# Design and Application of a Vibration Exciter System
# Bir Titreşim Uyarıcı Sisteminin Tasarımı ve Uygulamalası

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***Abstract*—In this work, beside estimation of mechanical and electrical design parameters the dynamic analysis of a vibration exciter system using computer calculations is investigated. Depending on basic mechanical parameters such as shaking force and power, strength of the motor shaft according to torsion and bending has been made. With determined electrical parameters, the characteristic curve of the AC motor is drawn. Finally, the frequency response of the vibration system depending on various shaking masses is described by test curves. Damping and stiffness coefficients of the exciter system are then found from measured data of test curves. Results are rebuilt with relevant computer simulations and graphics. The system is designed for determining the elasticity and damping properties of the vibration elements.**

***Keywords*—Vibration; Motor; Bending; Torsion; Response***

***Özetçe*—Bu çalışmada, mekanik ve elektriksel tasarım parametrelerinin tahmin edilmesinin yanı sıra, bilgisayar hesaplamaları kullanılarak bir titreşim uyarıcı sisteminin dinamik analizi incelenmiştir. Titreşim kuvveti ve gücü gibi temel mekanik parametrelere bağlı olarak motor milinin burulma ve eğilmeye göre dayanımı yapılmıştır. Belirlenen elektriksel parametreler ile AC motorun karakteristik eğrisi çizilmiştir. Son olarak, çeşitli sarsma kütlelerine bağlı olarak titreşim sisteminin frekans tepkisi test eğrileri ile tanımlanmıştır. Daha sonra test eğrilerinin ölçülen verilerinden uyarıcı sistemin sönümleme ve sertlik katsayıları bulunur. Sonuçlar, ilgili bilgisayar simülasyonları ve grafiklerle yeniden oluşturulur. Sistem, titreşim elemanlarının esneklik ve sönümleme özelliklerini belirlemek için tasarlanmıştır.**

***Anahtar Kelimeler*—Titreşim; Motor; Eğilme; Burulma; Tepki***

## I. Introduction

Vibration motors as reliable and powerful vibration exciters are used for all applications of active vibration technologies as sieving, conveying, shaking, compressing, and testing. The design considers the requirements of practice. A mechanical exciter is a mechanical device to generate vibrations. The vibration is often generated by an electric motor with an unbalancing mass on its driveshaft [1].

There are many different types of vibration exciters. Typically, they are components of larger systems. The unbalancing weights attached to the shaft ends on one or both sides generate a circular vibration when rotating, which imparts a vibration movement to the components coupled via the motor base such as conveyor troughs or screening machines. Directional vibrations are generated when two vibration motors are operated in opposite directions. The selection of the vibration motors depends on the required centrifugal force and the desired speed (Fig. 1).

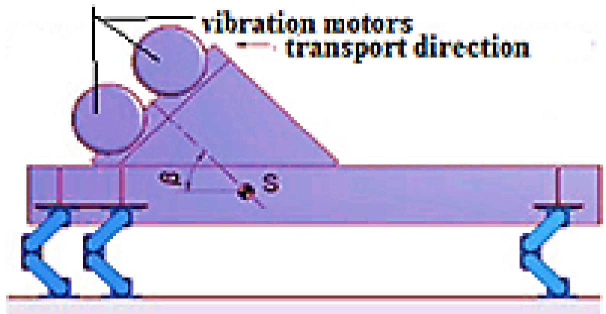



Figure 1: Circular vibration exciter

## II. Industrial Vibrators

Vibrators are used in a wide range of industrial applications, both as stand-alone equipment and as part of larger systems [2]–[4].

In the food, pharmaceutical and chemical industries, vibratory bowl feeders, vibratory conveyors and vibratory hoppers are often used to move and position bulk materials or small components. Using vibration in conjunction with gravity to move materials through a process is often more efficient than using other techniques. Small components are often moved by vibration so that they can be mechanically picked up by automated machinery when required for assembly, etc (Fig. 2).

