DESIGN AND MANUFACTURING OF WHEEL BALANCING MACHINE

A Final Year Project Report

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ABSTRACT

Many methods have been employed in past to overcome the issue if an unbalance in wheels which include Merrill balance method which used light for balancing purposes, while the wheel remains mounted on the vehicle. Other methods include the free rotation of the wheel, if the same point comes to bottom repeatedly then that implies the presence of extra mass at that point. Presently, a dedicated wheel balancing machine is used for this purpose which serves to balance the wheel both statically and dynamically. It comprises of a set of sensors, mounted on two different planes, and interfaced together, which detect the imbalance after rotating the wheel for a certain period of time at a constant RPM. After the processing of the results from these sensors, a final result in the form of mass to be added and position of the mass is given to the user. As limited information was available online regarding the manufacturing of this machine, we had to buy a used machine and perform revere engineering on it. The major challenges were faced in interfacing the two piezo sensors and optical encoder together. As in Pakistan, these types of machinery are imported from the foreign countries including China and Europe which costs more than 1 lac, there is a need of local manufacturing and repairing services of the wheel balancing machine which will cause a major drop in its price. The machine also needs to meet the standards of quality and accuracy to compete in the market. In light of the above-mentioned reasons, his project aims to design and assemble a locally manufactured, highly accurate and cost-effective wheel balancing machine.

Key Words: Static, Dynamic, piezo sensor, optical encoder, constant r.p.m, imbalance.

PREFACE

I and my group members' interest led us to visit Dr. Riaz Ahmed Mufti in search of a Final Year Project related to the field of designing. It was after visiting him numerous time that he offered us this project of making a wheel balancing machine. After briefly looking into the machine online, we found this project very interesting and decided to take it. We made use of our summer vacations and spent summer 2017 in doing research regarding the machine itself. The main areas we researched included history, methods of balancing, standards followed and types of imbalance. Most of this research became part of our literature review of this report.

As not much details were there online on the manufacturing of machine, we decided to buy a second-hand machine at low prices. Upon visiting the market of Islamabad, Rawalpindi, and Karachi, we were finally able to get an old machine from Karachi for PKR 16,000. It was after the arrival of this machine in Islamabad, our main work started. We did critical reverse engineering of the machine by studying its each and every component, its function and its purpose. In addition to this, we used the sensors of this machine to form a code on Arduino in which we interfaced piezo sensor and optical encoder to give us the extent and position of imbalance. Apart from this, the design of this machine was set as a benchmark and its solid structure and drive mechanism was designed, the details of which is mentioned in this report.

ACKNOWLEDGMENTS

As already mentioned, the manufacturing information was very difficult to extract from the internet. As a result, we had to rely on the machine we were studying and the people around us for technical knowledge. First of all, we would like to thank our supervisor Dr. Riaz Ahmed Mufti for guiding us through the course of this project. Secondly, we would like to thank Dr. Mian Ashfaq, our co-supervisor, who helped us with various problems and also kept us motivated. Thirdly, we would like to thank Dr. Samiur Rehman Shah who helped us in designing drive mechanism system. We would also like to thank our senior Talha Yousaf, who gave initial help regarding the sensors. And finally, we would also like to thank Mr. Abid, who works in a wheel balancing shop in Rawalpindi. Mr. Abid helped us in understanding the procedure of balancing and also provided us with valuable information with the help of his immense experience in this field. Without the part of these people, this project would not have been possible.

ORIGINALITY REPORT

I, Hammad Naseem, along with my group members, Abdul Moiz Farooq, Abdul Rafaeh and Haris Iftikhar, fully agree that the contents of this report were not directly copied from any sources, including internet and books. The sources which did help us during the course of this project are mentioned in the references section, whereas people who helped us during this project are acknowledged above. I hereby agree that this work is original. The report was also checked from 'Turnitin', and had only 7% similarity. A complete report of this is also available as a proof which can be provided upon request.

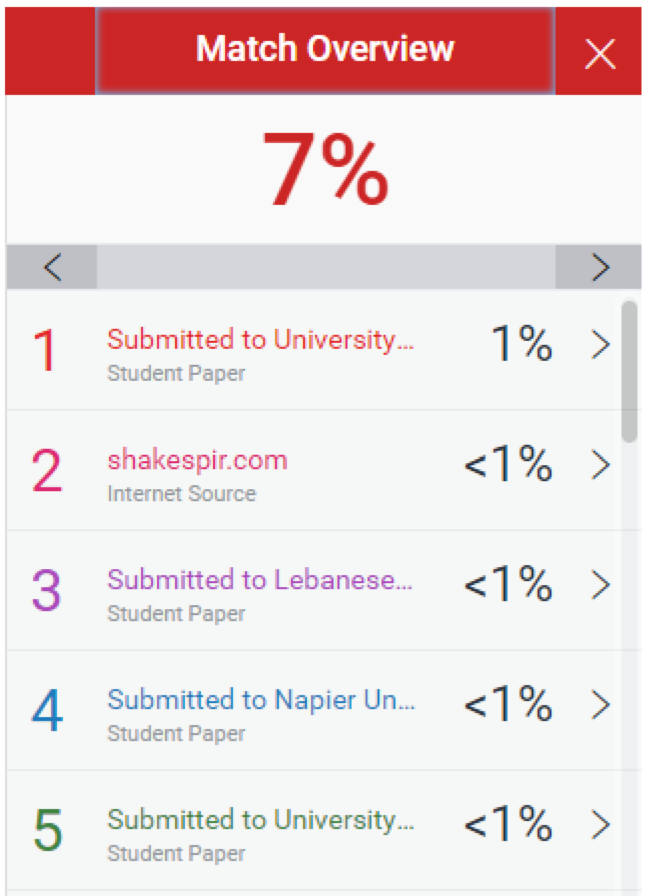

Figure 0: Match Overview

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ABBREVIATIONS

hp	Horsepower
rpm	Revolutions per minute
ISO	International Organization for Standardization
ANSI	American National Standards Institute
FOS	Factor of Safety
Hz	Hertz
PAMA	Pakistan Automotive Manufacturers Association

NOMENCLATURE

r	radius of wheel
W	width of wheel
L	distance of wheel from the reference plane
e	radius of unbalance mass on the rotor
G	Center of gravity
F	Force
i	terms for total unbalance in the rotor
b1	balance mass introduced in plane 1
b2	balance mass introduced in plane 2
ω	rotational speed in r.p.m
m_c	correction mass
m	Mass
M_y	moment in y axis
M_x	moment in x axis
F_x	force in x axis
F_y	force in y axis
V_c	vertical imbalance couple
H_f	horizontal imbalance force
V_f	vertical imbalance force
H_c	horizontal imbalance couple

CHAPTER 1

INTRODUCTION

The main objective of this project was to design and manufacture a wheel balancing machine that is not only cost-effective but also has the same accuracy as that of commercially available wheel balancing machine. We hope to achieve this objective by setting commercial wheel balancing machine available today as our benchmark and designing machine in such a way that the per unit cost of the machine can be reduced. We intend not to compromise on the quality but at the same time, we are looking for the cost-effective components to be used in our machine, so the main objective of our project can easily be achieved. The need for this project arises due to several reasons. Firstly, balanced wheels play an important role in determining the performance of an automobile. The unbalanced wheel does not only add up to unpleasant driving experience but also causes wear and tear of tires, which further leads to a bad fuel economy. Secondly, Pakistan automobile industry is growing exponentially. According to PAMA [1], there is a 4 % increase in the manufacturing of local cars and apart from these 3000 cars are being imported monthly from countries like Japan and China. This ever-growing size of Pakistan automobile industry demands a local manufacturer of wheel balancing machine. Currently, there is no local manufacturer of this machine in Pakistan. It is being imported from Europe and China costing 1-5 lacs. This amount adds up to the import bills of Pakistan, leaving a bad affect on Pakistan's economy. Another problem related to this machine is that imported machines are available but they cannot be easily repaired in Pakistan because there is no formal study being carried out on this machine and there is no local manufacturer in Pakistan. Moreover, no one has ever taken on a project like this because of its complexity and various fields involved. This project requires the designing of mechanical component (shaft and belt drive mechanism) as well as the designing of the control system and instrumentation part for the sensors involved. This machine also needs a user-friendly interface, so anyone with a least technical knowledge can operate this machine. Summing up, because of all these factors, no one has ever been able to commercially manufacture this machine in Pakistan. We have chosen this project because it's about time that Pakistan starts manufacturing its own wheel balancing machines rather than importing them. Our project is industry-funded, so we can easily commercialize this machine through various networks. We aim to manufacture a wheel balancing machine that can accommodate a wide range of wheel sizes, eliminating the need for different machines for different wheel sizes. Our target is to design a machine that has a long life and how we hope to achieve our target is by critically analyzing each aspect of our machine and designing each mechanical part according to the available standards.

CHAPTER 2

LITERATURE REVIEW

2.1 Types of balancing

Unbalance in the rotor is due to the uneven distribution of mass along the rotor, and this causes the rotor to vibrate. The unbalanced mass, when rotates, generates a centrifugal force. The centrifugal force rotates when mass rotates and it tries to move the rotor along the line of action of force. This force will be experienced by the bearings.

Balancing is the process of improving the mass distribution of the rotor, so that it does not generate any extra centrifugal force on the bearings. This can be done by either adding compensating masses to the rotor at given locations or by removing the fixed mass of material.

An object is said to be **statically unbalanced** when the center of mass and the center of axis of rotation of the object doesn't coincides. Static imbalance has a tendency to rotate due to the force of gravity.

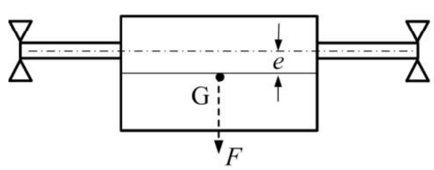

Figure 1: Static Unbalance

An object is said to be in the state of **couple imbalance** when, in the rotatory motion, the rotation axis does not coincide with the inertial axis of object.

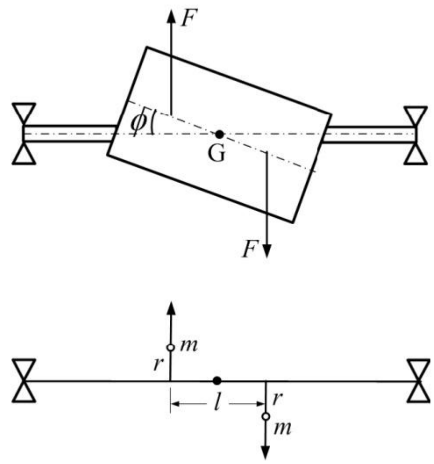

Figure 2: Couple Unbalance

Dynamic unbalance is the combination of static and couple unbalance, and it is the most common and frequent type of unbalance found in rotors. For correct measurements of dynamic imbalance, it is essential to measure the imbalance while the rotor is rotating.

