data were obtained by disassembling the machine for all the rotating elements mounted on shaft and coupled to the motor.

Machine name - Hammer Mill

Shaft & Rotor material - SS 304

Bearing house - Masta SNA 520

Bearing - 1220 (self-aligning ball bearing)

N1=1475 rpm, N2=2360 rpm, D1=406.4mm, D2=254mm.

Working speed = 2800 rpm

Motor = 75 HP (55950 w)

$$P = \frac{(2 * \pi * N * T)}{60} W$$

$$T = 226.60 \text{ N.m}$$

$$M = 82.78 \text{ kg}$$

$$\rho = 7.85 \text{ gm/cm}^3$$

• Dimensions Shaft

Length = 1486 mm

Diameter = Varying (85, 90, 100, 110)

No of step of the shaft = 5

Calculating mass of the shaft

$$V_1 = \pi^2 h = 1332.83 \text{ cm}^3$$

$$m_1 = \rho * V = 7.85 * 1332.83$$

$$= 10462 \text{ gm} = 10.462 \text{ kg}$$

Similarly the mass of other step sections of the shaft

$$m_2 = 13.71 \text{ kg}, m_3 = 4.846 \text{ kg}, m_4 = 38.51 \text{ kg}, m_5 = 2.68 \text{ kg}, m_6 = 12.57 \text{ kg}$$

Total mass of the shaft

$$m = m_1 + m_2 + m_3 + m_4 + m_5 + m_6 = 82.78 \text{ kg}$$

• Rotor

Diameter = 450 mm

Inner diameter = 100 mm

Thickness

16 mm (6 no.): Thicker disk

$$V_1 = \pi \frac{d_1^2 - d_2^2}{4} = \pi * (450^2 - 100^2) * 16 = 2417.8 \text{ cm}^3$$

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$$m1 = \rho * V = 7.85 * 2417.8$$

$$= 18979.73gm = 18.97kg$$

9 mm (5 no.): Thinner disk

$$V_1 = \pi \frac{d_1^2 - d_2^2}{4} = \pi * (450^2 - 100^2) * 9 = 1360.01cm^3$$

$$m1 = \rho * V = 7.85 * 1360.01$$

$$= 10676.0gm = 10.676kg$$

- Hammer

Total no. = 90

Length = 195 mm

Diameter = 6.5 mm

$$V_1 = bh - 2\pi r^2 h = 1.04 * 10^{-4} - (2 * \pi * 0.0125^2 * 0.008)$$

$$= 96.67cm^3$$

$$m = \rho * V = 7.85 * 96.67$$

$$= 6836.4gm = 6.836Kg$$

- Bearings

Inner diameter = 100 mm

Outer diameter = 140 mm

Thickness = 38 mm

Weight = 17.5 kg

Locking plate

Inner diameter = 140 mm

Outer diameter = 450 mm

Thickness = 14

$$V1 = \pi \frac{d_1^2 - d_2^2}{4} = \pi * (450^2 - 140^2) * 14 = 2010.071cm^3$$

$$m = \rho * V = 7.85 * 2010.071$$

$$= 157790.0gm = 15.779Kg$$

- Bush

Total no = 10

Inner diameter = 100 mm

Outer diameter = 140 mm

Thickness = 48 mm

IV. ANSYS MODAL ANALYSIS

A. Modal Analysis: The dynamics behavior of rotating structures (shafts, bearings, and other rotating elements) under dynamic excitation is the subject of modal analysis. The first critical speed, or fundamental resonance frequency, can be computed with the aid of the modal analysis. Identifying the causes of vibrations that compromise the integrity of system components is helpful. In certain operating conditions, this can enhance the system's overall performance and prevent the design stage from reaching the critical speed. The numerical modal analysis and the experimental modal analysis are the two fundamental modal analysis techniques.

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Modal analysis is the technique used to determine a structure's vibration characteristics; Natural frequencies, Mode shapes. Modal analysis in the ANSYS family of products is a linear analysis. Any nonlinearity, such as plasticity and contact (gap) elements are ignored even if they are defined. Mode participation factors (how much a given mode participates in a given direction)

Benefits of modal analysis [3]:

Allows the design to avoid resonant vibrations or to vibrate at a specified frequency (speakers, for example).

Gives engineers an idea of how the design will respond to different types of dynamic loads.

Helps in calculating solution controls (time steps, etc.) for other dynamic analyses.

