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Development of a Multi Degree-of-Freedom Vibration Exciter for Laboratory Applications

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Abstract: Introduction of vibration to manufacturing operations such as casting and welding has proved to improve the physical and mechanical properties of manufactured parts. A vibration exciter is developed for the purpose of generating and inducing vibration, along different degrees of freedom, on objects placed on its surface. The equipment applies an eccentric mass drive system which gives the equipment an overall advantage in varying the vibration parameters. The acceleration of the vibratory motion of the equipment was measured using an accelerometer and oscilloscope set-up. The natural frequencies of the different vibration modes are also obtained from a developed mathematical model executed using the MATLAB Simulink software. The developed equipment successfully generated random sinusoidal vibrations of accelerations ranging from −5 $\boldsymbol{m/s^2}$ to 8 $\boldsymbol{m/s^2}$ along the principal axes and angular accelerations ranging from −40 $\boldsymbol{rad/s}$ to 40 $\boldsymbol{rad/s}$ about the pitch and roll axes. Natural frequencies of $\boldsymbol{f_x}$ = 3.78 *Hz*, $\boldsymbol{f_\theta}$ = 7.94 *Hz* and $\boldsymbol{f_\varphi}$ = 9.89 *Hz* are obtained along the vertical, pitch and roll directions respectively. The presented results indicate that the developed machine successfully satisfied the proposed hypothesis of being able to measure vibrational characteristics along different degrees of freedom.

Index Terms: Vibration Exciter, Eccentric Mass, Mechanical Vibration, Design, Fabrication.

1. Introduction

Microstructural grain refinement has been reported as one of the most important factors of improving the quality of castings [Tamura *et al.*, 2011, Dahunsi and Audu, 2006, Eshan et al, 2009]. Thus various methods have been proposed to refine and homogenize castings microstructures. Studies [Audu and Dahunsi, 2003, Omura *et al.*, 2006, Zhao *et al.*, 2010] have shown that the imposition of vibration is a proven approach among these methods. However, it necessary to determine the vibration characteristics responsible for the structural grain refinement. A vibration exciter, with the capacity of imposing vibration on an object placed on it, measuring and varying the vibration characteristics, is required for the task.

Application of vibration to casting manufacturing dated back to 1868 when the solidification of a melt was induced by vibration [Thoguluva *et al.*, 2012]. Since then, developments have been made to establish the optimal vibration parameters for improving casting operations. According to [Abugh and Kuncy, 2013], introduction of high intensity ultrasonic vibration into melt controls the undesirable columnar structure and refines equiaxed grains. Microstructural uniformity of cast structure was achieved in [Zuo *et al.*, 2016] by imposing electromagnetic vibration on Al-Zn-Mg-Cu alloy. The strength of aluminum A356 alloy was enhanced using ultrasonic vibration during the production of its castings [Jianbo *et al.*, 2009]. Vibration has also been applied to other manufacturing processes such as burnishing of aluminum [Travieso *et al.*, 2015], welding of stainless steel [Sakthivel and Sivakumar, 2014], laser powder deposition on alloy steel [Eshan *et al*, 2009] characterization of composite structure [Apalowo and Chronopoulos, 2018] and sieving of agricultural grains.

Several vibration exciters have been developed to generate and induce vibration for industrial and laboratory applications. They can be classified based on the type of mechanism used to produce the vibration excitation. Cam and follower mechanism [Pawar *et al.*, 2016] produces a constant amplitude regardless of the applied load. However, the mechanism is comparatively difficult to set up. Meshing gear mechanism [Wowk, 1991] allows for an easy variation of shaft speed but it generates noise. Scotch yoke mechanism [Anekar *et al.*, 2014] is easy to apply but only applicable for low speed device and wears easily. The rotating eccentric mass mechanism [Anekar *et al.*, 2014, Gregory, 2011] is easy to fabricate, generates minimal wear and relatively easy to adjust. Though it does not allow for shaft speed variation, but the speed can be varied using DC motor with variable speed [Anekar *et al.*, 2014]. Based on these comparative advantages, the eccentric mass mechanism is adopted in this work.