2.2. Causes of imbalance

Common causes of imbalance include:

- Premature casting and casting defects
- Eccentricity
- Corrosion and wear
- Material build up in rotors used in material handling process
- Addition of keys and keyways in shafts
- Thermal distortion or stress distortion
- Deposits build-ups on rotor like lime or dirt
- Improper assembly of parts

2.3. Old methods for calculation of static and dynamic imbalance:

Static Balancing:

For static balancing a photoelectric tachometer probe and accelerometer is used to measure the extent of imbalance in the rotor. The tachometer is used to determine the rotational speed of the rotor while accelerometer determines the vibrations created by unequal mass distribution in the system, accelerometer produces a vibration signal that is amplified and filtered and compared with the tachometer reading to determine the phase of correction mass. The accelerometer also gives the vibration level reading to determine the correction mass.

A simpler method that is sometimes used for flywheels, etc., is illustrated by the Figure 3. In this method, a wheel is mounted on the perfectly horizontal shaft. If the wheel is in an unbalanced state, the heavy side will turn downward. After applying counterbalancing by adding extra weight or reducing heavy portions, if it will stand in any position it is said to be in or static balance.

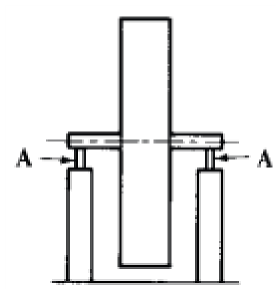

Figure 3: Simpler method

Dynamic Balancing:

In dynamic balancing, balancing in two planes is required, so two accelerometers are needed together with a photoelectric tachometer. A vibration signal is chosen from the two signals and is amplified and filtered and is compared with the tachometer reading to determine the phase. Now the same process is repeated for the other accelerometer vibration signal and in the end correction mass ad phase is determined.

2.4. History of balancing Machines:

In the past, it would take nearly two to three weeks of tiring labor work in order to balance rotors. Still, at greater speeds, there was the risk of damages due to the vibrations caused by the inaccurate balancing methods.

A Canadian Engineer H. Martinson was one of the first people to look into the issue of balancing. In the early 1870s, he was granted the first patent for a balancing machine [2]. In that machine, the rotor was mounted on coiled springs and was driven by a shaft with a universal joint. That machine had a small degree of accuracy. No official records were ever found whether this machine was actually built and it worked, or if it did, whether it was built in bulk quantity.

In 1908, Carl Schenck with Lawaczeck concluded a license agreement which remained valid up to 1940s [3]. This principle of balancing, named Lawaczeck model, was capable of a very accurate balancing which was equivalent to the displacement of the center of gravity up to one-hundredth of a millimeter, which is acceptable for many current applications.

Balancing machine by Norman Bassset (1909)

Referring to figure 4, this balancing machine has a main shaft (1) upon which a supporting system (2) rests, (2) is connected to the shaft with a universal bearing so that the supporting frame (2) can bend in any direction. Above (2) there is a frame that supports the rotor to be balanced. In this machine heavy or light side of the rotor is determined with respect to some assumed axis. When a rotor is placed on the frame (3) and its center of mass does not match with the center of the shaft (1). The main frame will bend in a direction indicating the heavy side of the rotor [4].

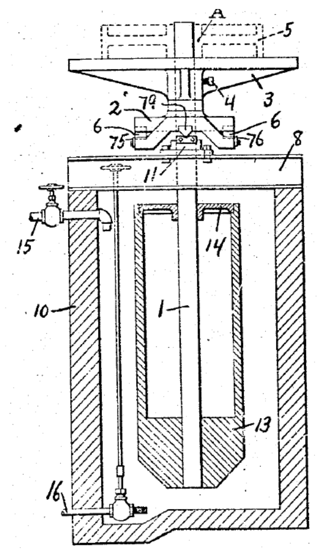

Figure 4: Basset’s model

Balancing Machine by P.F Hatch (1939):

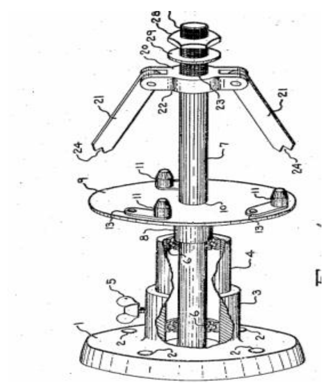

Figure 5: Balancing Machine by P.F Hatch

This balancing machine consists of a main shaft and a base which is connected to wall. The shaft supports a frame 9 which has pins 11 so that the wheels could be fixed to the frame 9. After the wheel has been mounted on the frame, it is made to freely rotate and eventually stop indicating the heavy point on the wheel. [5]

Merrill Wheel-Balancing System (1945):

The first dynamic wheel balancing machine was made in 1945 by the name of Merrill Wheel-Balancing System. [6] Machines before this were static balancing machine which balanced the rotor through static method of trial and error as shown below.

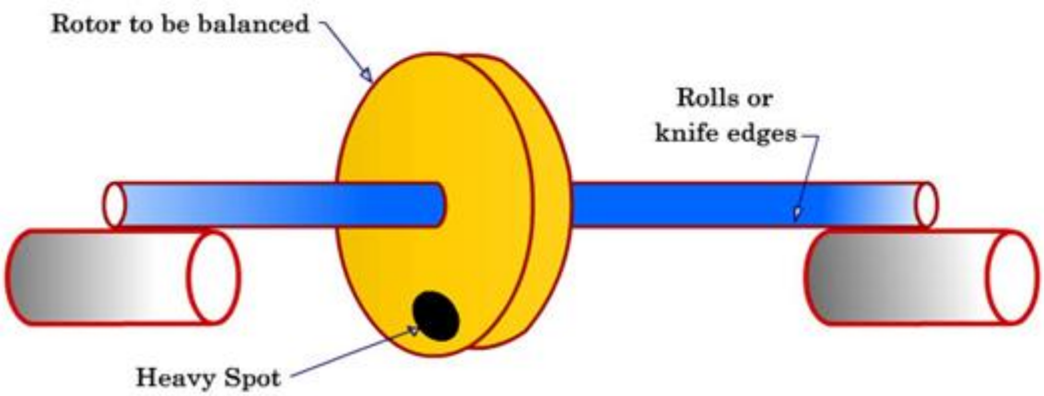

Figure 6: Merrill Wheel-Balancing System

The breakthrough in this invention was that it can measure the vibrational reading and its angular position on the rotor. The two instruments used were electronic pickup and stroboscope. Electronic pickup is the instrument used to measure the reading of the vibrations. It consists of a fixed magnet and a ferromagnetic material. The vibration of this material produces an electric current, just like the principle of electromagnetic induction which is used to tell us the extent of vibration. A signal was used to trigger a stroboscopic light which made the wheel appear to be still and not rotating. This light thus identified the point at which the balance weight should be placed. The point is initially placed to mark the reference point on the object.

Today these devices have been replaced by accelerometer/piezo electric sensors to measure to vibration and optical encoder to measure the position of the imbalance.

Road force machines:

In the current era, wheel balancing machines comes with a lot of extra features. This machine by Hunter [7] just not only solves imbalance, but also radial force related problems. This machine simulates the weight and forces applied to tires and wheels during driving conditions. This method detects tire imperfections and provides more accurate and detailed readings that allow more precise wheel balancing. It can also detect tire and rim runout. It has a diagnostic load roller which is coupled with the wheel while wheel is rotating.

Figure 7: Hunter's 'Road Force' Machine

2.5. Need of Wheel Balancing Machine

Wheel balancing is a vital part of maintaining the vehicle and the life of tires as well as bearings. Out-of-balance tires will wobble and cause a car to vibrate, and an imbalance even as small as of 10 grams is enough to cause a noticeable vibration in the steering wheel when the car is at a high speed of 100 km/h or more. These vibrations are also dangerous for the drivers and its passengers. These vibrations also result in stress being created in the lower ball joints, axles and other essential parts of the vehicle. This increased stress on these parts will lead to them wearing at a faster rate and will mean you will have to replace these parts a lot sooner than you would have had to with balanced tires. Balancing method will not fix or repair a bent wheel, out of round tire or irregular wear and cannot prevent vibrations from these problems.

Tires go out of balance for a number of reasons including mud and deposits on the wheel, tires weights falling off, or wearing of a tire over time.

Generally, it is recommended to have the tires checked for balancing at least every two years, or have your wheels balanced every 3,000 - 6,000 miles [8] and even more often in areas where roads are not well-maintained.

2.6. Inside Wheel Balancing Machines

A common and modern wheel balancing machine contains a drive mechanism, sensors, display with user interface, and a controller for motor.

The shaft is usually driven by belt drives and pulleys, in which a higher rpm from motor is converted to lower rpm for shaft to rotate. Sensors used are piezoelectric for measuring the imbalance force and optical encoder for measuring the position of imbalance.

Motor commonly used are low power AC motors with power up to 0.5 – 2 horsepower.

The parts and their mechanism are further discussed in detail in next chapter.

2.7. Standards:

Following are some of the ISO standards regarding balancing that we studied and aim to follow, and these standards are followed globally.

Table 1: Standards

Standard	Description
ISO 21940-2:2017 [9]	Defines the terms on balancing. It also complements ISO 2041, which is a general vocabulary on mechanical vibrations and shock.
ISO 21940-21:2012 [10]	It is applicable to those balancing machines that support and rotate rotors with rigid behavior at a certain balancing speed and that specify the quantity and position of a required unbalance alteration in one or more than one planes.
ISO 21940-11:2016 [11]	Establishes procedures and unbalance tolerances for balancing rotors with rigid behavior. It specifies the required number of planes, magnitude of permissible unbalance, and how to reduce errors in balancing process.
ISO 19499:2007 [12]	Introduce to balancing and directs the user through the existing International Standards related with rotor balancing. It also gives guidance on which of the standards should be used.
ANSI S2.19 [13]	Gives recommendations for determining unbalance and for specifying related quality requirements of rigid rotors. It is quite similar to ISO 21940-11:2016

2.8. Mathematical Model:

2.8.1. Dynamic Balancing:

The mathematical model for dynamic balancing is obtained from [14]

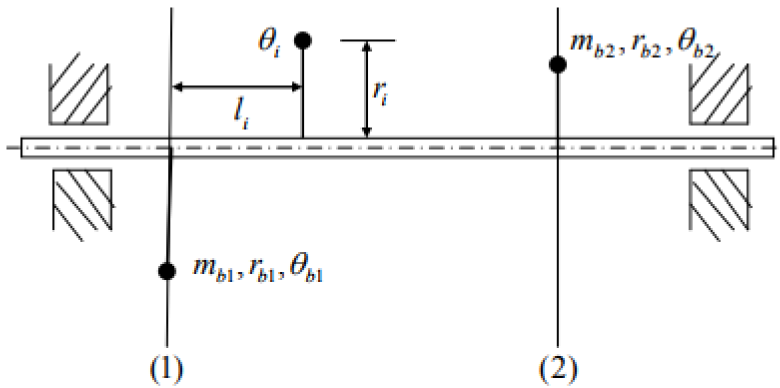

Figure 8: Dynamic Balancing

Double plane balancing is needed for rotors having significant thickness. It is done by introducing two separate masses in two different planes of the rotor at a known radius in different angular positions. The amount of mass and angular position is determined by unbalance force in the rotor. These masses are introduced in the rotor to nullify the effect of unbalance and make rotor fully balanced, Double plane balancing is the sum of static balance and couple balance.