General equation of motion:

$$m(\ddot{y}) + c(\dot{y}) + k(y) = F(t)$$

Assume free vibrations and ignore damping

$$m(\ddot{y}) + k(y) = 0$$

Assume harmonic motion

$$K - \omega^2 m = 0$$

The roots of this equation are ω_i^2 , the eigenvalues, where i ranges from 1 to number of DOF. Corresponding vectors are $\{y\}_i$, the eigenvectors. The square roots of the eigenvalues are ω_i , the structure's natural circular frequencies (radians/sec). Natural frequencies f_i are then calculated as $f_i = \omega_i / 2\pi$. It is the natural frequencies f_i that are input by the user and output by ANSYS[6]. The eigenvectors $\{y\}_i$ represent the mode shapes - the shape assumed by the structure when vibrating at frequency f_i .

B. Modal Analysis in Ansys work bench: ANSYS is universal software, which is used on simulation of the interactions in physics structures, vibration, fluid dynamics, thermal transfer and electro mechanics for engineers [8].

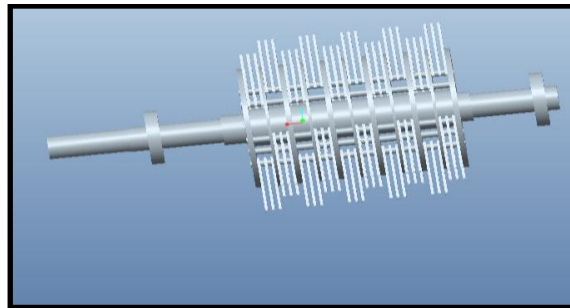


Fig. 2. CAD Model of full assembly of Hammer Mill

- The geometry is imported to Ansys, which is generally a STEP file (.stp) that can be created in Pro-Engineer. Fig. 2 is the complete assembled model of the hammer mill shaft, bearings, rotors and the rotor pins carrying the hammers.
- Allocating Material Properties of Structural Steel.

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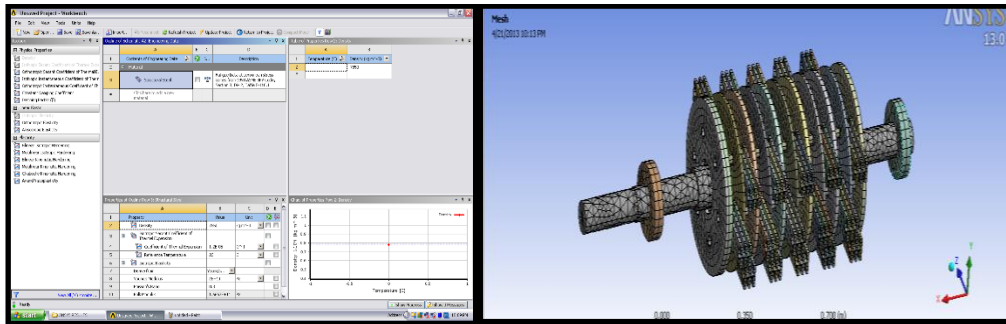
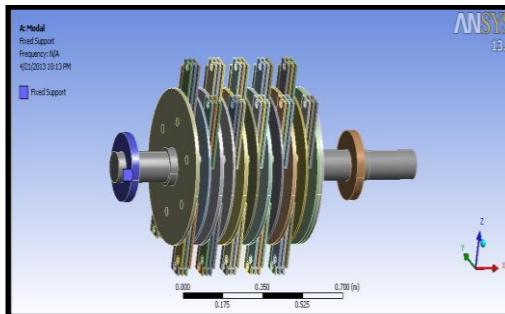


Fig. 3. Meshed Model of the hammer mill

- Meshing the geometry into small elements
- Giving constraint by taking both bearings as a fixed support
- Finally, solving the model to obtain the results



**Fig. 4 Constrained Model
for the fundamental frequency**

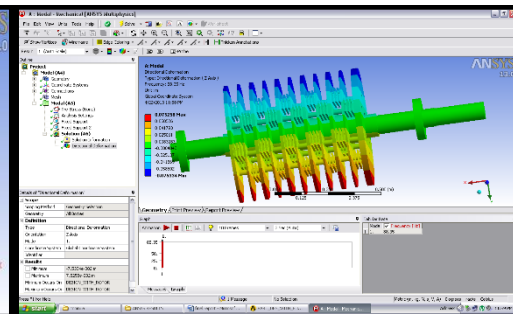


Fig. 5. Modal analysis

V. RESULTS AND DISCUSSION

The manual calculation of the shaft and rotors by Dunkerly and Rayleigh methods gives the fundamental resonance frequency and then with the maximum deflection of the shaft helps to find the critical speed of this multirotor shaft. When further analyzed in ANSYS for the same speed the frequency of resonance was almost matching with the hand calculation. Tables-1 to 4 compares the three methods for the given multi degree system. Table 1&2 compares the result of ANSYS, Dunkerly and Rayleigh. Table 3&4 gives the critical speed comparison.