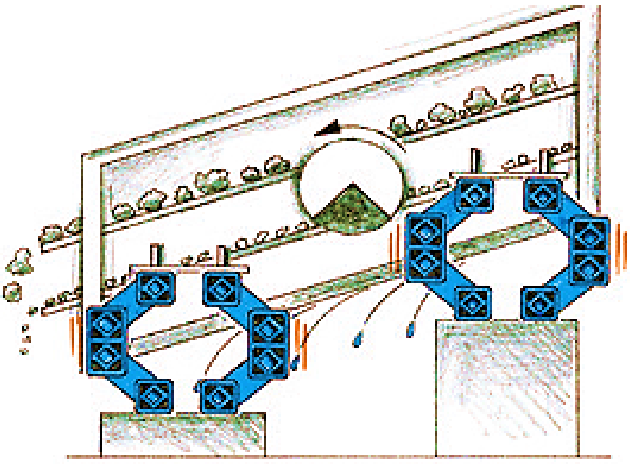


Figure 2: Material transport via vibration motor

Bulk materials in a mixture of particles of varied sizes are separated using vibrating screens. For example, vibrating screens are often used to separate materials such as sand, gravel, river rock, crushed rock, and other aggregates according to their size [5].

Vibrating compactors find application in soil compaction, particularly for foundations in roads, railways, and buildings.

Concrete vibrators serve the purpose of consolidating freshly poured concrete, ensuring the release of trapped air and excess water, and achieving a secure settling of the concrete within the formwork. Improper consolidation of concrete can lead to product defects, compromise its strength, and result in surface blemishes such as bug holes and honeycombing [6].

Vibrating tables or shake tables are occasionally employed to assess products, assessing their ability to withstand vibrations. Such testing is commonly conducted in the automotive, aerospace, and defense industries.

### A. *Free-Swinging Systems with Unbalancing Exciter*

Cantilever chairs are made using unbalancing exciters, vibration motors or unbalancing shafts vibration. The vibration amplitude, the vibration shape and the vibration direction of the device are determined by their dimensioning or arrangement. The excitation force, the angle of attack, the exciter, the inclination of the table, the payload, and the position of the center of gravity determine the resulting oscillation range of the device. By changing parameters, the amplitude can be determined and thus the conveying speed of the machine can be optimized.

The spring bearings support the desired oscillating movement of the screening machine. Because of their form and function, they help to achieve a linear conveying movement without undesirable lateral displacement. These ideal spring bearings harmoniously support the running of the vibration device. Due to their high devotion, circular vibratory cantilever systems with unbalance excitation technology offer an exceptionally low natural frequency, which guarantees good detuning of the excitation frequency and a prominent level of insulation from the machine bed. When starting and stopping or passing through the spring resonance frequency built up by the bearings, the large residual force reaches its peak effectively and quickly [7], [8].

### B. *Mechanical Calculation of Motor Shaft with Computer for a Freely Lying Beam with External Individual Load*

A mechanical exciter is a mechanical device to generate vibrations. The vibration is often generated by an electric motor with an unbalancing mass on its driveshaft. The unbalancing weights attached to the shaft-end generate a circular vibration when rotating, Motor shaft can be modelled as freely lying beam with external individual load as shown in Fig. 3.

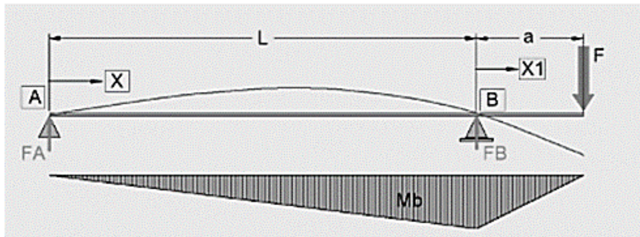



Figure 3: Freely lying beam with external individual load

Using given design data, the relevant values can be calculated by a specially written program as follows. After determining and calculating the maximum bending moment, the necessary shaft diameter can be calculated as follows.