Suppose a rotor in which there is unbalance mass present at radius r_i and at some angular position, then the equation for total unbalance in a rotor becomes

$$\frac{\vec{F}}{\omega^2} = \sum_{i=n}^{n} m_i r_i e^{j\theta_i} = \sum_{i=n}^{n} m_i r_i \cos\theta_i + \sum_{i=n}^{n} m_i r_i \sin\theta_i \qquad (1)$$

This unbalance is removed by introducing two different masses n two planes at different angular position. Now the forces due to the two masses introduced should be equal to the force due to unbalance mass present. The equation for static balance becomes

$$0 = \sum_{i=n}^{n} m_i \omega^2 r_i e^{j\theta i} + m_{b1}\omega^2 r_{b1} e^{j\theta_{b1}} + m_{b2}\omega^2 r_{b2} e^{j\theta_{b2}} \tag{2}$$

This equation can be further expanded as:

$$\sum_{i=n}^{n} m_i r_i \cos\theta_i + m_{b1} r_{b1} \cos\theta_{b1} + m_{b2} r_{b2} \cos\theta_{b2} = 0 \tag{3}$$

$$\sum_{i=n}^{n} m_i r_i \sin\theta_i + m_{b1} r_{b1} \sin\theta_{b1} + m_{b2} r_{b2} \sin\theta_{b2} = 0 \tag{4}$$

If for a given rotor the radius at which balance mass is to be introduced is known then the above equations have 4 unknowns, m_{b1}, m_{b2}, Θ_{b1} and Θ_{b2}. To solve for these unknowns, we need two more equations and those equations come from couple balance.

Assuming our reference plane to be plane 1 and taking moment about reference plane:

$$0 = \sum_{i=n}^{n} m_i \omega^2 r_i l_i e^{j\theta i} + m_{b2}\omega^2 r_{b2} l_{b2} e^{j\theta_{b2}} \tag{5}$$

The above equation can be further expanded as:

$$m_{b2} r_{b2} \cos\theta_{b2} = -\frac{\sum_{i=n}^{n} m_i r_i l_i \cos\theta_i}{l_{b2}} \tag{6}$$

$$m_{b2} r_{b2} \sin\theta_{b2} = -\frac{\sum_{i=n}^{n} m_i r_i l_i \sin\theta_i}{l_{b2}} \tag{7}$$

Now putting the equation (6) in equation (3) and equation (7) in equation (4), we get 2 equations for m_{b1}.

$$m_{b1}r_{b1}\cos\theta_{b1} = \frac{\sum_{i=n}^{n} m_i r_i l_i \cos\theta_i}{l_{b2}} - \sum_{i=n}^{n} m_i r_i \cos\theta_i \qquad (8)$$

$$m_{b1}r_{b1}\sin\theta_{b1} = \frac{\sum_{i=n}^{n} m_i r_i l_i \sin\theta_i}{l_{b2}} - \sum_{i=n}^{n} m_i r_i \sin\theta_i \qquad (9)$$

Dividing the equations (6) and (7) we get the angular position for the mass in plane 2.

$$\theta_{b2} = \tan^{-1}\left(\frac{\sum_{i=n}^{n} m_i r_i l_i \sin\theta_i}{\sum_{i=n}^{n} m_i r_i l_i \cos\theta_i}\right) \qquad (10)$$

Dividing the equation (8) and (9) we get the angular position for the mass in plane 1.

$$\theta_{b1} = \tan^{-1}\left(\frac{\frac{\sum_{i=n}^{n} m_i r_i l_i \sin\theta_i}{l_{b2}} - \sum_{i=n}^{n} m_i r_i \sin\theta_i}{\frac{\sum_{i=n}^{n} m_i r_i l_i \cos\theta_i}{l_{b2}} - \sum_{i=n}^{n} m_i r_i \cos\theta_i}\right) \qquad (11)$$

Now we have the angular position for both masses, so we can easily get the value of masses in both planes from the equations (5), (6), (7), and (8). These are basic equations for dynamic balancing of an ordinary rotor. The same concept is used in wheel balancing machine; the only difference is that the reference plane is some other predefined plane in the machine rather than the any plane on the wheel.

2.8.2. Static Balancing:

The mathematical model for static balancing is obtained from a book [15].

For single plane balancing, the total unbalance in the rotor is given by:

$$F = m\omega^2 e \qquad (12)$$

Where e is the radius of unbalance mass on the rotor and ω is the rotation speed of the rotor. Now the correction mass ‘m_c’ can be determined from the following equations and it should be placed 180^o from the unbalance mass, on radius ‘r’.

$$F = m\omega^2 e = m_c\omega^2 r \tag{13}$$

$$m_c = \frac{me}{r} \tag{14}$$

Mathematical model Used in Machine:

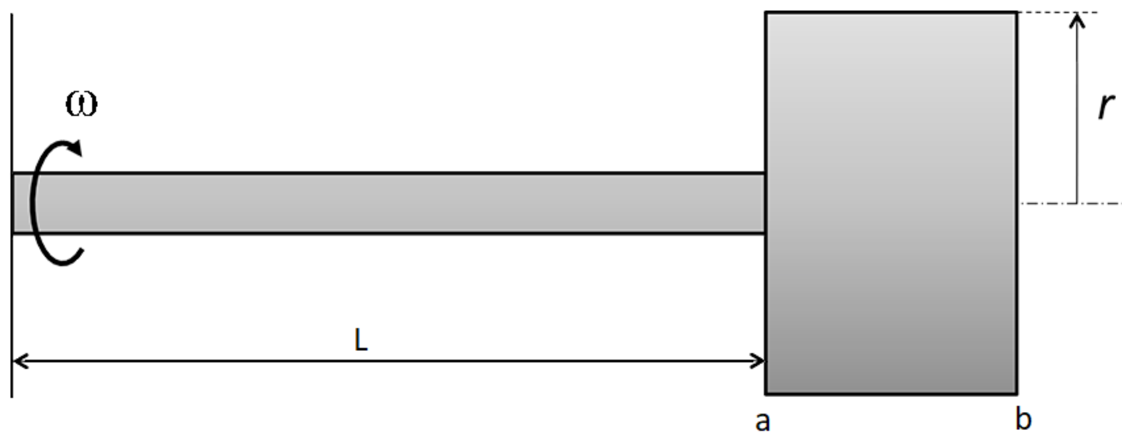

Figure 9: Wheel Balancing Machine Model

In wheel balancing machine same mathematical model is followed as that of dynamic balancing model. The equations are as follow:

$$\sum M_y = 0 \qquad m_a r\omega^2 l_a \sin\theta_a + m_b r\omega^2 l_b \sin\theta_b + V_c = 0 \tag{15}$$

$$\sum M_x = 0 \qquad m_a r\omega^2 l_a \cos\theta_a + m_b r\omega^2 l_b \cos\theta_b + H_c = 0 \tag{16}$$

$$\sum F_x = 0 \qquad m_a r\omega^2 \cos\theta_a + m_b r\omega^2 \cos\theta_b + H_F = 0 \tag{17}$$

$$\sum F_y = 0 \qquad m_a r\omega^2 \sin\theta_a + m_b r\omega^2 \sin\theta_b + V_F = 0 \tag{18}$$

In these equation V_f, H_f, V_c and H_c are the imbalance forces as calculated by the two piezo sensors one in horizontal plane, one in vertical plane. By solving the above equations, we get the value of masses and their angular position in both planes of a wheel.

2.9. Necessity of this project in the context of previous work

The necessity of this project arises due to several reasons. Currently, there is no local manufacturing available in Pakistan, and it is being imported at high cost. There is no formal study being carried out on this machine, and thus repairing is not easily available for the imported machines. Our project aim is to develop a cost-effective machine with the same accuracy as that of imported machines, following the globally recognized standards. This machine will also have a user-friendly interface, so anyone with a least technical knowledge can operate this machine easily. This project is necessary to promote and develop an efficient and local solution for expensive and imported wheel balancing machines.

CHAPTER 3

METHODOLOGY

3.1. Calibration of Piezo Sensor:

Piezo sensor is a device which gives voltage when a fluctuating force is applied to it. This sensor is used in commercial wheel balancing machines available today. Piezo is calibrated prior to the proper functioning of the machine. Its method of calibration is simple. First piezo is installed in the machine under the bearing housing, then a balanced rotor is placed on the shaft and a known imbalance mass is placed at a known radius on that balanced rotor. This unbalance mass generates an outwards centrifugal force which is transferred to the piezo sensor through the bearing housing, the maximum value of the voltage that the piezo generates due to this unbalance mass is noted down. Now the unbalance mass is increased and gain maximum value of voltage corresponding to this unbalance mass is noted down. This gives us a relation between the unbalance mass and voltage of piezo in the form of an equation, which will be used in our program to calculate the amount of unbalance whenever the machine is used.

3.2. Drive mechanism:

Design of the drive mechanism of wheel balancing machine is one of the major parts of our project. Drive mechanism consists of motor, belt drive mechanism, shaft and bearing. These mechanical parts determine the life of the machine, so it is essential that these are designed accordingly within the safety limits.

3.2.1. Selecting a motor:

The motor is the main component of the drive mechanism, which is driving the shaft through a belt drive mechanism. There are several factors that need to be considered before finalizing a motor. Motor selected should have enough starting torque to drive a stationary wheel mounted on the shaft. It should be strong enough to support varying loads on the shaft because all the wheels do not have same inertia. What happens when we select a weak motor is that when a heavier load is applied to the shaft, motor's speed decreases and it not able to rotate the wheel at the required balancing speed. This further causes motor to draw more current in it and this high amount of current can cause problems and damage motor's windings. The motor used in the machine we were reverse engineering was a single-phase A.C motor of 0.5 hp. and 1320 rpm. One of the reasons to select a motor with less horsepower is that our belt drive has a large speed ratio. This speed ratio was because of the fact that balancing speed of most of the machines ranges from 150-200 rpm, so he has to go from 1320-1400 rpm (motor) to 150-200 rpm (shaft on which wheel is mounted). Large speed ratio means large pulley diameter which adds to the cost and weight of the machine. We needed to use a motor with less horsepower rating because of the National Electrical Manufacturers Association (NEMA) minimum sheave (Pulley) requirements. NEMA is an association which has issued a range of minimum recommended sheave diameters for electric motors [16]. For a 1 Hp motor, the minimum recommended sheave diameter is 2.4 inches, so if we use the motor of 1 Hp and 1400 rpm we would require a large pulley of 16.8 inches, which would be not feasible. Another solution to this problem is that we use a large motor of 2 Hp but with a motor controller (Variable frequency drive). By using a VFD, we can rotate the motor at lower rpm than the rated rpm, so this reduces our speed ratio, and thus reduces the need for larger diameter pulleys. By keeping these factors in mind, we selected an A.C motor of 2 hp and 1425 rpm for our drive mechanism along with a VFD compatible with this motor.

