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Design and Fabrication of a Mass Balancing Machine

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Abstract: **This work is aimed at producing a mass balancing laboratory apparatus that will serve for the demonstration of the principle of balancing of rotating masses which is a fundamental part of the study of Theory of Machines in engineering. The machine was constructed with the locally available materials and fabrication methods, and consists of a shaft, a pair of bearings, metal sheets, springs, dampers, a 0.25KW one phase motor, set of detachable balance mass blocks and a transparent safety dome to cover the moving components. The dome prevents them from harming users especially if failure occurs. ANSYS simulation tests results showed that they were within acceptable stress limits. The study established that locally sourced materials could be used to produce excellent Mass Balancing Machine at a low production cost.**

Keywords: **Static Balance, Dynamic Balance, Imbalance, Laboratory Apparatus.**

1. INTRODUCTION

Machines comprise of links or members in relative motion, some reciprocating while others rotate. Many different machines and machinery that find application in various fields including construction, machining, aviation and mining consist of rotating parts. In these machines with rotating parts, defect in configuration of the rotating part especially relating to the axis of rotation results in imbalance (Ranjit et al., 2018). The major consequence of this imbalance is vibration leading to noise, increased stresses on bearing and possible failure of the machine component, be it bearing, shaft or rotor. According to Ranjit et al., (2018), imbalance related vibrations account for approximately eighty percent of all vibration problems and they lead to the failure of the rotating equipment component in operation. The vibration produced by mass imbalance in a system can be acceptable up to a certain level beyond which it should be eliminated or minimized (Singiresu, 2015).

Engineers have attempted to proffer solutions for balancing problems in machines. Canadian engineer, H. Martinson was in 1870 granted what was probably the first patent for a balancing machine for a rotor that was mounted isotropically on soft coil springs, driven by a universal joint shaft. Franz Lawaczeck in his paper “zur theorie und konstruktion der balanziermaschine” (Theory and design of balancing machine), the “Lawaczeck model” was capable of achieving a balancing quality equivalent to a center of gravity displacement of 0.001mm, which could still today be perfectly accurate for many applications. With improvement in technology cars and other machines got faster. Unbalance became a real challenge. High speed bearing of aircrafts that may operate at over 20,000 rpm may become catastrophic if there is unbalance in any of its members. As force is directly proportional to speed according to $F = m\omega^2 r$, doubling the speed of

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rotation will quadruple the resulting vibrational force; this is the extent of damage that can be caused if unbalance prevails in rotating members.

It is very important that those studying engineering gain insight into the cause of undesirable vibrations of rotors and understand the static and dynamic unbalance conditions of the system (Nisbett, 1996). Proper understanding of balancing principles will help students of engineering to be better prepared to solve real life challenges relating to mass balance in rotating members of a machine and also know the consequences corresponding from its negligence in the design of mechanical systems. This work is geared towards producing a mass balancing apparatus capable of demonstrating the different kinds of unbalances.

2. MATERIALS AND METHODS

2.1 Machine Description

The mass balancing apparatus comprises the following components: a set of balance masses, a shaft operated by a system of pulleys and belt powered by an electric motor, a pair of ball bearings, springs, a bench and a resting plate. A transparent safety covering, covers the assembly to protect the operator from injury that may result at the event of failure.