$$d_a \geq \sqrt[3]{\frac{32Mb}{\pi\alpha_{b,zul}}} \tag{1}$$

$$w = \sqrt{\frac{c}{m}} \tag{2}$$

$$n_k = f_k 60$$

$$f_k = \frac{w}{2\pi} = \frac{1}{2\pi}\sqrt{\frac{F_G}{f_G m}}$$

$$= \frac{1}{2\pi}\sqrt{\frac{g}{f_G}} \tag{3}$$

$$C = \frac{3EI1000}{a^2(a+L)} \tag{4}$$

$$M = \frac{P}{w} = \frac{Pn}{30\pi} \tag{5}$$

$$\tau_t = \frac{M_t}{W_t} \tag{6}$$

$$d_a \geq \sqrt[3]{\frac{16M_t}{\pi\tau t, zul}} \tag{7}$$

Input values: imbalance load (F) is 290 N, shaft length (L) is 450 mm, force path (a) is 350 mm, modulus of elasticity (E) is 210000 N/mm2 and maximum bending moment at x = 450 mm is 1015000 N.mm. The calculation result of the shaft with (1), calculation of the critical frequency and speed with (2) and the natural frequency with (3), calculation of the bending constant C with (4) for Fig. 4.

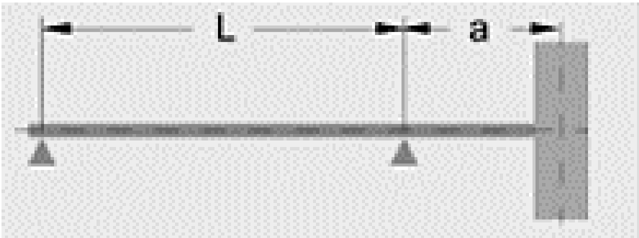



Figure 4: Unbalanced shaft calculation diagram

Shaft diameter against torsion (6) is determined from the mechanical power (5) of motor via (7).

## III. Basic Parameter Calculation of AC Motor

A complete drive consists of the motor and the working machine. The electric motor that generates a torque that changes with speed is described with the speed-torque characteristic curve. The driven machine contrasts this motor torque M with a load torque at the same speed. Fig. 5 shows the different speed-torque characteristics of drive machines. These properties influence the selection of a drive. Torsion load related to the bending load is small; therefore, the calculated shaft diameter against bending is decisive.

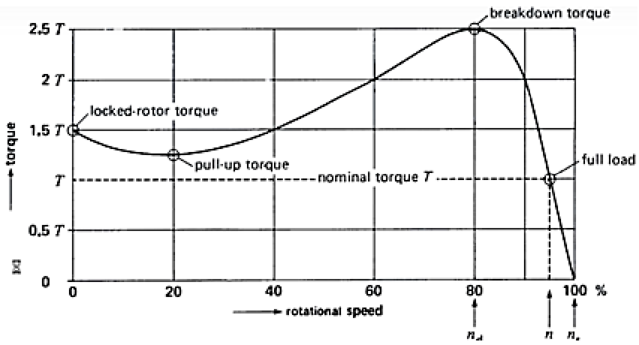



Figure 5: Torque-Speed curve of asynchronous motor [8]

Three-phase squirrel cage AC motors can be applied to all kinds of machines, such as machine tools, air compressor, pump, ventilation, reduction gear transmission belt, mining equipment, metallic equipment. As synchronous motor can be compared structurally to transformers. Rotor as transformer secondary windings and stator corresponding to the primary windings of the transformer.

The effects of the idle working current (excitation circuit) are presented with parallel connected $X_0$ and $R_0$ equivalent.

### A. Motor Current Calculation

Once the equivalent circuit parameters are known, it is easy to calculate the motor current, by reducing the circuit to an equivalent impedance $Z_{eq}$, giving (8).

$$I_s = \frac{V_1}{Z_{eq}} \tag{8}$$

By inspecting the equivalent circuit, we can see that $Z_{eq}$ is of the form (9)

$$Z_{eq} = R_{eq} + \frac{R_2'}{s} + jX_{eq} \tag{9}$$

From (9), as the rotor speeds up (slip reduces), the circuit impedance increases and stator current decreases [8].