As reported in the literatures, focus has been placed on varying the eccentric force of an eccentric mass mechanism by changing the speed of the electric motor as it has the greatest influence on it [Sayuti *et al.*, 2012]. The main contribution of this work therefore is to develop a vibration exciter which uses the rotating eccentric mass mechanism in which the eccentric force can be varied by varying the unbalanced mass and its eccentric distance. This is easier and cheaper to incorporate compared to using a variable speed DC motor. Also, developing an exciter which can generate vibration characteristics in different directions would be a great contribution to the literature. Therefore, the objective of this work is to propose a new varying-speed vibration exciter which can measure vibration characteristics along different degrees of freedom.

The remainder of this article is organized as follows: Section 2 presents the design development which involves the design calculations, drawing and description. Section 3 presents the fabrication of the designed device and its brief of operation. Results and analysis are presented in Section 4. Finally, Section 5 presents the concluding remarks of the work.

2. Design Development

Initially, a number of conceptual designs were proposed for the vibration exciter. Upon preliminary design calculations and analysis, the concepts were narrowed down to a viable solution. The final proposed design is a vibration exciter system which uses eccentric mass to generate vibration, and which has the capacity to attain seven degrees of freedom. The proposed design is presented in Fig. 1.

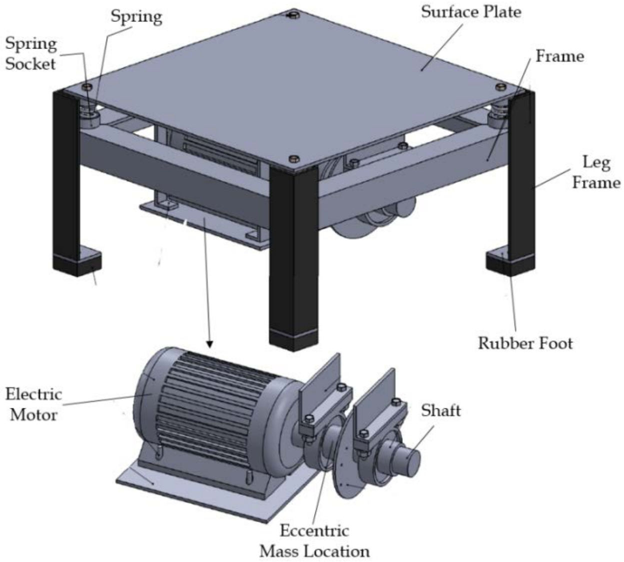


Fig. 1: Vibration Exciter

2.1 Design Description and Calculations for Essential Machine Components

The essential components of the 7-degree of freedom vibration exciter are briefly described in this section. Design calculations for these components are also presented. The calculations are based on the models presented in standard design reference and data texts [Khurmi and Gupta, 2005, Hall *et al.*, 1982, Bhandari, 2014].

2.1.1 Electric Motor

The vibration exciter is to be used for laboratory applications, therefore the electric motor required should be of average size and capacity ratings. Hence, an electric motor of rated power (P) of 750 W and speed (N) of 1400 rpm was selected. This will transmit a torque (T) of 5.1 Nm based on Eq. (1).

$$T = \frac{60 \times P}{2\pi N} \quad (1)$$

2.1.2 Shaft

Power is transmitted to the eccentric mass, which generates vibration on the vibration exciter, through a shaft on which the mass is mounted. The shaft transmits motion and torque from the electric motor to the vibration exciter. Therefore, it is essential to design for the adequate size of the shaft which can withstand the twisting and bending moments it is subjected to. The length of the shaft was chosen as 200 mm to minimize bending moment and deflection. The shaft configuration is presented in Fig. 2, where points A and B represent the positions of the bearings on the shaft while point C is the eccentric mass location.

The dynamic eccentric force on the shaft (F_C) can be expressed as

$$F_C = me\omega^2 \tag{2}$$

where m and e are the mass and radius of the eccentric mass and $\omega = \frac{2\pi N}{60}$ is the angular velocity of the shaft. In this work, the considered mass and radius of the eccentric mass are limited to maximum of 30 g and 60 mm respectively. Hence, the force at point C is computed as F_C = 38.68 N. Through force analysis, the forces at the two bearing locations $F_A = F_B =$ 19.34 N. Through bending analysis, bending moment at the bearing locations are zero. At point C, $M_C = 0.967$ Nm. From Fig. 2, the maximum bending moment is obtained as $M = 0.967$ Nm.