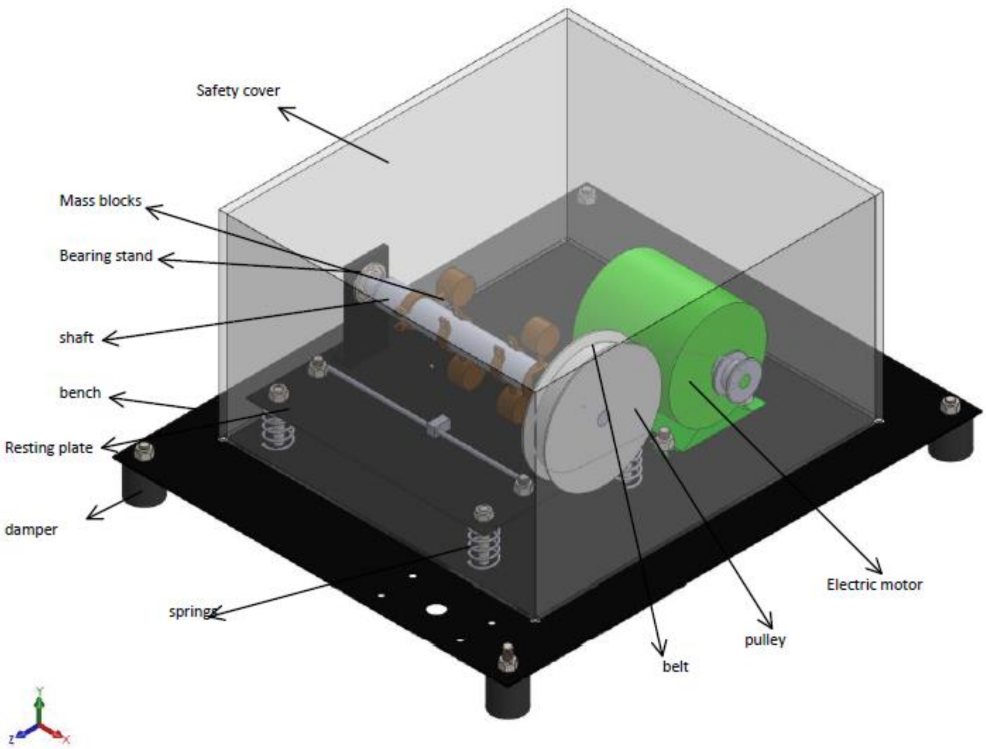

Fig. 1. Diagram of the mass balancing apparatus and its components.

2.2 Design Requirements

This design should put the following into consideration; the ease of attachment and detachment of balance masses, the strength of fasteners used to attach the masses, the safety of the machine during operation, the visibility of the rotating components during operation.

2.3 Material selection

Material selection uses design requirements through performance and shape properties to select the best combination of materials and processes that will perform the necessary task at the least cost. While selecting the materials for any design, due consideration must be given to the method of production or fabrication as the case may be. Table 1 below shows the list of machine components, materials selected and criteria for the selection.

Table 1: Material Selection

Machine component	**Material selected**	**Selection criteria**
Safety covering	Perspex (poly methyl methacrylate)	Transparency, toughness, fatigue strength.
Shaft	Mild-steel	Stiffness, strength, machinability, cost and availability.
Bearing stand	Mild-steel	Availability and ease of machining.
Bench	Mild-steel	High stiffness, fatigue strength, machinability, cost and availability.
Resting plate	Mild-steel	Availability and ease of machining.
Balance masses	Mild-steel	Availability, high density, strength, machinability.
Flexible mount	Music-wire	High resilience and reliability.
Belt	Fiber & rubber	Availability

2.4 Machine Design

2.4.1 Formulation of mathematical model

The system can be modeled as mass damper spring system with eccentric loading under the action of rotational force (Ralph, n.d.).

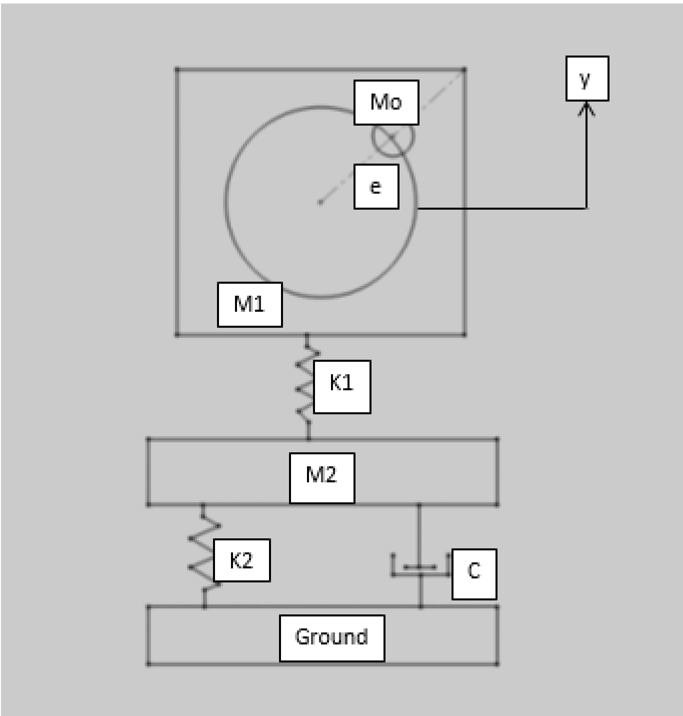

Fig. 2. model of the mass balance apparatus

y_{m1}= displacement of the M_1 mass element, y_{m2}= displacement of the M_2 mass element, m_o= sum of the eccentric masses on the shaft or mass of rotor, e = eccentricity, m_{m1}= mass of the M_1 mass element of the system, m_{m2}= sum of mass of the M_1 & M_2 mass element of the system, c = material damping, k = material stiffness.