3.2.2. Variable Frequency drive:

VFD is a device which is used to control an A.C motor. It controls the motor by changing the voltage and frequency provided to the motor. It is also known as variable speed drive because it can change the speed (RPM) of the motor by changing the frequency. Frequency and RPM of the motor are related as follow:

$$RPM = \frac{120 \times Frequency}{No.\ of\ Poles} \tag{19}$$

So, if VFD increases the frequency, the RPM of motor also increases and vice versa. We are using a VFD to reduce the speed the motor. Moreover, a VFD also increases life of the motor by reducing power consumed by the motor.

3.2.3. Working of VFD:

VFD works by first converting AC voltage into DC voltage by the help of the diodes [17]. This DC signal obtained has ripple which is removed by using a capacitor. Afterwards, this DC voltage is again converted back to AC voltage by help of transistor switches. The waveform of the voltage obtained is rectangular which perfectly acceptable for the motor but not safe for other appliances. The timing with which these transistor switches are turned on and off determine the frequency of the motor, thus determining the speed of the motor.

3.2.4. Belt drive:

Belt drive is needed for power transmission and for reducing speed to the required value. As our drive involves high speed ratio it would not be feasible to use flat belt drives because a flat belt drive is prone to slippage at high speeds. So, we designed a micro V belt drive for our machine. Micro V belt is used because they run smoothly and are oil resilient. Moreover, they are preferred in situation which require high speed application. For the designing of micro V belts a manual of Gates Cooperation [18] was used. The derivation of belt drive is as follow:

3.3.1. Starting point:

As we are using a VFD to reduce the speed of the motor from 1425 rpm to 700 rpm, the horsepower of the motor also drops to half, so we designed a belt drive mechanism by considering an A.C motor of 1 hp and 700 rpm

- Single phase motor of 1 Hp to drive a shaft.
- 700 rpm of motor.
- 204 rpm of the shaft.
- Center distance 12 inch (because of limited space).

3.3.2. Derivation:

- Service Factor:

  It is calculated on the basis of nature of driver and driven machines and service hours of the machine per day. It was noted to be 1.2

- Design Horsepower:

  Design horsepower is calculated by multiplying the horsepower of the motor with the service factor.

$$\text{Design Horsepower} = 1 \times 1.2$$
$$= 1.2 \text{ Hp}$$

- Selecting proper belt section:

  Now proper belt type appropriate for our requirements is chosen from a graph using design horsepower and rpm of the motor.

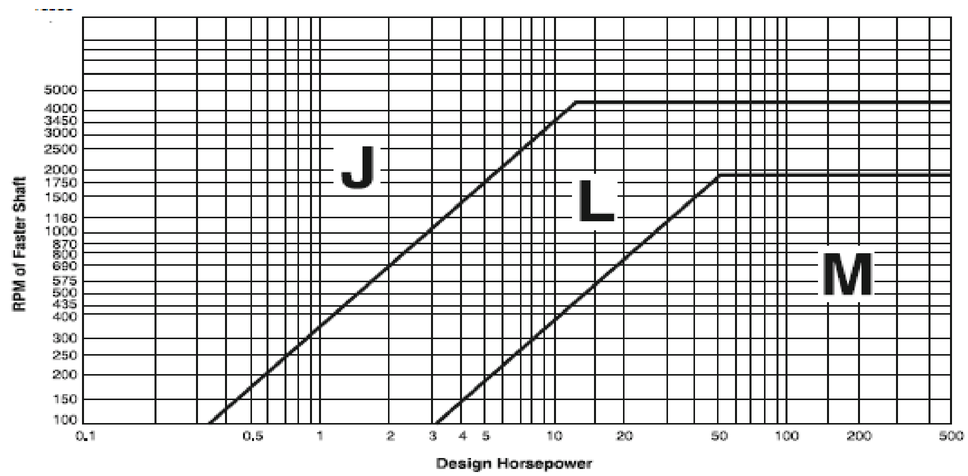

Figure 10: Selecting a V-Belt

Micro J belt is chosen from the above graph.

- Calculating speed ratio:

  Speed ratio is calculated using rpm of motor and shaft.

$$speed\ ratio = \frac{motor\ rpm}{shaft\ rpm} \tag{20}$$

$$speed\ ratio = \frac{700}{204} = 3.41$$

- Calculating diameter of Larger pulley:

  Now diameter of smaller pulley is assumed to be 3 in and diameter of the larger pulley is calculated from the speed ratio.

$$speed\ ratio = 3.41 = \frac{diamter\ of\ larger\ pulley}{diamter\ of\ smaller\ pulley} \tag{21}$$

$$diamter\ of\ larger\ pulley = 3.41 \times 3$$

$$diamter\ of\ larger\ pulley = 10.2\ in$$

- Calculating belt length:

  Center distance of 11.6 inches is taken from the manual and the pitch length of the belt is calculated by the following formula:

$$L = \sqrt{4C^2 - (D - d)^2} + \frac{D_L - d_S}{2} \tag{22}$$

$$L = 43.57\ in$$

- Calculating No. of belts required:

  Horsepower per belt is noted down from the table given in the manual, it is divided by design horsepower to get the number of belts required.

- Finalizing the belt:

  So, after this calculation 440J6 belt was chosen from the catalogue.

- Selecting pulleys:

  The pulleys needed for this drive mechanism should be statically balanced, in order to get better results from the sensors installed in the machine. If unbalanced pulleys are to be used in a drive mechanism of a wheel balancing machine, the unbalance force associated with these pulleys can cause some serious problems. So, after an extensive research statically balanced cast iron pulley were selected for our drive.

3.3.3. Shaft design:

Stepped shaft was to be designed for our machine. On one end of shaft pulley is mounted through which motor transmits power, while on the other end wheel is mounted. In the middle of the shaft there is a hub for the wheel to be mounted on the shaft. Shaft is subjected to both torsional and bending stresses. First maximum normal stress theory is followed to determine the diameter of the shaft. According to this theory, maximum stress in the shaft is given by the following formula:

$$\sigma_{b(\max)} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{{\sigma_b}^2 + 4\tau^2} \qquad (23)$$

Now bending stress in the above equation is given by:

$$\sigma_b = \frac{32M}{\pi d^3} \qquad (24)$$

And shear stress is given by:

$$\tau = \frac{16T}{\pi d^3} \qquad (25)$$

The equation of σ_b and τ is substituted in the first equation and the first equation is solved for the value of the diameter and the value is:

$$d = 34.6\, mm$$

Now maximum shear stress theory was applied to determine the diameter of the shaft. According to this theory, maximum stress is given by:

$$\tau_{max} = \frac{1}{2}\sqrt{{\sigma_b}^2 + 4\tau^2} \qquad (26)$$

Now again the equation of σ_b and τ is substituted in the equation of τ_{max} and the resulting equation is solved for the value of diameter and the value is:

$$d = 36.3\ mm$$

So, the final dimeter of the shaft is selected as d = 36.3 mm. The procedure followed for the derivation of the diameter of the shaft was taken from a [19].

3.4. Instrumentation and Control Part:

This is one of the major portions of the working of the wheel balancing machine and controls the way the machine works. It includes the data acquisition by the sensors, using that data to produce the required results and the way machine mechanically responds to the acquired values. Since we know that the imbalanced force produced is measured with the help of two piezoelectric sensors oriented such at 90o so that they detect the static and dynamic imbalance. Moreover, there is a rotary encoder which measures the angular position of the shaft and RPM of the mounted wheel.

In the following project, the sensors and control part of the machine is programmed using Arduino MEGA 2560. The Arduino Mega 2560 is a microcontroller board based on the ATmega2560. It has 54 digital input/output pins of which 15 can be used as PWM outputs. 16 analog inputs. a 16 MHz crystal oscillator, a USB connection, a power jack, as a header, and a reset button. The way it has been used with each sensor of the wheel balancing machine is mentioned in the report. The reason it has been used is to reduce the overall cost of the machine and comparatively simpler programming techniques as compared to the conventional processing unit.

3.4.1. *Piezoelectric Sensor:*

A piezoelectric sensor is the sensor which takes the piezoelectric effect into account to measure the changes in pressure and force by converting them into electrical charge. The piezoelectric sensor used in the wheel balancing machine is ceramic pressure sensor usually 2 inches in thickness sensitive to fluctuating pressure. There are two such sensors in a wheel balancing machine which are oriented orthogonal to each other connected via T-joint through bolts. Their orientation is shown as below:

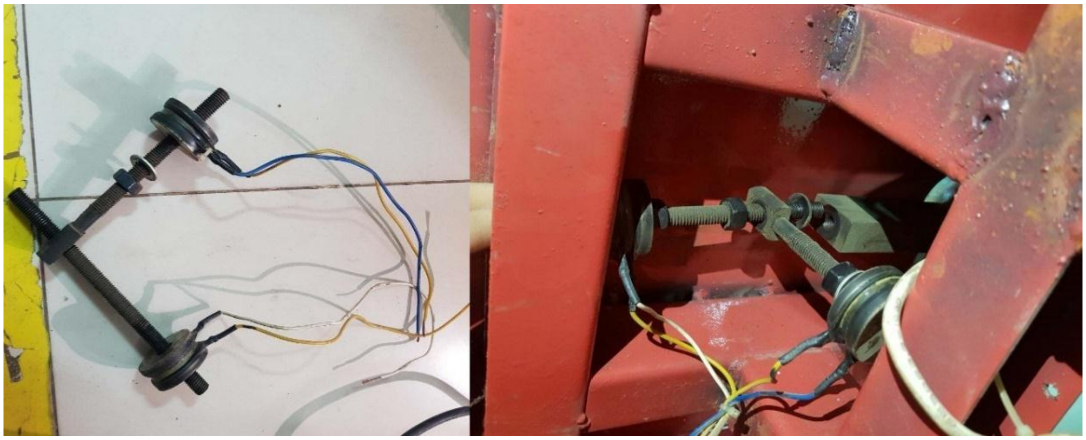

Figure 11: Piezoelectric Sensor orientation

The above-shown sensors were disassembled from the wheel balancing machine bought from scrap. The same sensor was being tested for the measuring the sensitivity and interfacing with Arduino. The sensors were first tested with an oscilloscope which showed a constant electrical noise of 50Hz while the knocks which the piezo sensor measures were superimposed on that noise. This noise needs to be eliminated for working purposes, otherwise, the signal will give an irregular reading at different stages thus resulting in a highly inaccurate wheel balancing machine. There are different methods used to eliminate this noise which includes the use of shielding wires, grounding the sensor and using the filtration. The following signal was obtained when the piezo sensor was connected to the oscilloscope.

The same sensor was then connected to the Arduino MEGA 2560 by connecting one of its lead to the one of the analog input of the Arduino while the other leading to the ground. The raw value obtained by Arduino is always in between 0 and 1023 means 1024 which comes from 2^{10}=1024 which is the property of Arduino. So whatever signal obtained is divided by 1023.0 in Arduino IDE and multiplied by 5000 to get the signal in voltages. The same electrical noise was obtained in the Arduino. This noise needs to be reduced to differentiate the original signal from that produced from noise.