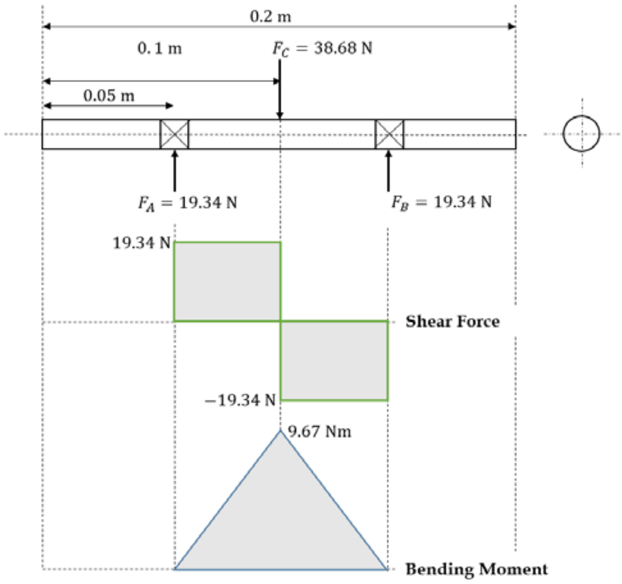


Fig. 2: The Shaft Configuration and its Shear Force and Bending Moment Analyses

The suitable shaft diameter is then determined using the calculated torque and bending moment as

$$d^3 = \frac{16}{\pi\tau}\sqrt{M^2 + T^2} \tag{3}$$

where τ is the permissible shear stress. Considering an adequate factor of safety, τ is taken as $0.4725\ MPa$ for a steel shaft. The shaft diameter is then calculated as $38.3\ mm$. The diameter is taken as $40\ mm$.

2.1.3 Bearing

The radial ball bearing is selected for this device due to its known advantages such as low friction as well as ease of maintenance and replacement. Bearing 6308, of inner diameter of 40 mm, outer diameter of 90 mm and a width of 23 mm, was chosen. The rating life L of the bearing is estimated as

$$L = \left(\frac{C}{W}\right)^3 \times 10^6 \tag{4}$$

where C and W are respectively the basic and the equivalent dynamic load ratings of the bearing.

2.1.4 Spring

Helical spring with squared and ground ends made of carbon steel material was selected for the device for a good stability. As shown in Fig. 1, the springs are introduced between the frames and eccentric assembly of the machine to help isolate the vibratory forces from the machine supports. Mechanical tests were conducted to determine the spring stiffness and weight. The experimental set up for the experiment is as shown in Fig. 3.

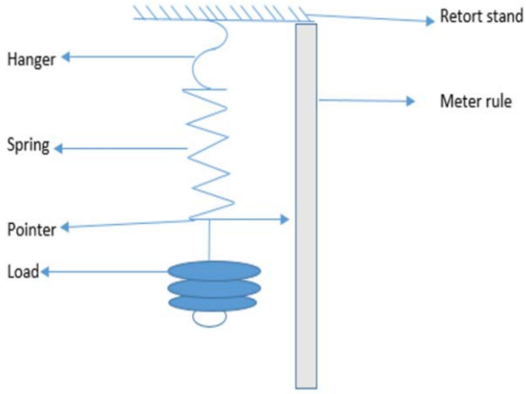


Fig. 3: The Experimental Set-Up for Determining the Stiffness of the spring.