$$(m_{m1} + m_o)\ddot{y}_{m1} + 4k_1 y_{m1} = m_o e\omega^2 \sin \omega t \quad (1)$$

$$m_{m2}\ddot{y_{m2}} + 4c\dot{y_2} + 4k_2 y_{m2} = TR \times m_o e\omega^2 \sin \omega t \quad (2)$$

$$y_{m1} = \frac{m_o e\omega^2}{-(m_{m1} \times \omega_n{}^2) + k_1} \quad (3)$$

$$y_{m2} = \frac{\frac{m_o e\omega^2}{k_1}}{\left[1-\left(\frac{\omega}{\omega_n}\right)^2\right]^2 + \left[2\zeta\left(\frac{\omega}{\omega_n}\right)\right]} \times \sin(\omega t - \varphi) \qquad (4)$$

Equation (1&2) represents the vibration model of the mass balancing apparatus, while (3&4) represent their respective solutions.

Where the amplitude force $(F_o) = m_o e\omega^2$ (5).

Angular speed of the system $\omega = \frac{2\pi N}{60}\left(\frac{\text{rad}}{\text{s}}\right)$ (6).

Natural frequency of the system $\omega_n = \sqrt{\frac{k}{m}}$ (7).

Force transmitted to the supporting foundation $F_t = ky + c\dot{y}$ (8).

Transmissibility $TR = \left|\frac{F_t}{F_o}\right|$ (9).

Desired reduction in transmissibility $R = 1 - TR$ (10).

$$\frac{\omega}{\omega_n} = \sqrt{\frac{2-R}{1-R}} \qquad (11).$$

$$\zeta = \frac{c}{2\times m_{m2}\omega_n} \qquad (12)$$

$$\varphi = \tan^{-1}\frac{c\omega}{k_2} \qquad (13)$$

2.4.2 Design of balance mass blocks

Design decisions:

- Mild-steel of yield strength$(s_y) = 270MPa$ (according to AZoM, 2012)
- Torque $T = 1.2Nm$,

$$\tau_{max} = \frac{S_y}{2n} \qquad (15)$$

$$\tau = \frac{16T}{\pi d^3} \qquad (16)$$

From (15) and (16), d = 0.5cm. Therefore, the minimum diameter of the 200g balance mass block will be 0.5cm.

2.4.3 Shaft Design

Design decisions: pulley diameters for motor and shaft respectively are D_1 = 3.3cm, D_2 = 12.8cm, motor speed N_1 = 1420rpm, motor power P = 0.25KW, factor of safety n = 2.5, key factor K_f=2, Mild-steel of yield strength$(s_y) = 270MPa$. Assumptions: coefficient of friction μ = 0.25, belt angle β = 20˚, length of belt (x) = 22cm.

$$V = \frac{\pi D_1 N_1}{60} = \frac{\pi \times 0.033 \times 1420}{60} = 2.45ms^{-1}$$

Tension on the both sides of the pulley = T_1 & T_2

$$P = (T_1 - T_2)V$$

$$\frac{P}{V} = T_1 - T_2 \qquad (17)$$

$$T^1 - T^2 = \frac{250}{2.45} = 102N \qquad (18)$$

$$log_e \frac{T^1}{T^2} = \frac{\mu\theta}{sin\beta} \quad (19)$$

$$\theta = (180 - 2\alpha) \times \frac{\pi}{180} \quad (20)$$

$$\alpha = \sin^{-1}\frac{D_2 - D_1}{2x} = \sin^{-1}\frac{0.12 - 0.033}{2 \times 0.22} = 11.4^{\circ}$$

From (20), $\theta = \frac{\pi}{180} \times \left(180 - (2 \times 11.4)\right) = 2.74rad$

From (19), $log_e \frac{T_1}{T_2} = \frac{0.25 \times 2.74}{\sin 20} = 2.00 \quad \therefore \frac{T_1}{T_2} = e^2 = 7.4098$

Hence, $T_2 = \frac{T_1}{7.4098}$

Putting T_2 into (18), $T_1 - T_2 = 102 \quad \therefore T_1\left(1 - \frac{1}{7.4098}\right) = 120$

Hence, $0.86504T_1 = 102 \therefore T_1 = 117.$

Putting T_1 into (18), $T_2 = T_1 - 102 = 117.913 - 102 = 15.913N$

$$F = T_1 + T_2 = 117.913 + 15.913 = 133.826$$

Resolving F into horizontal and vertical components $\theta = \tan^{-1}\frac{6.2}{21} = 16.45^{\circ}$

$$F_y = F \sin\theta = 133.826 \times \sin 16.45^{\circ} = 37.897$$

$$F_x = F \cos\theta = 133.826 \times \cos 16.45^{\circ} = 128.348$$

For a shaft simply supported at two ends and having a 5cm overhang and a pulley attached at the overhanging end and four rotating mass blocks 5cm apart from each other also attached to the shaft.