### B. Motor Power Calculation

As a simplification, if we neglect the core losses (RC and giving $I_S = I_2'$) the power ($P_{in}$) delivered to the motor per phase is given by (10).

$$P_{in} = I_s^2 + (R_1 + \frac{R_2'}{s}) \tag{10}$$

The power loss dissipated by the windings is given by

$$P_w = I_s^2(R_1 + R_2') \tag{11}$$

The difference between the power supplied to the motor and losses in the windings is the power (Pm) delivered to the connected load. This (per phase) is given by (12) and for all three-phases by (13) [8]

$$P_m = P_{in} - P_w = I_s^2(\frac{1-s}{s})R_2' \tag{12}$$

$$P_{m3\phi} = 3I_s^2(\frac{1-s}{s})R_2' \tag{13}$$

### C. Motor Torque Calculation

Power can be calculated with the multiplication of motor torque (T) and angular velocity (w), this can be rearranged to derive a formula for torque as in

$$\begin{aligned} T &= \frac{P_{m3\phi}}{w} \\ w &= \frac{2\pi}{60}(1-s)n_s \\ T &= I_s^2\frac{90}{\pi n_s}\frac{R_2'}{s} \end{aligned} \tag{14}$$

## IV. Mechanical Implementation of Vibration System

A vibration system is designed to produce a given range of harmonic or time dependent excitation force and displacement through a given range of frequencies. The system is used to determine the elasticity and damping properties of the support elements. Vibrations are generated by the mechanical vibration exciter through the centrifugal force of a rotating eccentric mass. The vibrations produced lie in the low frequency range. The system is considered as a machine with rotating unbalance supported on a spring and a damper (i.e., mounted on an elastic support) (Fig. 6).

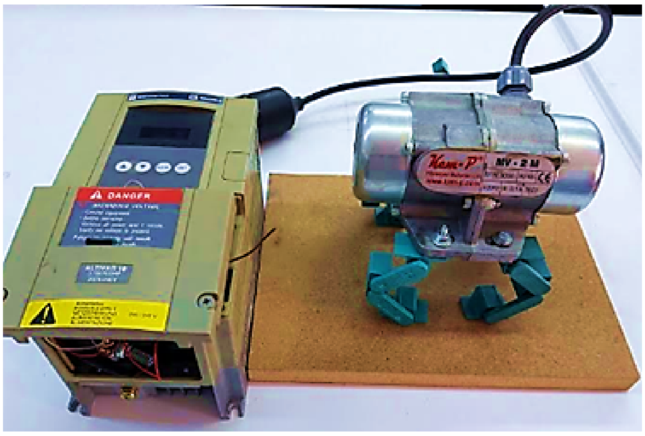


Figure 6: Implementation of the test system

The exciter consists of five main parts: Base frame, Drive system (motor with, eccentric mass), Speed Control Unit, Support (Spring-Damper) Elements and Top plate with quidding rolls [9]. The most important part of exciter is rotating unbalanced mass attached at one end of motor shaft. Motor speed control unit drives the system with different frequencies and provide the investigation of frequency response.

### A. Dynamic Modelling of Vibration System

The system can be modelled and considered as a machine with rotating unbalance supported on a spring and a damper (i.e., mounted on an elastic support) (Fig. 7). The concept of unbalanced mass excitation is used as source of forced vibrations. The unbalance is measured in terms of mass (m) rotating with its center of gravity at a distance (e) from the axis of rotation. The centrifugal force due to unbalance mass acts as harmonic excitation force [10]–[13].