As shown in Fig. 3, a known mass was loaded on the hanger and the new position of the pointer was recorded. This was repeated for a number of masses and a plot of mass against displacement was recorded. The stiffness of the spring was obtained as 4479 *N/m* and the weight of the spring as 0.13 *kg*. Other essential properties of the spring include mean *D* and wire *d* diameters of 25 *mm* and 4 *mm*, 50 *mm* free length and 8 *mm* pitch. It is essential to determine the maximum shear stress of the spring for fatigue design. This is obtained as

$$\tau_{max} = k_w \frac{8FC}{\pi d^2} \tag{5}$$

where *F* is the weight acting on the spring, which is computed as the sum of the eccentric assembly and the table top weights as 307.84 *N*. The spring index *C* is obtained as

$$C = \frac{D}{d} \tag{6}$$

and the Wahl stress factor k_w as

$$k_w = \frac{4C-1}{4C-4} + \frac{0.615}{C} \tag{7}$$

Hence, the maximum shear stress is calculated as 33.07 *MPa*. This is significantly lower than the allowable maximum shear stress of 42 *MPa*, so the design is within a good limit.

2.1.5 Numerical Modelling of the Kinematics of the Equipment

A numerical model of the seven degrees of freedom motion of the designed system was developed by applying physical laws. The physical model of the system is represented in Fig. 4. The system has two different types of masses, namely sprung mass and unsprung mass. The sprung mass is the portion of the equipment supported by the springs which include the table surface and its attachments (i.e. the eccentric assembly). The unsprung mass is mainly contributed by the four springs. Overall, these masses constitute seven degrees of freedom in the system.

The vertical displacements of the unsprung masses are represented as x_{21}, x_{22}, x_{23} and x_{24}. By applying D'Alembert principle, assuming uniform spring stiffness, the equations of motion at the corners can be expressed as

$$\begin{aligned} m\ddot{x}_{21} + kx_{21} &= 0 \\ m\ddot{x}_{22} + kx_{22} &= 0 \\ m\ddot{x}_{23} + kx_{23} &= 0 \\ m\ddot{x}_{24} + kx_{24} &= 0 \end{aligned} \tag{8}$$

where *m* is the unsprung mass of the springs and *k* the spring stiffness. The sprung mass has three vibration modes, namely pitch, roll and vertical. The pitch mode θ rotates around a transverse axis (angular displacement), the roll mode φ rotates around a longitudinal axis (angular displacement) and the vertical mode *x* twists and oscillates about a vertical axis (vertical displacement). The equation of motion for the vertical, pitch and roll vibration modes of the table surface are respectively obtained as

$$\begin{aligned} m_s\ddot{x} - kx_{21} - kx_{22} - kx_{23} - kx_{24} &= 0 \\ I_\theta\ddot{\theta} + (kx_{21} + kx_{24})a - (kx_{22} + kx_{23})b &= 0 \\ I_\varphi\ddot{\varphi} + (kx_{23} + kx_{24})c - (kx_{21} + kx_{22})d &= 0 \end{aligned} \tag{9}$$

where m_s and I are the mass and the mass moment of inertia of the sprung masses. a, b, c and d are as shown in Fig. 4.

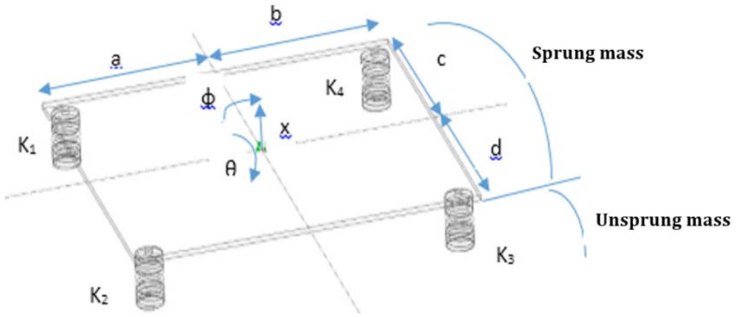


Fig. 4: The Physical Model of the Vibration Exciter.

3. Fabrication of the Vibration Exciter

The components of the device were machined and assembled with proper tolerance, dimensioning and specification with respect to the design analysis and drawings. Material selection was based the design parameters, material availability and cost in that order. Specifications of the main components of the device are summarized in Table 1.