Force analysis

The two important equations for equilibrium are:

- The sum of forces in any given direction is zero,$\sum_n^{i=1} F_i = 0$ (21)
- The moment about any point is zero $\sum_n^{i=1} M = 0$ (22)

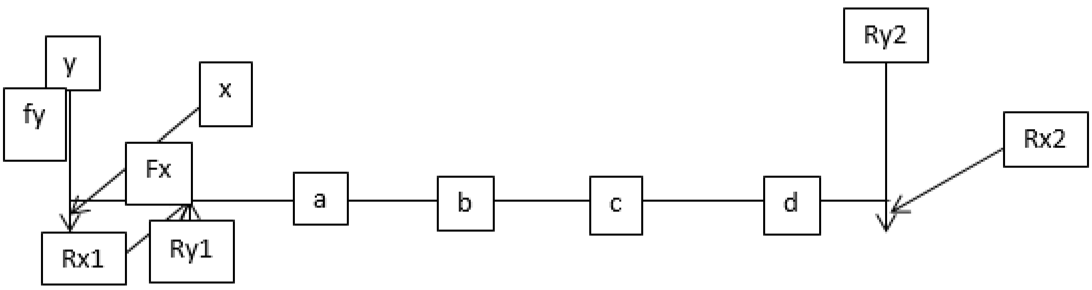

Fig. 3. Diagram showing forces on the shaft with torque applied at points a ,b ,c and d.

Applying (21) & (22), $R_{x1} = 150.477N, \ R_{x2} = -22.129N, R_{y1} = 44.427N, \quad R_{y2} = -6.53N$

$$\frac{N_1}{N_2} = \frac{D_2}{D_1} \quad (23)$$

From (23) speed of the shaft was derived $N_2 = 366.1rpm$

$$P = \frac{2\pi NT}{60} \quad (24)$$

From (24), for$N = N_2$, shaft torque $T = 6.25Nm$

Bending moment = $F \times d$ (25)

Therefore bending moments at $R_{x1} = 6.417Nm$ & $R_{y1} = 1.89Nm$. Combined bending moment at $R_1 = 6.6895Nm = \sigma_{bmax}$.

Torque dues to the rotation of the balance mass blocks =$T_q = mV^2 = 1.2Nm$. Maximum shear stress = $T_{max} = 6.52Nm$

Applying maximum shear stress theory (MSST)

$$\tau_{max} = \frac{S_y}{2n} = 54MPa$$

$$d_{min} = \left[\frac{16}{\pi\tau_{max}} \times \sqrt{K_m M^2 + K_f T^2}\right]^{\frac{1}{3}} \quad (26)$$

Applying (26) where

$K_m = 1.5,$

$M = \sigma_{bmax} = 6.6895,$

$K_f = 1.5,$

$T = T_{max} = 6.52Nm,$

minimum diameter of shaft, $d_{min} = 1.15cm.$

2.4.4 Design of Bearing stand

Design decisions: Mild-steel of yield strength$(s_y) = 270MPa$, Factor of safety (n) = 2.5, width = 5cm, with factor of safety (n) = 5.

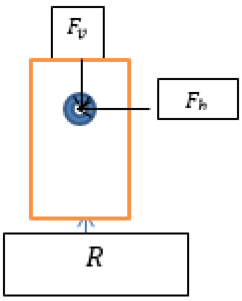

Fig. 4. A diagram showing the forces acting on the column member.

From Fig. 4, W = weight of shaft =6.5N, F_T = belt tensioning force = 44.427, F_o= amplitude force of vibration

Normal stress $(\sigma_n) = \frac{P}{A}$ (27)

Where P = force load, A = area.

Total vertical force acting on the member = F_v = W + F_t+F_o = 68.59N

Horizontal force F_v = 150.477N , hence bending moment due to horizontal force M_h = 12m4Nm.

Bending stress $(\sigma_b) = \frac{My}{I}$ (28)

Where M = bending force, y = distance from the neutral axis to the outer surface of member, I = principal moment of inertia.