Assuming that the system is constrained to move vertically having single degree of freedom. The differential equation of motion can be written as

$$M\ddot{x} + c\dot{x} + Kx = mw^2 esin(wt) \tag{15}$$

K is spring stiffness of the support (N/m), C is damping coefficient (Ns/m). The equation (15) is linear second order differential equation of motion for a forced damped vibrations

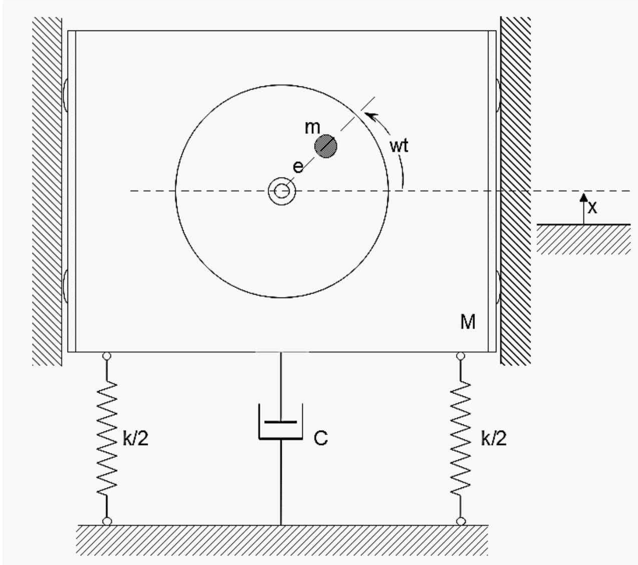



Figure 7: Schematic diagram of a mechanical vibration exciter

due to rotating unbalance. By using the following transfer function, amplitude of steady state vibrations, resonance frequency and resonance amplitude can be derived [14]–[16].

$$F(jw) = \frac{w_n^2 K}{s^2 + 2\xi w_n s + w_n^2} \tag{16}$$

The amplitude of steady state vibrations for $r = w/w_n$ is given as

$$X = \frac{r^2}{\sqrt{(1 - r^2 + (2\xi r)^2}} \tag{17}$$

and the transfer function for the system as

$$F(w) = \frac{X(w)}{mw^2 e} \tag{18}$$

To obtain resonance frequency; local maxima is found via derivation of corresponding variable. For this purpose, in the following derivation is conducted, and the resonance frequency is found.

$$\frac{dF(w)}{dw} = 0 \rightarrow w_r = w_n\sqrt{1 - 2\xi^2} \tag{19}$$

Thus, the resonance amplitude can be written as

$$F_r = X_r = \frac{1}{2\xi\sqrt{1 - \xi^2}} \tag{20}$$

From (17) for $w = w_r$ (that is shown as $F_r$) can be written as

$$F_r = \frac{MX_r}{mw_r^2 e} = \frac{MX_r}{me}\frac{1}{1 - \xi^2} \tag{21}$$

and, finally,

$$\frac{MX_r}{me}\frac{1}{1-\xi^2}=\frac{1}{2\xi\sqrt{1-\xi^2}} \tag{22}$$

The plot of dimensionless amplitude vs frequency ratio is given in Fig. 8 which gives the information about when the speed is zero and the dimensionless amplitude is also zero. Consequently, every curve originates from the origin. At resonance, the dimensionless amplitude is solely restricted by the damping of the system. When frequency ratio is exceptionally large, the dimensionless amplitude tends to unity. By solving (22) with following data from amplitude-frequency graphics (Fig. 8), $\xi$ can be calculated [14].

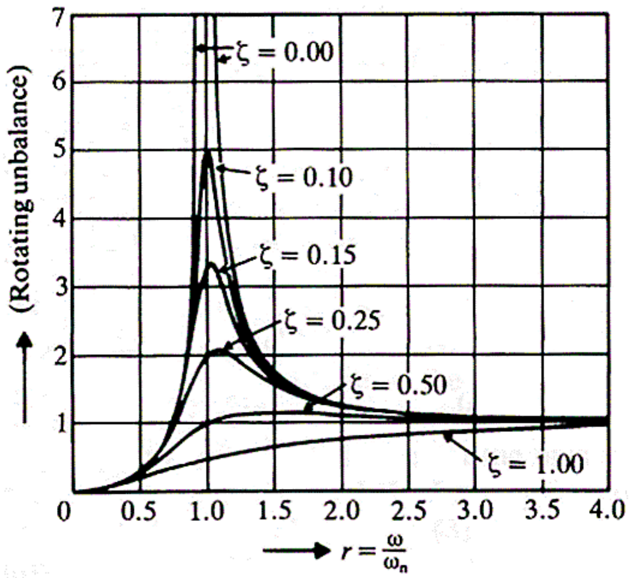



Figure 8: Dimensionless amplitude-frequency plot [17]