Table 1: Part Lists and Specifications of the Device Components

Component	Material	Specification	Quantity
Surface Table	Mild Steel	480 mm × 480 mm × 5 mm	1
Lower Table	Mild Steel	200 mm × 200 mm × 5 mm	1
Frame	Mild Steel	50 mm × 50 mm × 450 mm	4
Shaft	Mild Steel	Ø40 mm × 200 mm	1
Compression Spring	Steel	Ø30 mm × 70 mm	4
Disc Plate	Mild Steel	Ø150 mm	1
Eccentric Mass	Mild Steel		3
Bearing	Steel	Ø40 mm	2
Electric Motor		1 HP	1
Rubber Foot	Rubber	50 mm × 50 mm × 5 mm	4

The fabrication procedures of the main components are outlined as follows:

a) *Frame*

The frame is made from angle iron. It was introduced to withstand the dynamic load reactions of the system without causing deflections. The frames were cut from a blank bar into the required sizes using a cutting disc and were joined as required by welding using E115 mild steel electrode.

b) *Shaft and Coupling*

The shaft was machined on the lathe machine by turning a solid cylindrical blank into the required shaft diameter and length. The shaft was then coupled to the electric motor by centre boring one end of the shaft 19 mm diameter 40 mm deep. This is equivalent to the diameter of the electric motor shaft. The motor shaft was then inserted into this shaft and both were bolted together for firmness and to avoid misalignment when in operation. This shaft transmits angular rotation and torque from the electric motor to the vibration exciter with the aid of the bearings attached on it.

c) *Disc and Eccentric Mass*

The disc plate was machined into the required size through marking-out and cutting operations. A 40 mm diameter hole was made at the centre of the disc for attaching it to the shaft. Also, three holes of diameter 6 mm at angle 45° to one another were drilled, at points 30 mm, 50 mm and 65 mm from the edge of the disc, for attaching the eccentric loads.

d) *Surface Plate, Lower Plate and Bracelet*

The surface plate and the lower plate were machined into the required sizes from a blank. The lower plate is acting as a support for the electric motor. The two plates were connected together rigidly by 4 bracelets of 190 mm long. The bracelets were welded to the upper plate, while they were bolted to the lower plate. This is to allow for easy maintenance of the device.

3.1 Brief Description and Operation of the Vibration Exciter

The device is designed to generate vibratory motion using eccentric masses. Under the influence of the external excitation, the system masses move and set up inertia force. The main rotating shaft of the device is set into unbalance dynamically thereby inducing the cyclic or regular vibratory motion to the system.

A circular disc is fixed to the shaft coming from the electric motor. Eccentric loads are attached to the disc in such a way that it could be detached and attached easily as shown in the design. The distance from the centre at which the mass is placed to the shaft can also be varied and adjusted. This made it possible for the parameters of the vibration to be varied even at a constant speed of motor.

The arrangement is attached to a surface plate hanging beneath from the table surface while the table surface is supported on flexible elements (i.e. springs) at the four corners of the frame. This induces an oscillatory motion to the surface table. The force required to set up the vibratory motion is therefore generated by the electric motor and eccentric load, so minimal or no force is lost or transmitted to the surrounding structures (frames).

The transmission is direct from the electric motor via the shaft to the table surface through the bearings. These therefore eliminate the use of meshing gears, and so help to reduce noise and provide simplicity in the design and operation of the equipment.

4. Results and Analysis

Testing was performed on the developed vibration exciter in order to evaluate its performance and functionality. Initial test produced a satisfactory performance.

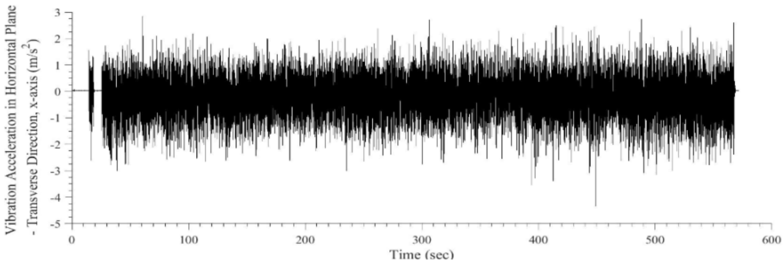
4.1 Acceleration and Natural Frequency Measurements

Accelerations of the vibration exciter, for vibrations along different directions, were measured using an accelerometer and oscilloscope set-up. The set-up displays the acceleration waveform with respect to time. The speed of the electric motor was monitored with a DT − 2235B+ Contact Surface Digital Tachometer. Results obtained from the oscilloscope were post-processed in MATLAB. The accelerometer has a sensitivity of 2048/count with a 16-bit resolution. Measurements were taken after running the equipment for about ten minutes in order to ensure a steady-state measurement.