For thin sheets, bending moment $(\sigma_b) = \frac{6M}{t^2}$ (29)

$$A = bt \quad (30)$$

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Where t = thickness of material and b = width of material.

$$\sigma_{all} = \sigma_b + \sigma_n \quad (31)$$

Allowable stress = $\sigma_{all} = \frac{S_y}{n}$ (32)

Where S_y = yield strength of material, hence allowable stress on member (from 31) $\sigma_{all} =$ 54MPa

From (30) minimum thickness of material $(t) = 1.18mm$

2.4.5 Design of resting plate

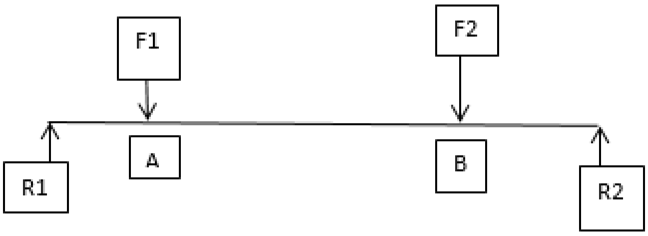

Fig. 5. Diagram showing forces acting on the plate.

Design decisions: Mild-steel with $s_y = 270MPa$, Factor of safety (n) = 5

From Fig. 5, $F_1 + F_2 = R + W_b$, r= reaction due to bearing stand at A & B =34.23N, W_b = weight of bearing stand = 4N.

Therefore $F_1 = F_2 = 38.23$. From (25), bending moment is at A = 6.77Nm & 12.4Nm.

Force analysis

From (21) & (22), $R_1 = 24.49N, R_2 = 32.4N$

Maximum bending force = M=12.4

Maximum shear force = 51N,

from (27), (29), (31) & (32), t = 1.17mm.

Therefore, minimum thickness of the sheet metal = 1.17mm.

2.4.6 Design of spring (according to Richard and Keith, 2011)

Design decision: music wire of G =81.7GPa, E = 200GPa, Coil diameter (D) = 22.65mm, Spring diameter (d) = 1.5mm, number of turns (N_t) = 6, factor of safety (n) = 2.5.

Given: spring load (P) is the sum of dead weight of the mass components and the maximum amplitude force from (3.2.4) = 52 N spring index(C) =11.92.

Assumptions: arbitrary wire diameter (d) = 15mm, clash clearance (r_c) =20%

Where A = a coefficient, b = an exponent, d =wire diameter. Values for A & b are gotten from Richard and Keith (2011, p 517).

From (3.3.19), S_u=2426.65MPa

Yield strength in tension $(S_{ys}) = 0.450S_u = 970.66MPa$

$$\tau_{max} = \frac{S_{ys}}{n} \quad (33)$$

From (33), τ_{max} =388.26MPa

Spring index$C = \frac{d}{D}$ (34)

Spring rate $k = \frac{P}{\delta} = \frac{dG}{8C^3N_a}$ (35)

Where number of active coils = N_a,

From (34) $\mathrm{k} = 1909.4\mathrm{N/m}$ and $\delta = 18mm$, Solid height of spring $(h_s) = (N_a + 2)d = 8.8mm$, Free height of spring $(h_f) = (p \times N_a) + 3d = 69.3\mathrm{mm}$, P = pitch of coil = 11mm.

2.4.7 Design of bench

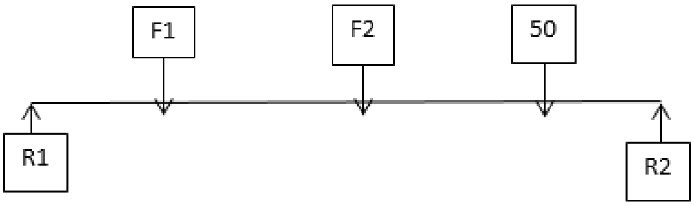

Fig. 6. Diagram showing forces acting on the bench.

Design decisions: mild-steel of $s_y = 270MPa$, factor of safety(n) = 5

$F_1 = 53.11N$ & $F_2 = 46.197N$

Force analysis (according to Richard and Keith, 2011)

From (3.3.6) & (3.3.7), $R_1 = 46.31N, R_2 = 97.997N$

Maximum bending force (M) = 22.67Nm

Maximum axial stress = 97.997N

From (3.3.12), (3.3.16), (3.3.17) & (3.3.18), t = 1.58mm

Therefore, the minimum thickness of the sheet metal for the bench (t) = 1.58mm

2.4.8 Design of Safety cover (according to Richard and Keith, 2011)

Design Decisions: Perspex fiber glass s_y = 2000Mpa, E = 72GPa (according to E-Glass Fiber, 2019), length of material(l) = 30cm, factor of safety(n) = 5.

Given: mass of impact object (m) = 0.3kg

For a member subject to impact load, maximum stress

$$\sigma_{max} = \sqrt{\frac{m \times V^2 AE}{L}} \quad (36)$$

From (3.3.13) allowable stress (σ_{all}) =400MPa where A = impact area = t × b and V = velocity(m/s).