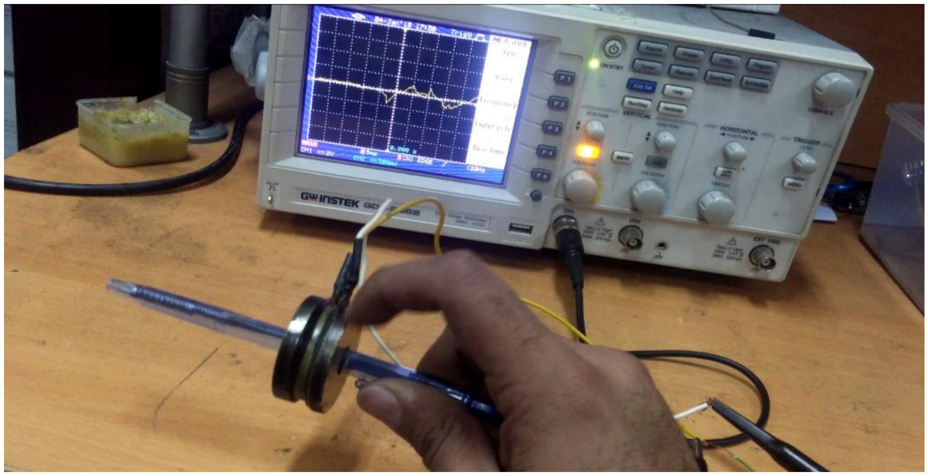

Figure 12: Oscilloscope Signal

Note that is also a considerable reduction of noise of the sensor when the sensor is tightly held near the center. This role is played by a bunch of washers held together by nut as shown in the above picture.

The method used here to reduce the noise was the method of digital filtration in Arduino, for which the filtration library provided by Arduino was used, in which high pass and low pass filters are available. [20]

The library is downloaded from the website of Arduino and then following libraries are added while programming the code, *FilterDerivative.h. FilterOnePole.h, Filters.h, FilterTwoPole.h, FloatDefine.h, RunningStatistics.h*

The filter frequency is set to 150 in our case with low pass filter being used which will allow the only the frequencies below 150 Hz to pass through to the final signal while the noise will get illuminated. However, this will also reduce the strength of our original signal. Therefore, the signal from the piezoelectric sensor was amplified using the circuit for voltage amplifier. Technically, we should be using charge mode amplifier since it does not include the capacitance of the cables involved and since piezo primarily produces the charges not the voltage, but since the IC (TVL2770 and TVL2771) [21] being used for charge mode amplification is not currently available in Pakistan. Therefore, for now, we are using voltage mode amplification in which the IC used is (LM741).

The circuit made for the voltage mode amplification is as follows [22]

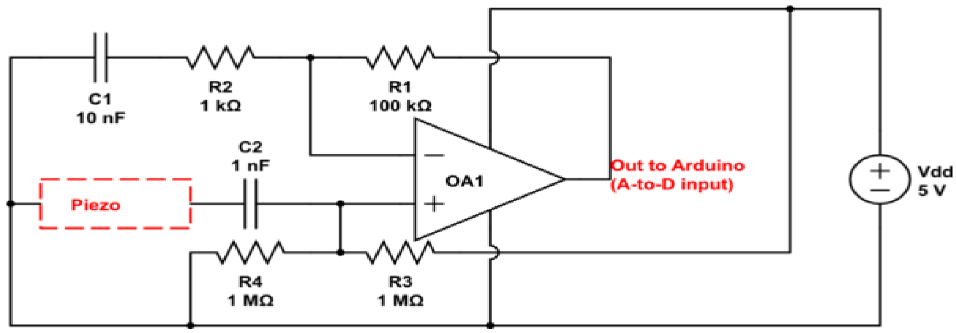

Figure 13: Amplifier Circuit

After the implication of the following circuit the final waveform obtained by knocking the piezo sensor while incorporating the digital filter as well. The raw value has been shown in the Figure 14. X-axis represents the time in milliseconds while the peaks show the presence of knocks.

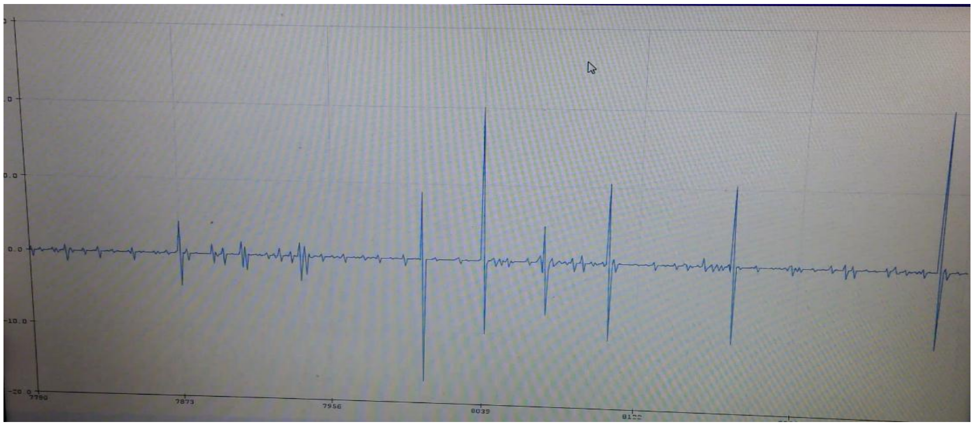

Figure 14: Piezo Sensor Value

3.4.2. *Rotary Encoder:*

An encoder is a device which is used to accurately measure the position, direction and velocity of the moving object with respect to some reference. It can be either absolute encoder or incremental encoder with respect to the reference of measurement. A rotary encoder does the same in angular terms. It is also known as shaft encoder since it is usually attached along the shaft to measure its position. Encoder is one of the most important component of the wheel balancing machine. It is attached right in front of the pulley. It consists of the 3 photo interrupters and an encoder plate with certain number of teeth depending upon the resolution.

3.4.3. *Photo interrupter:*

Photo interrupter acts as an optical switch having light emitting elements and light receiving elements that detects the light blockage by giving the high signal otherwise low.

Three of such interrupters are used in such a way that two of them are in line with each other separated by a certain distance so that their pulses are always out of phase by 90^o. The third photo interrupter is in between the two interrupters but not in line with the other two such that only one tooth pass between the two segments of the interrupter every revolution. The other two always have certain number teeth of encoder plate passing in between their segments. They produce the pulses such that one of the signal always leads, while the other one lags as shown in the Figure 15.

Here signal A is leading the signal B. In this case the shaft will be rotating clockwise which will also cause the encoder plate to rotate likewise. The orientation of encoder in the machine and the photo interrupters is shown in Figure 16:

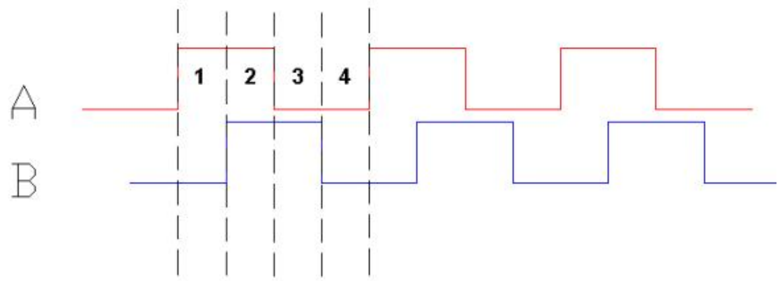

Figure 15: Photo Interrupter

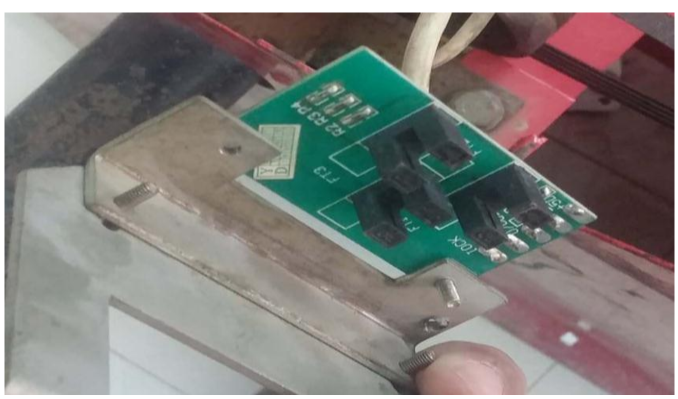

Figure 16: Encoder

3.4.4. ***Angular Positioning:***

The measurement of the position of the shaft with respect to some reference is always needed while balancing an imbalanced rotating object to know the exact location where extra amount of mass is present. Rotary encoder servers for this purpose. As we had mentioned that two of the photo interrupters of the rotary encoder are in line with each other and they are the ones used for measuring the angular position of the shaft with respect to some reference. The signals produced by both are out of phase by 90^o. So, there are always four different possibilities for the pair of signal, which are

A- high **B-** low

A- high **B-** high

A- low **B-** low

A- low **B-** high

Since there are 32 teeth in the encoder plate which being inspected. So, it will be giving us resolution of

$$Resolution = {360}\Big/{\frac{32 \times 4}{2}} \tag{27}$$

So, Resolution = 4.64^o.

This resolution can be increased by making an encoder plate with more teeth but likewise a photo interrupter of higher accuracy and sensitivity will be needed to read the Highs and Lows of the signal.

Another issue that arises while measuring the position is when the tooth keeps blocking the photo interrupter and the counter keeps adding up thus increasing the angular position, and we don't want this to happen for this another temporary variable is defined in the code which compares each upcoming value to the previous value and doesn't implement if it is the same value as the previous one. Moreover, in order to avoid the garbage value of Arduino the encoder starts measuring the values after the 2 sec delay using the function of present time. It is also important to note that digital inputs were used to read the values from the encoder which give high (1**)** and low (0**)** for the signals.

3.4.5. *Angular Velocity:*

The angular velocity is measured for the motor as it sends the feedback for the motor keeping the rpm constant. The rpm operates on the same procedure measuring the total number of revolutions for a certain period with the tooth which is usually separately attached to the encoder plate which passes through the third photo interrupter which is not in line with the other two photo interrupters. Revolutions per minute of the shaft are then calculated using the following formula in the logic

$$RPM = \frac{no.of\ revolutions}{time\ of\ measurement} \times 60 \tag{28}$$

In this scenario as well, the logic for not adding up the counter while tooth remains at the same place is added for accurate reading. In present case the RPM was measured to be around 307 RPM.

3.4.6. *Piezo encoder interfacing:*

The purpose of the wheel balancing machine is to locate the exact point where the imbalance is located and add that specific amount of mass at that position to remove that imbalance. For this reason, the piezo electric sensor and the encoder need to be interfaced. The piezoelectric sensor as usual is connected to the analog input while the encoder to the digital input pins. Here the encoder have a total of 3 pins corresponding to the 3 photo interrupters, where one interrupter is used for the increment pin which gets high after every single revolution, whereas the other to interrupters are used for positioning and gets high when the encoder teeth comes in front of it. The interfacing of the two sensors were done using the phenomena of interrupts in Arduino. Here the pins of encoder are connected into the digital input pins containing the interrupts. When this inputs gets high, Arduino leaves all what it is doing and excecutes the corresponding function allotted to that interrupt. This acted as a very useful method and also the last resort because in simple programming the Arduino was not able to process the data in a single sequence resulting in a lag and incorrect readings. Interrupts prevents this lag by prioritizing the important activities. So the logic of the program was that when the interrupt of the increment pin gets high a counter is initiated and when the position pin gets high, it reads the values of piezo from the analog input and saves it in array, as directed by the corresponding function. In a single run, two such cycles are repeated in which two different arrays save the value corresponding to every position pin's high, meaning 32 values. After this, each values of the array is compared giving us the maximum and the minimum values and the index of the array giving the corresponding angle.