(1) For FREQUENCY

Table .1 Frequency Calculations and Percentage Variations in Frequency

Node	ANSYS	DUNKERLY	RAYLEIGH	% Variation for the frequency	
1	88.35 Hz	89.42 Hz	92.33 Hz	ANSYS- DUNKERLY=1.19%	ANSYS- RAYLEIGH=4.31%

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(2) For CRITICAL SPEED

Table .2.Critical speed calculation and Percentage variation in critical speed

ANSYS	DUNKERLY	RAYLEIGH	% Variation for the rotating speed	
5301rpm	5365.49 rpm	5540.22 rpm	ANSYS-DUNKERLY=1.19%	ANSYS-RAYLEIGH=4.31%

The desired speed is 2800 rpm and the critical speed is greater than 2800 rpm, approximately 5300 rpm. Therefore critical speed will not be attained and machine will work without attaining critical speed. No vibration and no unbalance forces will be induced. The machine will work satisfactorily. If the equipment is required to run at a speed above 5300 rpm, then certain preventive measures can be taken. For critical speed attainment and go beyond that without damaging the machine. The designer can use various other methods like TMM (transfer matrix method, Stodola and Holzers) for the perfect fundamental frequency.

OBSERVATIONS OF THE ANALYSIS

- Weaker areas

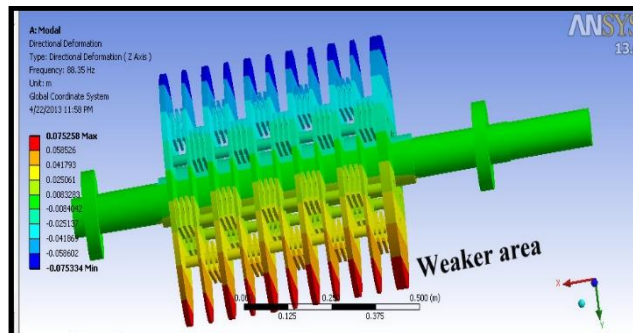


Fig .6. Weaker area.

To make weaker areas stronger, the diameter of the shaft can be increased. But the diameter will increase then weight of the shaft also increases. The weight reduction can be achieved by making the shaft hollow with high stiffness.

- Bearing suggestion

As per the machine's condition, two types of bearing can be suggested to modify the stiffness properties of the system. As the simple roller bearing increases the stiffness in the motion equation leads to the higher amplitudes of vibration. Change in bearing properties will lead to better results. So bearing can be changed to

- Needle type roller bearing which have extremely high radial load carrying capacity and are suitable for particularly compact designs.
- Tapered roller bearing has cone shaped design because of which each roller in the bearing can align itself perfectly between the tapered faces of the cup and cone, without guidance by the cage. This bearing has the ability to support radial and thrust loads simultaneously which can support such heavy rotors and also the thrust while the equipment is loaded. It can also take thrust load in both the directions.
- Stiffness: The stiffness of the shaft can be modified. Because increase in stiffness will increase critical speed
- Use of composite material

Dynamics of rotating composite material shaft have been an important issue in the field of rotor dynamics. The composite shaft with n number of rigid disks and supported in linear elastic bearings can give better

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fundamental frequency than normal shaft. Composite material like carbon epoxy or epoxy resin to increase the limit of critical speed. Composites have historically been used for their outstanding mechanical properties and low densities. They have low coefficients of thermal expansion (CTE). High damping capacity helps to minimize the lateral vibrations.

VI. CONCLUSION

- An attempt is made to compare the analytical equation of multi-degree freedom system of Dunkerly and Rayleigh method with the ANSYS software.
- It is observed that the percentage variation in the fundamental frequency and critical speed of the shaft rotor system using Dunkerly and Rayleigh methods with ANSYS Modal analysis is below 10%. This suggests that the ANSYS modal analysis of any complex multi rotor shaft can be carried at the design stage by the designer to avoid critical speed attainment at an early stage of working.
- If this hammer mill is required to run at a speed above 5300 rpm, then certain preventive measures can be taken for critical speed attainment and work beyond that without damaging the machine.

ACKNOWLEDGMENT

This study was possible with the co-operation of many supporting persons and well-wishers. I am grateful to my supervisor Dr.V.D.Dhiman for invaluable guidance and suggestions during the entire study. I would also like to thank Mr. Mikin Shah, GM of FINEX SIEVES PVT LTD., Vadodara, Gujarat, for the set of design data for the given food processor (Hammer Mill).

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