For unit width, b = 1.

From (36) t =2.7mm

The minimum thickness of the fiber glass material is 2.7mm.

2.4.9 Motor capacity

According to Okolie et al., (2019)

$$P = m\omega^2 r^2 \quad (37)$$

Where m = mass of rotating shaft (kg) and ω = angular velocity of shaft (rad/s)

Machine construction; in completing this project, a number of operations were performed on the raw materials in order to bring them to the desired finished stock. Each of the components were designed and fabricated according to due fabrication

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process as displayed in Fig. 7 below. Some of those operations are also discussed below. Shaft construction; the shaft was fabricated on a lathe machine by reducing the stock mild steel shaft material to the final required dimensions. To get to the final desired dimensions of the shaft, a number of operations were done performed on the stock material, including phasing and turning. Care was taken to operate the lathe machine at the prescribed speed for mild steel material. Resting plate and Bench Construction; the resting plate and bench were produced from mild steel sheet metals. In bringing the stock material to its final dimension, cutting, drilling and filling operations were performed on the stock material also ensuring that the right tools, speeds and safety measures were adhered to. Construction of balance mass; the balance masses were constructed by attaching round standard masses to a mild-steel frame. Bill of engineering measurement and evaluation; while selecting the materials and methods for this project, the following considerations were made. (i) Reliability of the materials in service (ii) Material availability. (iii) Cost effectiveness.

2.5 Software analysis of machine components

A series of test were carried out on the machine components using the mechanical simulation software ANSYS to determine the suitability of the material and design parameters proposed for use in production of the apparatus. Firstly, full 3D models of the machine components were generated with SolidWorks, then they were imported into the ANSYS environment for analysis yielding the following results shown in the discussion section.

3. RESULTS AND DISCUSSION

The result of the stress analysis simulation test on the mild steel shaft in Fig. 8a shows that the limits of the stress on the shaft is between 0.104MPa and 0.938Mpa which is well below the yield stress of mild-steel(270Mpa). Fig 8b shows the result of the total deformation simulation test on the mild-steel shaft with deformations between 0-0.217μm which is insignificant.

Table 3.1: Analysis of shaft

Simulation Type	Static structural
Mesh type	Tetrahedron
Material	Mild-steel
Number of elements	638
Number of nodes	1301

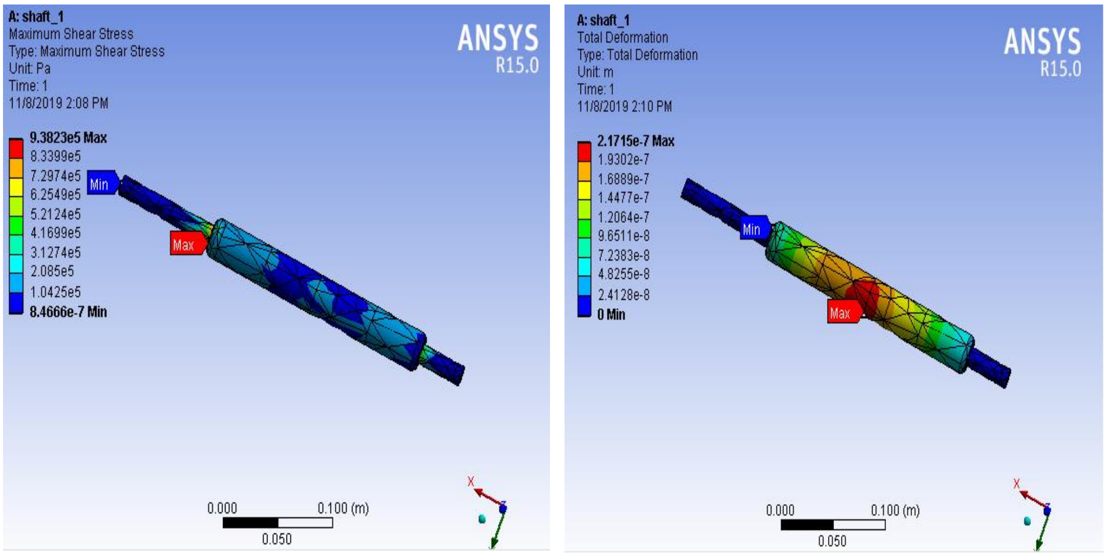

Figure 8; Representative of (a) stress analysis of shaft and (d) deformation analysis of shaft.

The results of the ANSYS simulation test on the bearing support shows from Fig. 9a that the stresses on the member lies between 5.9KPa to 3.06MPa which is well below the yield stress of mild steel(270MPa). Fig. 9b shows that the bearing support will have a deformation between 0-3.67μm which is insignificant.