3.5. Solid Model:

The solid model of the machine was designed using the dimensions of commercially available wheel balancing machine. In order to closely study the structure of the commercially available wheel balancing machine, we bought a second-hand wheel balancing machine from Karachi. As part of reverse engineering, critical analysis of the machine was done. The solid structure of our machine consists of two main parts;

3.5.1. Shaft:

The design of the shaft was picked up from the machine we are studying. The dimensions of the shaft were also kept same whereas the material of the shaft was 1023 carbon steel (Appendix 1). The analysis of shaft included the torsional analysis which includes maximum shear stress and strain, von-misses stress, total deformation and the factor of safety. The torsion force of 2.6 Nm. was applied on the inner side of the shaft which is attached to the pulley whereas the outer end of the shaft is held constant. Another analysis of shaft included bending moment analysis which shows us the nodal stress, nodal strain and the factor of safety of the shaft. In this case the inner part of the shaft was considered fixed whereas the outer part of the shaft where the wheel is mounted was analyzed by applying the force equal to the weight of the wheel. The final analysis of the shaft included that of Natural Frequency. The natural frequency is the frequency at which the structure vibrates under static conditions. The significance of this is in the application of rotating machineries where if the natural frequency becomes equal to the rotating frequency of the machine, resonance occurs. The phenomena of resonance signify that the particles of structure vibrate at infinite amplitude due to which the structure starts to damage, eventually leading to structural failure.

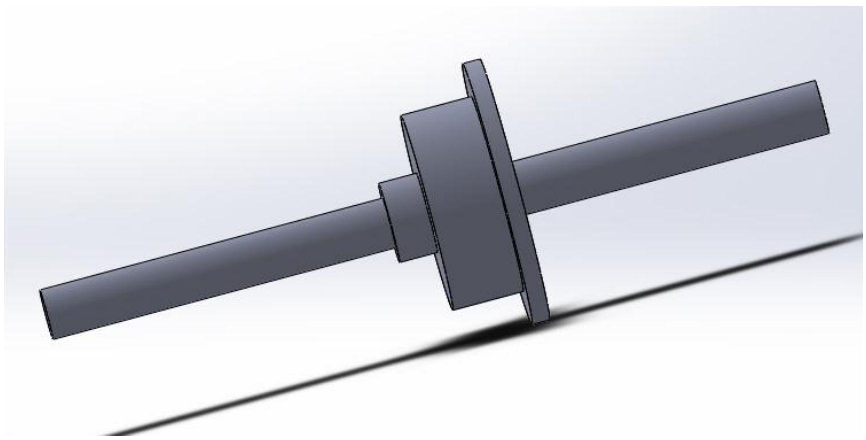

Figure 17: Shaft Model

3.5.2. Body

The design and dimensions of the outer body was also taken from the machine we are studying with slight changes in the inner structure due to the changes in dimensions of the motor. The outer part of the body consists of 2 mm thick Mild-Steel (Appendix 1) sheet which is bended and welded from 2 ends. The inner structure of the body consists of beams which is also made out of Mild-Steel material but with thickness of 3mm and then is welded to the inner side of the body. The base of the body also consists of Mild-Steel material with thickness of 4 mm and width of 64.3 cm. the purpose of this is to evenly distribute the weight of the machine which is around 90 kg after complete fabrication and assembly. One of the analysis carried on the body of the model included the static displacement which shows the displacements on the various points on the machine in a static position. In this state, the forces acting on the machine are the weight of the motor, the weight of the structure itself and the weight of the wheel which is mounted in a steady state. The weight of the wheel is taken to be 15 Kg, which is the maximum possible weight of the wheel. The second analysis was of Von-Mises Stress which shows the principal stresses acting on the body. The significance of this stress is that it shows us the possibility of structural failure which occurs when the maximum principal stress is less than the yield strength of the material. The third type of analysis was of the reaction force which was done to ensure that the total weight of the machine is evenly distributed at the bottom surface. The last analysis done was that of natural frequency of the machine which would then be compared to the rotating frequency of the motor.

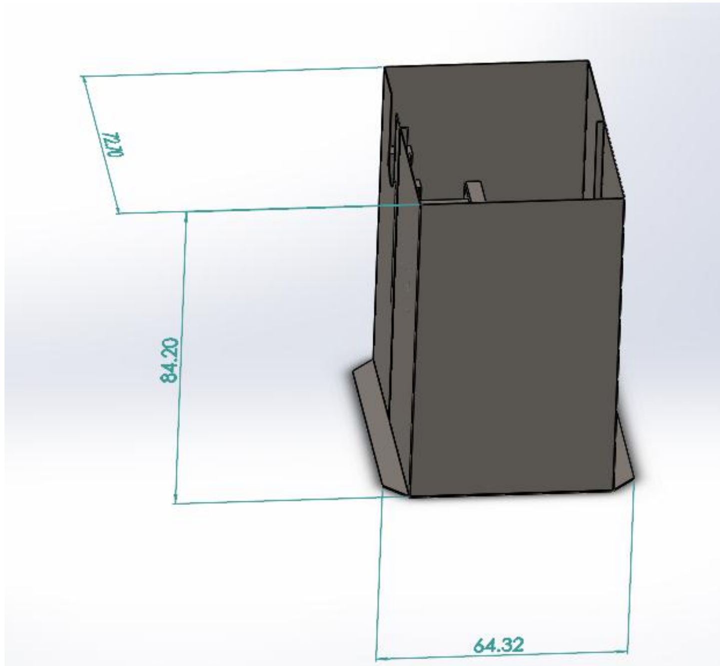

Figure 18: Outer Body Model

CHAPTER 4

RESULTS AND DISCUSSIONS

The outcome of this project is to design and manufacture a cost-effective wheel balancing machine with similar accuracy as that of a commercially available one. In order to make things easier for us, we bought a second-hand wheel balancing machine from a shop in Karachi and did critical reverse engineering on that machine. Doing this enables us to manufacture a machine by setting a lower benchmark equal to that machine so that our final product is better, or at least similar to the quality and performance of that machine. In order to achieve this outcome, we divided the assessment of the outcomes into 4 basic sections as follows, each adding significantly to the successful run of the machine.

4.1. Drive Mechanism:

The drive mechanism of the machine consists of the motor, variable frequency drive, pulley, belt, and the shaft. As the motor we chose was different from that of the machine we are studying, the overall drive mechanism design had to be re-designed. While doing so, we kept the life of the pulley-belts system to be 20 years, which is greater than the average life of a new machine before it breaks down. Furthermore, the motor will be running on a lower power than its maximum capacity, increasing the life of the motor as well. In addition to this, the use of VFD would ensure that the motor doesn't get overloaded at any point. As breakdown of motor is one of the major problems faced in the wheel balancing machines, use of such a quality motor running below its maximum capacity, with a VFD would ensure that no such problem occurs in our machine. The belt used in the current machine is a seamed belt. However, in our machine, we will be using a seamless belt. This would not only increase the life-expectancy of the belt, but would also stop the vibration caused by the seamed belt. In addition to this, the pulley used by us is a statically balanced pulley. This, combined with the seamless pulley, would make sure that any vibration on the shaft is due to the imbalanced wheel. Coming on to the shaft, the design of the shaft was picked up from the current machine we are studying and the analysis was conducted on it.

The types of analysis done on the shafts and their results are listed below:

4.1.1. Torsional Analysis of the Shaft:

As far as the results are concerned, the maximum shear stress analysis of the shaft shows that maximum shear stress is applied on the inner half of the shaft due to the applied torque. Followed by this the maximum torque is applied on the outer half of the shaft. The center part of the shaft which has the greatest area faces the least shear stress. Similar pattern is shown for the maximum shear strain of the shaft. The total deformation shows that the maximum deformation occurs at the central part of the shaft. The maximum shear stress of the shaft is less than the yield strength of the shaft.