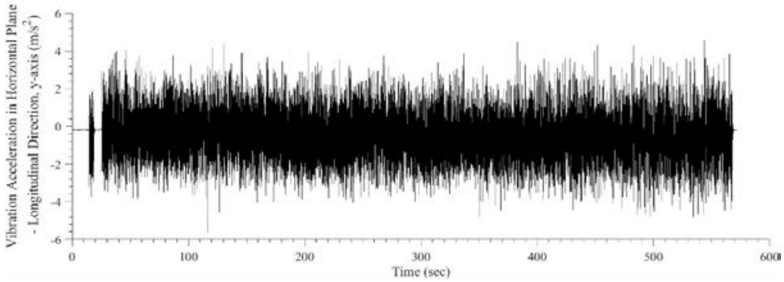
The accelerations of the equipment vibration along the principal x, y and z axes are presented in Fig. 5. Random sinusoidal signals are obtained in all the results. A maximum acceleration of 8 $\mathrm{m/s^2}$ is obtained along the z-direction compared to maxima of 3 $\mathrm{m/s^2}$ and 4 $\mathrm{m/s^2}$ obtained along the x and y directions respectively. Highest acceleration range of 10 $\mathrm{m/s^2}$ is also obtained along the z-direction compared to 7 $\mathrm{m/s^2}$ and 9 $\mathrm{m/s^2}$ obtained along the x and y directions.

The vibration angular accelerations of the equipment about the pitch θ and roll φ directions are presented in Fig. 6. Random sinusoidal signals are also obtained. A maximum angular acceleration of 40 rad/s is obtained for vibration about the pitch axis compared to 20 rad/s obtained about the roll axis. Also, the pitch axis has a higher angular acceleration range of 80 rad/s as against 60 rad/s about the roll axis.

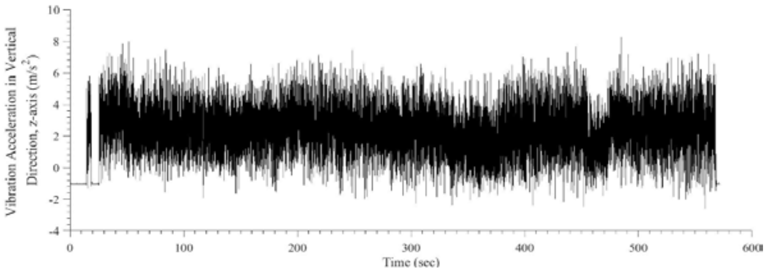
Natural frequencies are calculated using the model equations developed in Section 2.2. The natural frequencies at the four spring's locations are obtained by solving Eq. (8) dynamically. Similarly, solving Eq. (9) gives the natural frequencies of the surface along the vertical, pitch and roll directions respectively. To solve Eq. (8), the mass and stiffness of each spring are required. As presented in Section 2.1.4, the spring mass and stiffness were obtained as 4479 N/m and 0.13 kg respectively. Substituting these into Eq. (8) gives Eq. (10)