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Table 3.2: Analysis of bearing stand

Simulation Type	Static structural
Mesh type	Tetrahedron
Material	Mild-steel
Number of elements	3226
Number of nodes	6320

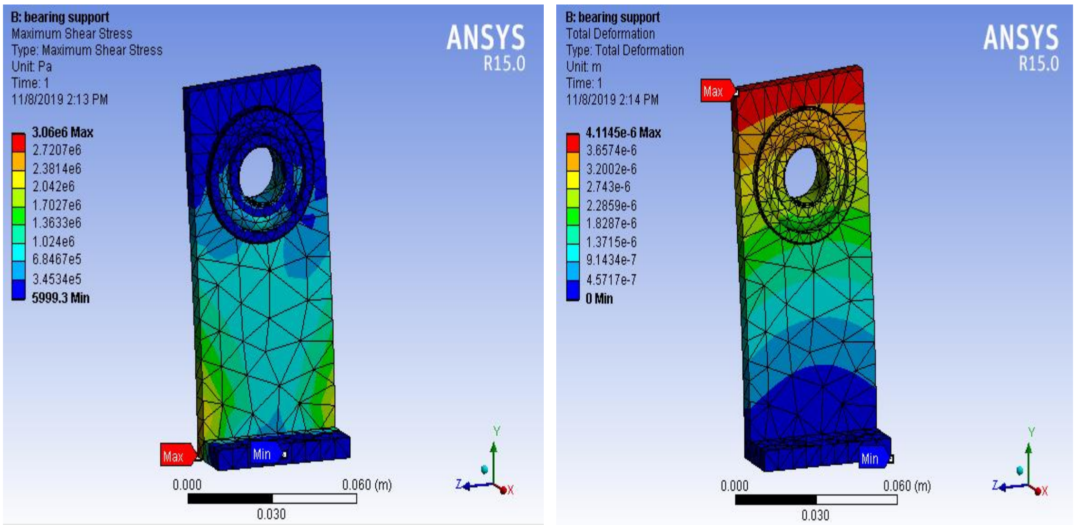

Figure 9: Representative of (a) stress analysis of bearing stand and (b) deformation analysis of bearing stand.

The ANSYS simulation results of the mild-steel resting plate shows in Fig. 10a that the plate will be subject to stresses between0.69KPa to 202.6MPa which is below the yield strength of the mild-steel. Fig. 10b shows that the plate will have a maximum deformation of 0.52mm which is manageable.

Table 3.3: Analysis of resting plate

Simulation Type	Static structural
Mesh type	Tetrahedron
Material	Mild-steel
Number of elements	269
Number of nodes	2121

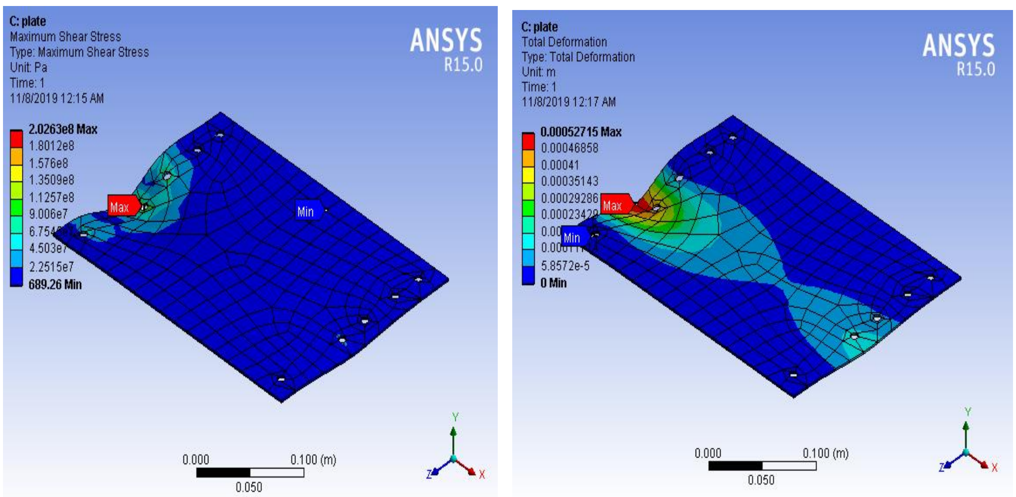

Figure 10: Representative of (a) stress analysis of resting plate and (b) deformation analysis of resting plate.