### B. Test Results and Theoretical Calculations of System Parameters

The exciter system is designed for experimentation purposes and testing of support elements at different frequencies. To determine the performance of the proposed system, the following tests with design data are performed and discussed. The actual measurement of amplitude is done by varying the speed of motor. To calculate natural frequency and damping co-efficient for rotating unbalanced mass, the taken design data are: Unbalanced masses (m) are 0.25 kg, 0.5kg, 0.75 kg. Eccentric radius (e) = 0.06 m. Total mass (M) including unbalanced mass is 30 kg. Fig. 9 indicates red dots for measured values and the solid line for calculated values.

Similarly, amplitude versus speed plots were achieved for three distinct eccentricities of 20 mm, 40 mm, and 60 mm with the unbalanced mass of 0.50 kg (Fig. 10a) and for three masses of 20 mm, 40 mm, and 60 mm with the eccentricity of 60 mm (Fig. 10b) .

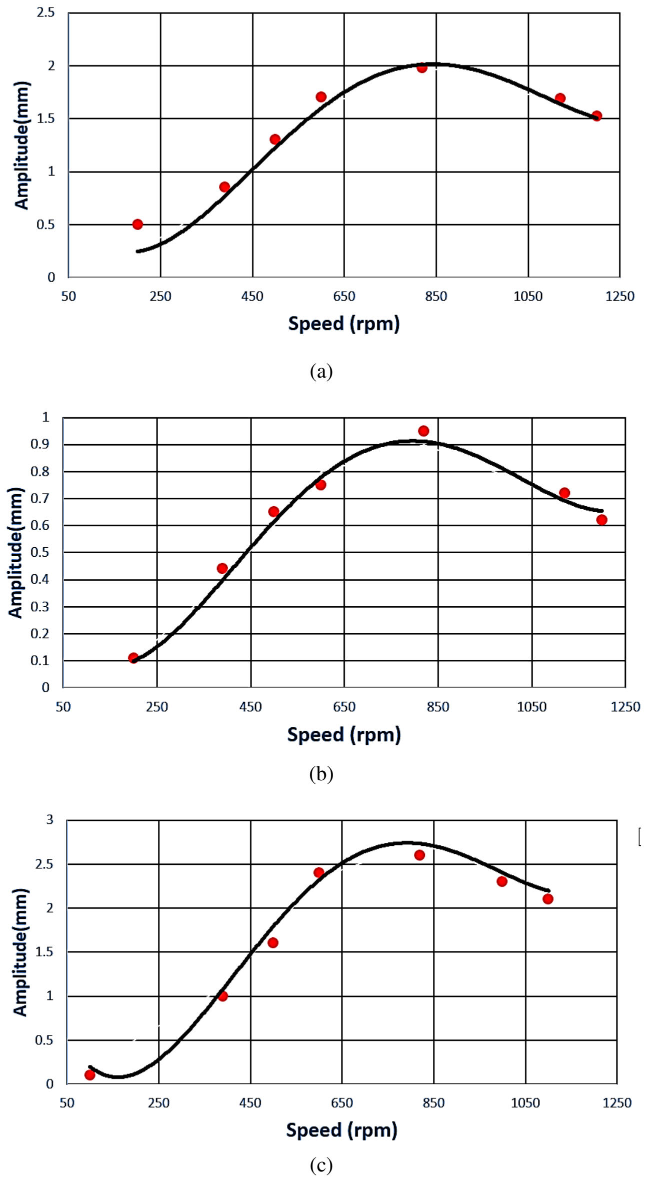
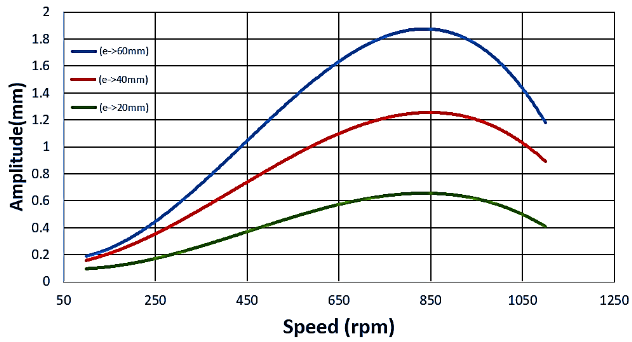