(a) *x*-axis

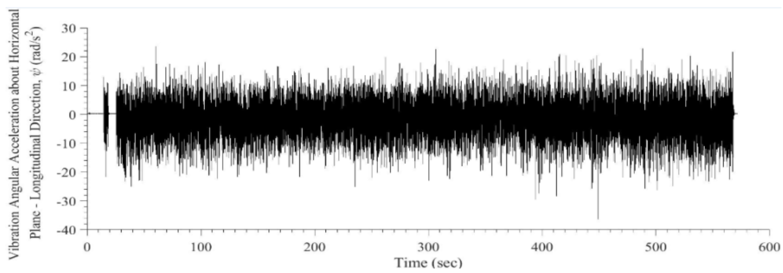


(b) *x*-axis

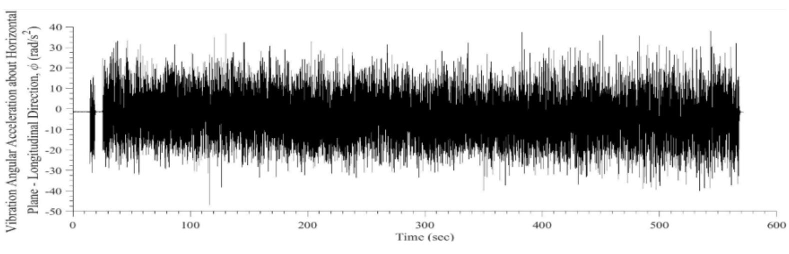


(c) *x*-axis

Fig. 5: Vibration Accelerations along the Principal Axes



（a） θ-axis



（b） φ-axis

Fig. 6: Vibration Angular Accelerations about the Pitch θ and Roll φ Directions

$$0.13\ddot{x}_{21} + 4479x_{21} = 0; \qquad 0.13\ddot{x}_{22} + 4479x_{22} = 0;$$
$$0.13\ddot{x}_{23} + 4479x_{23} = 0; \quad \text{and} \quad 0.13\ddot{x}_{24} + 4479x_{24} = 0 \tag{10}$$

Solving these equations gives a similar natural frequencies at the four locations of the springs as 29.5 *Hz*. To solve Eq. (9), the total sprung mass m_s of the equipment, which is constitutes of the eccentric assembly, the machine top surface and the eccentric masses, is obtained as 31.83 *kg*. The mass moment of inertia of pitch I_θ and roll I_φ vibration modes are obtained by summing the moment of inertia of the main component parts. These are obtained as $I_\theta = 0.863\ kgm^2$ and $I_\varphi = 0.556\ kgm^2$. Substituting these and other parameters into Eq. (9) gives

$$31.83\ddot{x} - 4497x_{21} - 4497 - 4497x_{23} - 4497x_{24} = 0$$
$$0.863\ddot{\theta} + 1075x_{21} + 1075x_{22} - 1075x_{23} - 1075x_{24} = 0 \tag{11}$$
$$0.556\ddot{\varphi} + 1075x_{24} + 1075x_{23} - 1075x_{22} - 1075x_{21} = 0$$

Solving these equations, the natural frequencies of the machine top surface along the vertical, pitch and roll directions are obtained as f_x = 3.78 *Hz*, f_θ = 7.94 *Hz* and f_φ = 9.89 *Hz* respectively. The presented results indicate that the developed machine has successfully satisfied the proposed hypothesis of being able to measure vibration characteristics along different degree of freedom.

4.2 Stress Analysis

Stress analysis was conducted on the vibration exciter top surface, which carries the vibration load. The surface is designed for stress rigidity. The analysis was carried out using the SolidWorks CAD software. It was assumed during analysis that no geometrical irregularities occur in the component parts of the device. For instance, the springs contact points are modelled as stationary pivot points to avoid stress raisers (usually caused by holes or gaps). The vibration load on the table surface was modelled by subjecting the surface to a distributed acceleration force. Acceleration forces of 200 N, 250 N, 300 N and 350 N were considered. The stress analysis results for these acceleration forces are presented in Fig. 7.

The factor of safety of the table surface, for an applied vibration load, is obtained as the ratio of maximum stress to working stress. It was observed that, at the point of maximum stress values, the top surface has minimum safety factors of 2.14, 2.51 and 2.68 when the surface is subjected to vibration loads of 250 N, 300 N and 350 N respectively.

5. Concluding Remarks

A vibration exciter plays an essential role in improving the physical and mechanical properties of parts produced from manufacturing operations like casting and welding. In this work, a vibration exciter, which has the capability of attaining seven degrees of freedom, was designed, fabricated and tested. Using eccentric mass, vibration was generated and induced on the surface plate of the device. The vibration parameters were able to be adjusted by changing the mass and/or the eccentricity distance, depending on the degree of vibration needed. The device performed satisfactorily as it was able to generate different vibration parameters. The main contribution of this work has therefore been the development of an eccentric mass exciter whose eccentric force is varied using the unbalanced mass and its eccentric distance, and which generates vibration characteristics along different degrees of freedom. The project can be extended by incorporating a speed regulation device with the electric motor for the purpose of varying the motor speed especially during the machine operation. This would further enhance the variation of the vibration parameters.