The ANSYS simulation test results of the mild-steel bench show from Fig. 11a pressure between 17KPa to 44.4Mpa acting on the mild steel bench which is well below the yield stress of mild steel. Fig. 11b shows a maximum deformation of 2mm along the center hence the need for reinforcement along that axis.

Table 3.4: Analysis on bench

Simulation Type	Static structural
Mesh type	Tetrahedron
Material	Mild-steel
Number of elements	13672
Number of nodes	27973

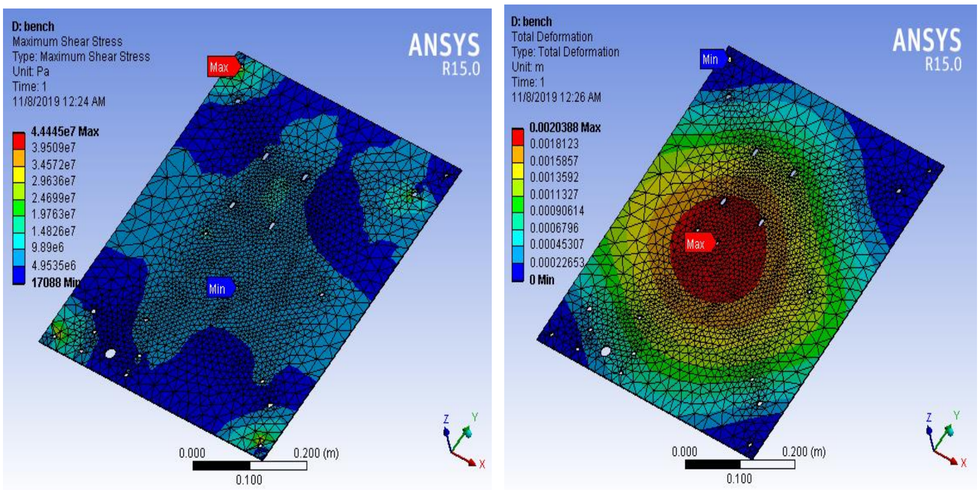

Figure 11: Representative of (a) stress analysis of bench and (b) deformation analysis of bench.

The result of the ANSYS simulation test on the mass balance block shows in Fig. 12a that stress between 5.7MPa to 51.65MPa is acting on the block which is well below the yield strength of mild steel. Fig. 12b shows that a maximum deformation of 0.136 mm is acting on the block which is manageable. These results show that the selected material and design parameters will serve the desired purpose as all their stress and deformations are well within the allowable limits.

Table 3.5: Analysis of mass balance block

Simulation Type	Static structural
Mesh type	Tetrahedron
Material	Mild-steel
Number of elements	673
Number of nodes	1547

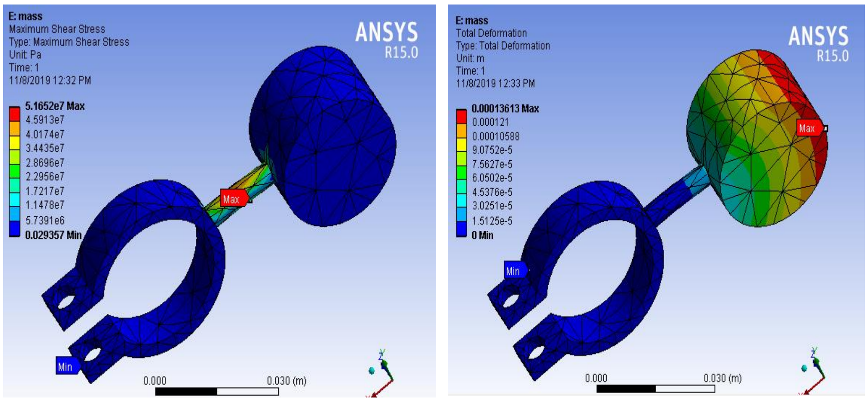

Figure 12: Representative of (a) stress analysis of mass balance block and (b) deformation analysis of mass balance block.

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4. CONCLUSION

This study successfully undertook the design and fabrication of a mass balancing apparatus for use in the demonstration of experiments on the balancing of rotating mass, employing adequate material selection and design methodology. The device powered by a 0.25KW motor via a belt mechanism, turns the masses attached to the shaft which is mounted on the bearing stand. The shaft and mass system generates a vibration force, depending on the arrangement of the masses on the shaft; this vibration force is then observed as it is transmitted through the spring system to the machine bench. The whole system is covered by a transparent safety cover so as to avoid accident that might follow an eventual failure. During the design, the machine components were analyzed to simulate their performance and observe behavior; hence the design was found to be in order. It was established that locally sourced materials could used to produce a mass balancing apparatus at a low cost which is relatively cheaper than apparatus at the market.

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