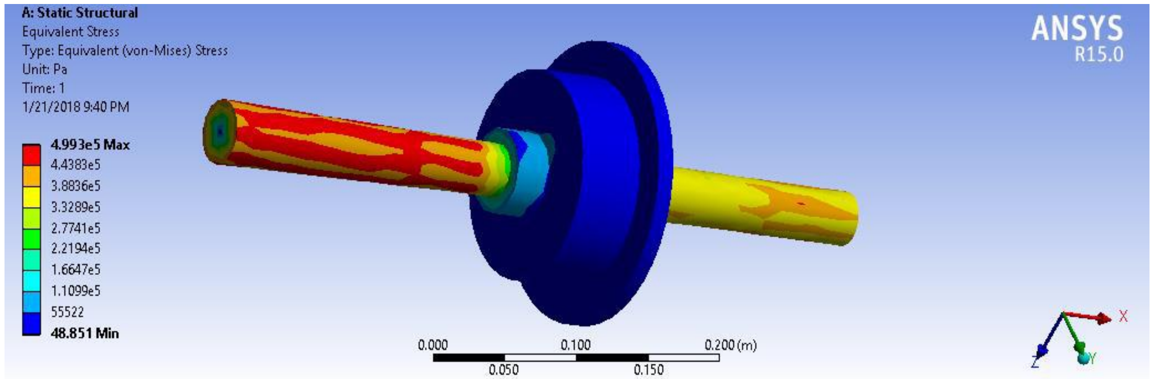

Figure 18: Torsional Analysis-Von Mises Stress

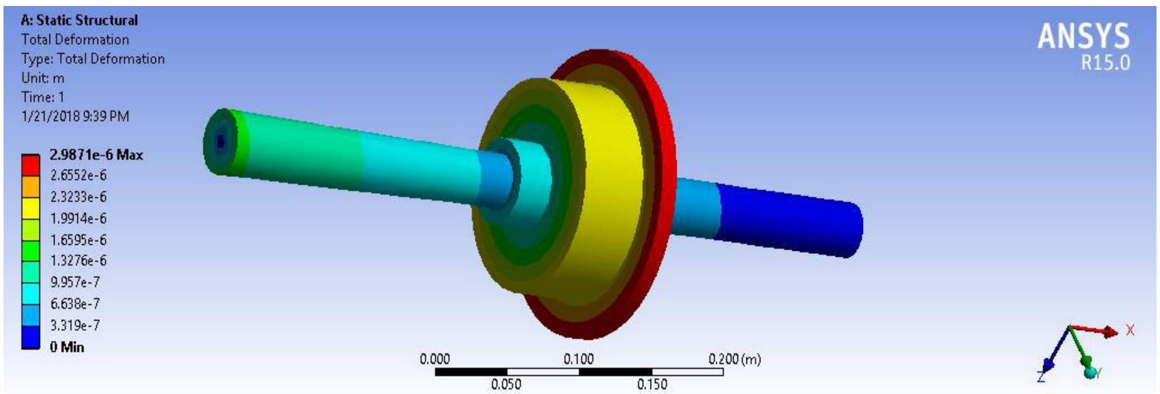

Figure 19: Torsional Analysis-Total Deformation

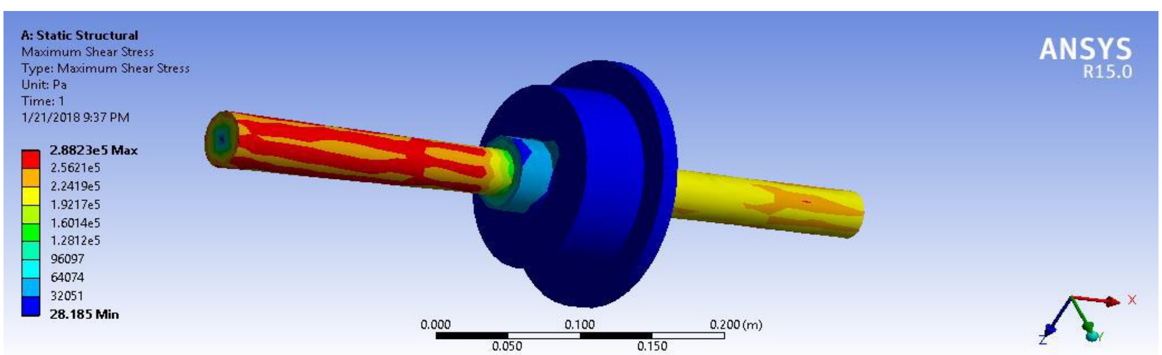

Figure 20: Torsional Analysis-Maximum Shear Stress

4.1.2. Bending Moment and Shear Force Diagram:

The results of the bending moment analysis show us the nodal stress, nodal strain and the factor of safety of the shaft. The results show that the maximum bending stress is at the tip of the inner side of the shaft as that part experiences the maximum bending moment due to greatest moment arm. The bending displacement is maximum at the outer end of the shaft here the load is applied. The actor of safety of the shaft varies from 6 at the inner side where the bending moment is the maximum, to 150, which is at the outer edge due to least bending moment. As the maximum von-misses stress is less than the yield stress of the shaft, this signifies that the material is within safe limits. This result evaluated through the analysis satisfies the theoretical calculations shown above.

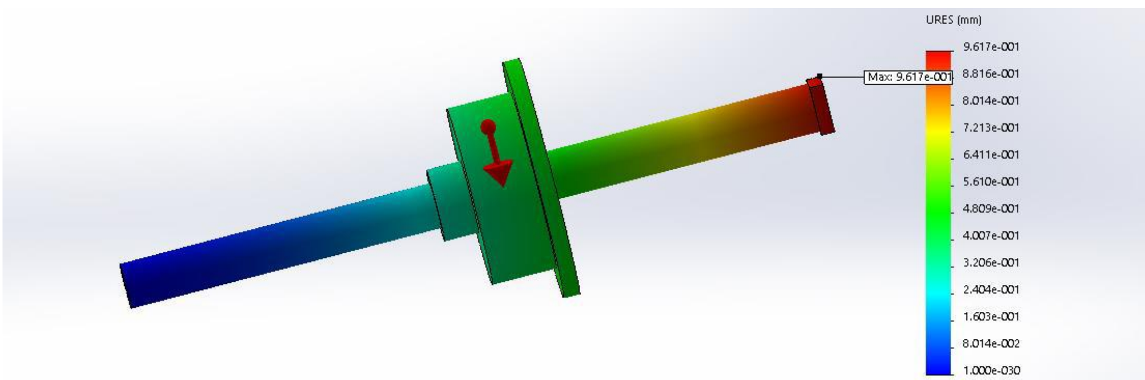

Figure 21: Bending Analysis-Total Deformation

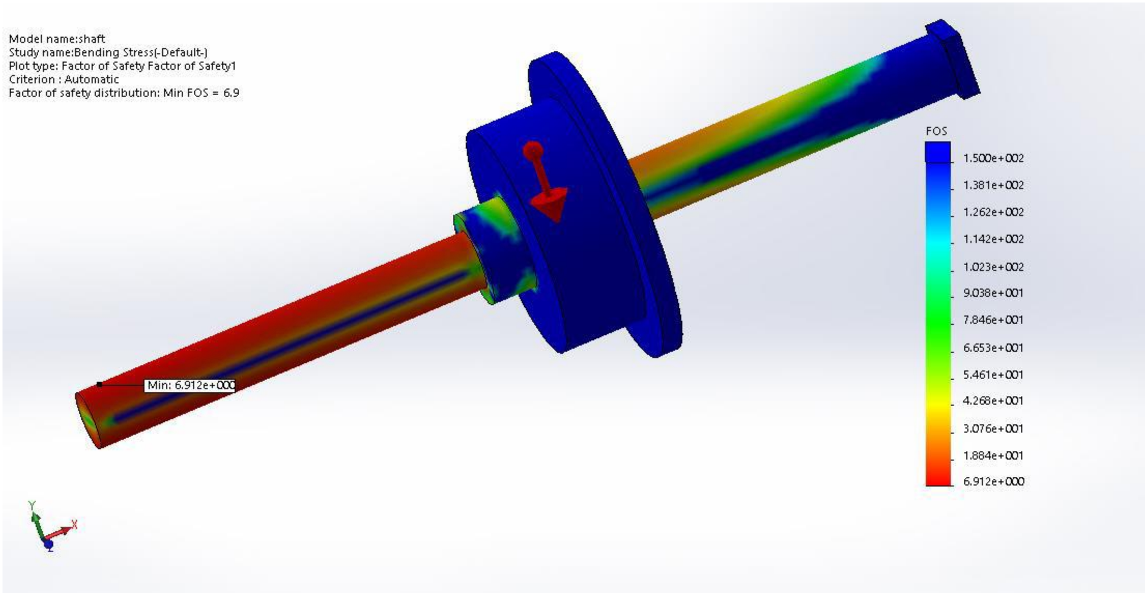

Figure 22: Bending Analysis-Factor of Safety

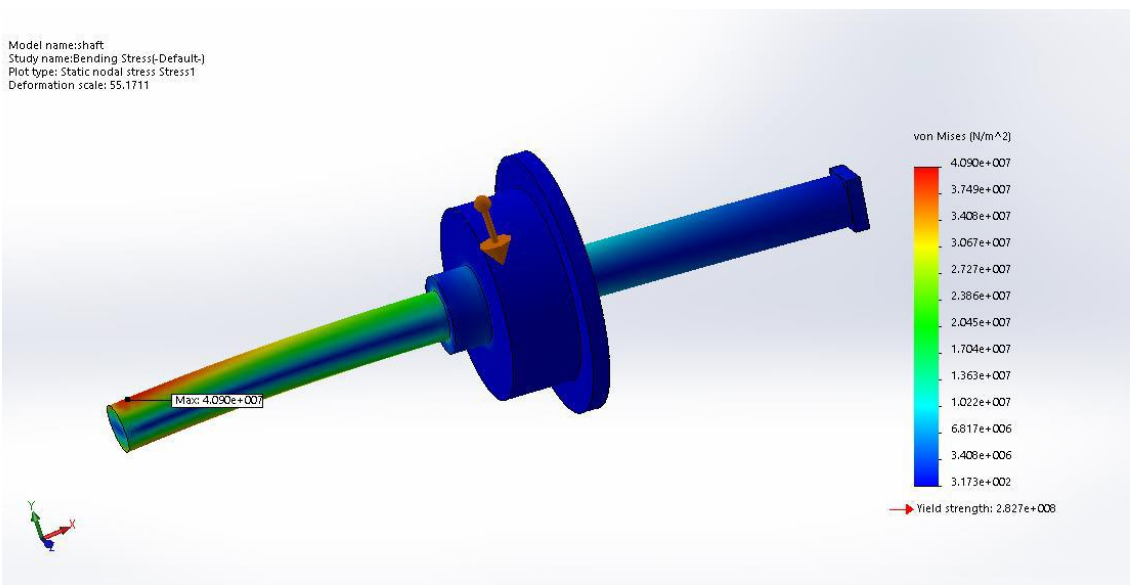

Figure 23: Bending Analysis-Von Mises Stress

4.1.3. Natural Frequency:

The natural frequency analysis of the shaft shows us that the natural frequency is equal to 55.103 Hz. whereas on the other hand the frequency of the shaft when rotating at 208 RPM is equal to 3.47 Hz. This significant difference between the two frequencies shows us that the frequency ratio (r) is almost 0.06 which ensures minimum vibration of the shaft when it us rotating.

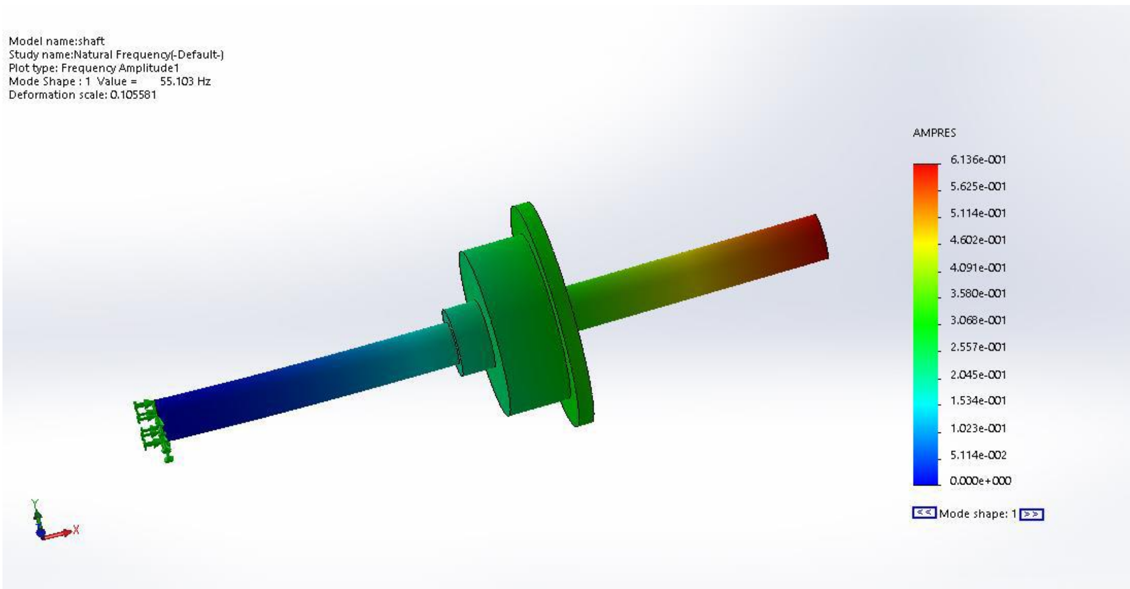

Figure 24: Natural Frequency

4.2. Sensors Instrumentation:

As mentioned in the early part of the report, the two main sensors that we will be using are piezo-electric and optical encoder. In order to improve that quality of piezo's signal, we used the Butterworth digital filtration of lowpass filter of 4 Hz. This reduced all the noises created by the electrical and other signals. The commercially available wheel balancing machine use Data Acquisition card for reading and processing the sensors' values. We, however, are using Arduino Mega board microprocessor which has significantly reduced our cost. The interfacing procedure of the piezo sensors and optical encoder have been mentioned in the methodology section of the report.