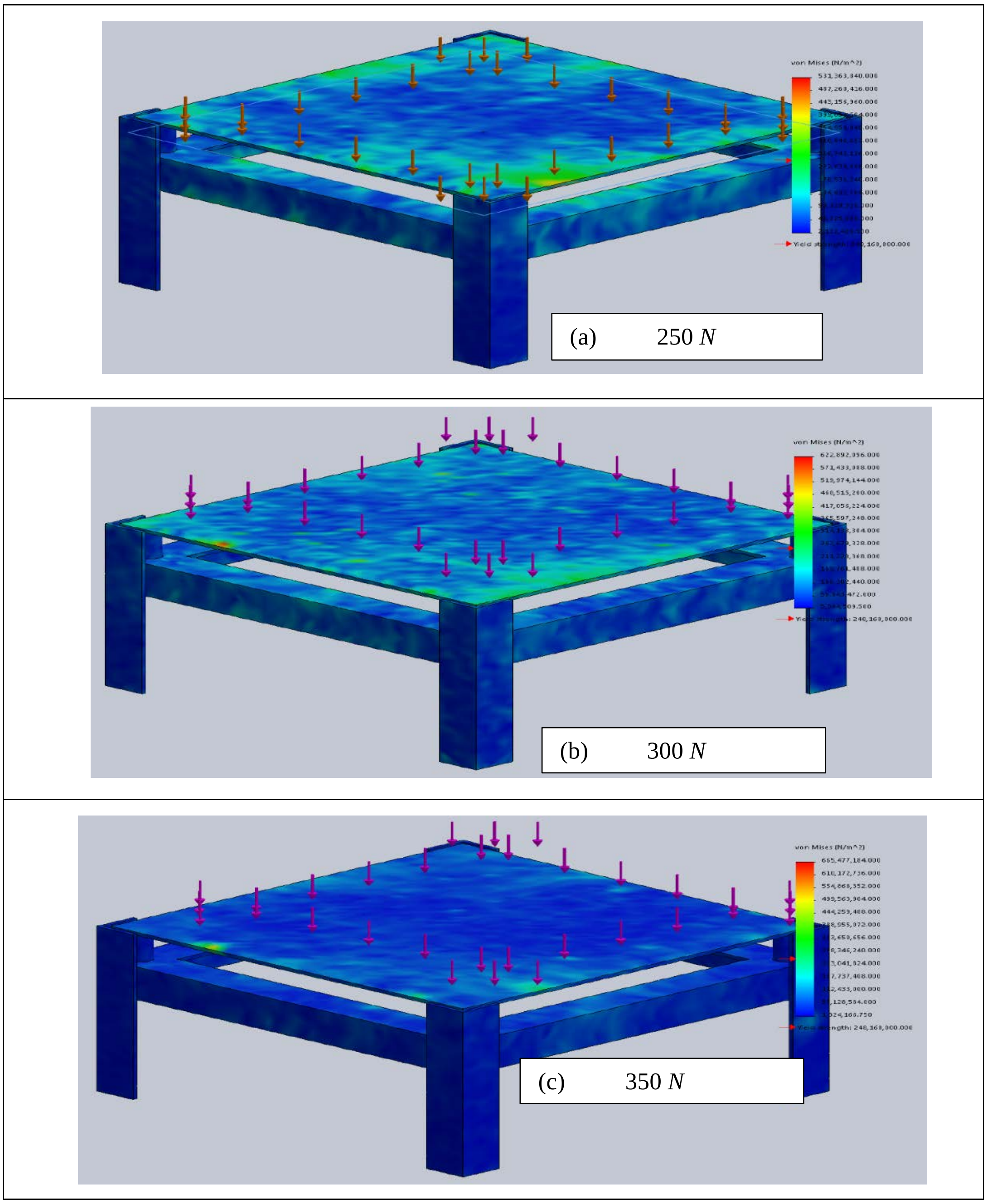


Fig. 7: Stress Analysis of the Vibration Exciter Top Surface at different Vibration Loads

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