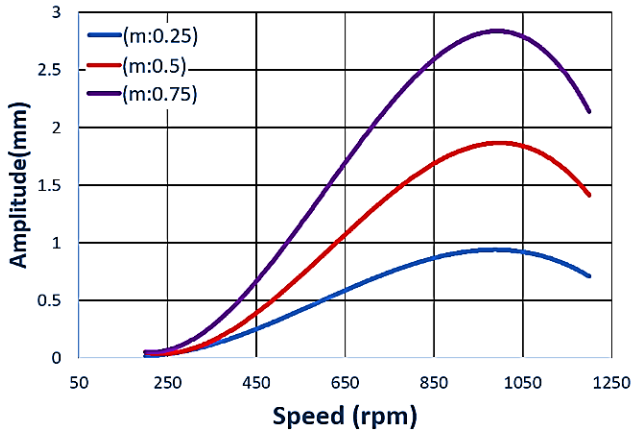

Figure 9: Amplitude versus Speed plot for unbalanced masses of (a) m=0.25 kg, (b) m=0.50 kg, and (c) m=0.75 kg

By using the equation (13) and Figures 9 and 10, $\xi$ and $w_n$ can be calculated. From these figures as speed increases, amplitude of vibration increases up to resonance speed and then decreases. The frequency of maximum amplitude can be calculated from $w=\pi n/30$. From the figure maximum speed can be read as about n=880 rpm and the amplitude are X=2 mm; so $w_r$ can be calculated as 83.7 rad/s.

By using the measured data, e=60 mm, m=0.5 kg, M=30 kg and amplitude value of 1.8 mm (Fig. 10), damping factor $\xi=0.25$ can be found via (22). Also, by using (19) with $w_r=83.7$ (rad/s) and $\xi=0.25$, natural frequency is calculated as $w_n=89.5$ (rad/s). After calculation of $w_n$, spring constant




(a)




(b)

Figure 10: Amplitude versus Speed plot for (a) eccentricities at 20, 40, and 60 mm with the unbalanced mass of 0.5 kg, and (b) unbalanced masses of 0.25, 0.50, and 0.75 kg with the eccentricity of 60 mm

$K$ can be calculated by using $w_n = (K/M)0.5 = 24,037$ kN/m.

Damping coefficient C can also be found from calculated parameters. The amplitude of natural frequency can be estimated via

$$F_r(w) = \frac{MX_r}{me} = \frac{1}{2\xi} \tag{23}$$

With given values M=30 kg, m=0.50 kg, e=60 mm and $\xi$=0.25. Amplitude $X_n = me/(2M\xi) = 2$ which also be confirmed from measured curves in Fig. 8. Finally, the transfer function of system in all can be written as

$$G(jw) = \frac{8100}{s^2 + 45s + 8100} \tag{24}$$

## V. Simulations & Conclusions

Using the given data and calculation basics above, system behavior is proved with MATLAB simulation graphics, which are a confirmation and representation of the measured test curves with various unbalancing masses and eccentricities. From these figures as speed increase, amplitude of vibration increases up to resonance speed and then decreases. It can clearly be seen that the experimental data matches with theoretical results. The plot shows that as unbalanced mass increases, amplitude of vibration increases, which is true as compared with theory given (Fig. 9 and 10).

The eccentricity of the unbalanced mass can be altered by incorporating holes at various radii on the disc. Test results using three different eccentricities (20mm, 40mm, and 60mm) are presented. The plot shows that as the eccentricity increases, the vibration amplitude also increases until reaching the resonance speed, after which it starts to decrease.

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