4.3. Calibration:

The calibration of the piezo sensors were next important challenge after interfacing them with the encoder. The procedure we followed was that we first balanced the tire from a wheel balancer shop. Than we mounted the tire and took the reading of the peak-to-peak value and the corresponding position of it. This gave us the net imbalance in the shaft and hub of the machine. Another important aspect was to calculate the lag in the readings which is fixed. For this, we added a mass on different positions of the tire with respect to the zero position of the machine and figured out the imbalance point using the graph of the raw data. The difference of theoretical and practical positions of imbalance remained constant which was able to give us a lag in terms of this angle. We later on incorporated this lag and the imbalance of the drive mechanism in our final formulas which gives us the final mass and its imbalance position. After this we added known masses at fixed radius and took the raw peak-to-peak values of the system. The mass was increased with the increment of 5g, starting from 5g and going till 50g. After this the corresponding graph of centripetal force and mass was plotted on excel to calculate the formula linking the two variables.

Following is the graph and the line of best fit.

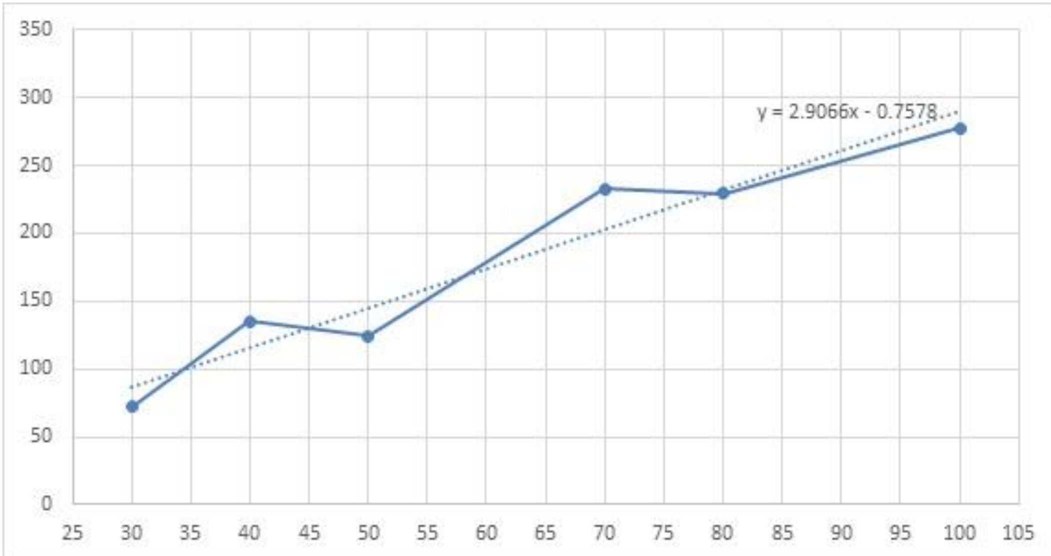

Figure 25: Calibration graph

4.4. User Interface:

For the user interface of the machine a 3x4 keypad and 4x20 green LCD display was used. Apart from this a relay was used which turned the motor on when a button on the keypad is pressed.

Figure 26: Display Components

4.5. Solid structure

As our solid structure was different than that of the machine we are studying, we had to redesign the model and carry out its structural analysis on SolidWorks. The types of analysis carried out are shown and explained below.

4.5.1. Static Displacement

The static displacement shows the displacements on the various points on the machine in a static position. The results of the static displacement analysis on SolidWorks shows us that the maximum displacement occurs on the surface on which the motor is mounted. This is because of the heavy weight of the motor itself. Another point of static displacement is just above the mounting place of the shaft. This displacement is caused by the weight of the wheel which is transferred to the structure via shaft. The overall maximum static displacement on the structure is 0.01 mm.

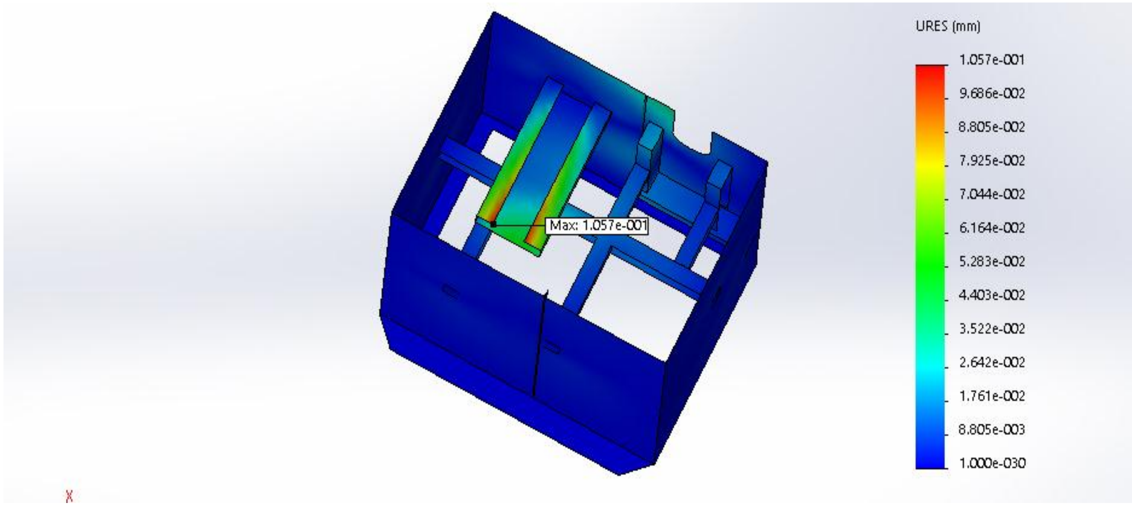

Figure 27: Static Displacement (Total Deformation)

4.5.2. Von-Mises Stress

The analysis for Von-Mises stress, which shows the principal stresses in the structure, is shown below. The areas of high stresses include the surface of which the motor and shaft is mounted and the beam which is supporting the two surfaces. The maximum stress is found to be on the surface on which the shaft is mounted.

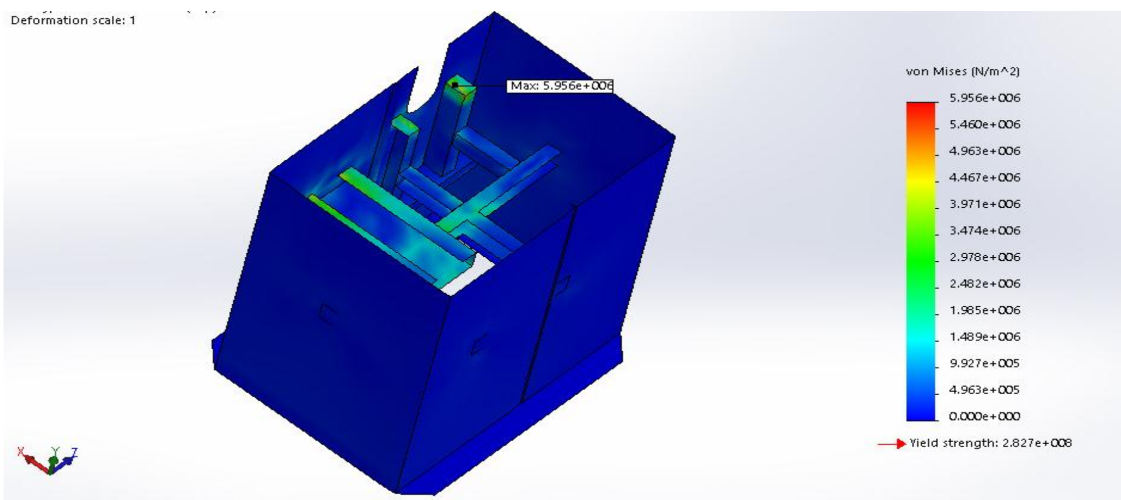

Figure 28 : Von-Mises Stresses

4.5.3. Reaction Force

The reaction force analysis was carried out to ensure that the reaction forces are distributed on the base section of the machine. The results show that the width of the base structure is enough to distribute the weight of the machine which is nearly 100 Kg. The results show that major part of reaction forces is experienced on the right side of the machine, as on this side the wheel and motor are placed. The maximum reaction force is at the corner which is almost 40 N. This force is small enough to neglect the effect of bending due to it.

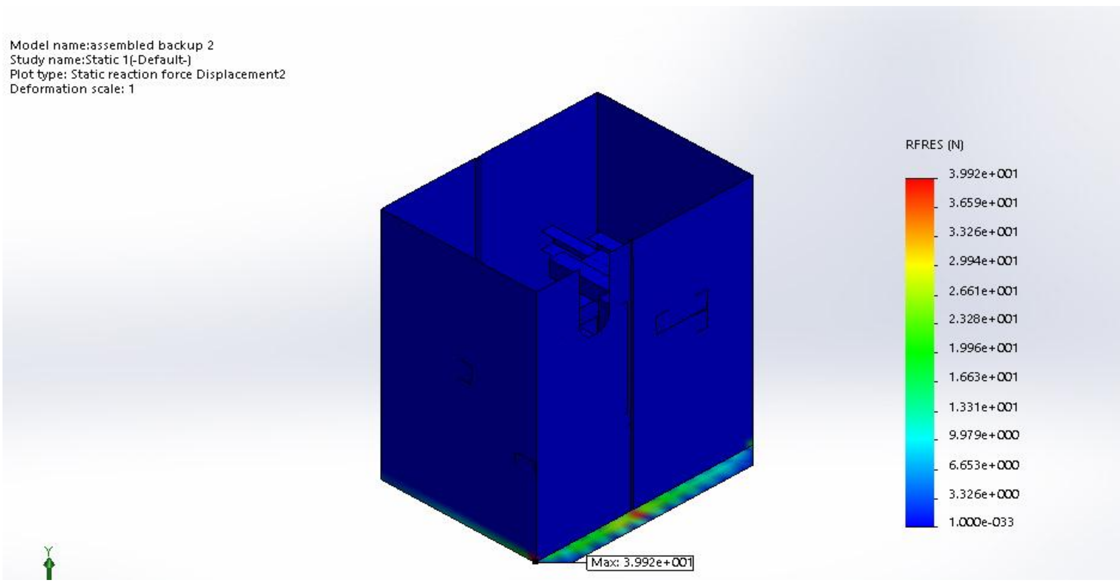

Figure 28: Reaction Force

4.5.4. Natural Frequency

The natural frequency analysis results show 5 different frequencies, each at different modes. The natural frequencies ranges from 36.96 Hz to 69 Hz, whereas the frequency of the machine due to the running of 1400 RPM motor is 23.3 Hz. This shows that the maximum frequency ratio is 0.67 as compared to 1, where resonance occurs. The result of natural frequency analysis and its position of maximum amplitude is shown in the figures below.

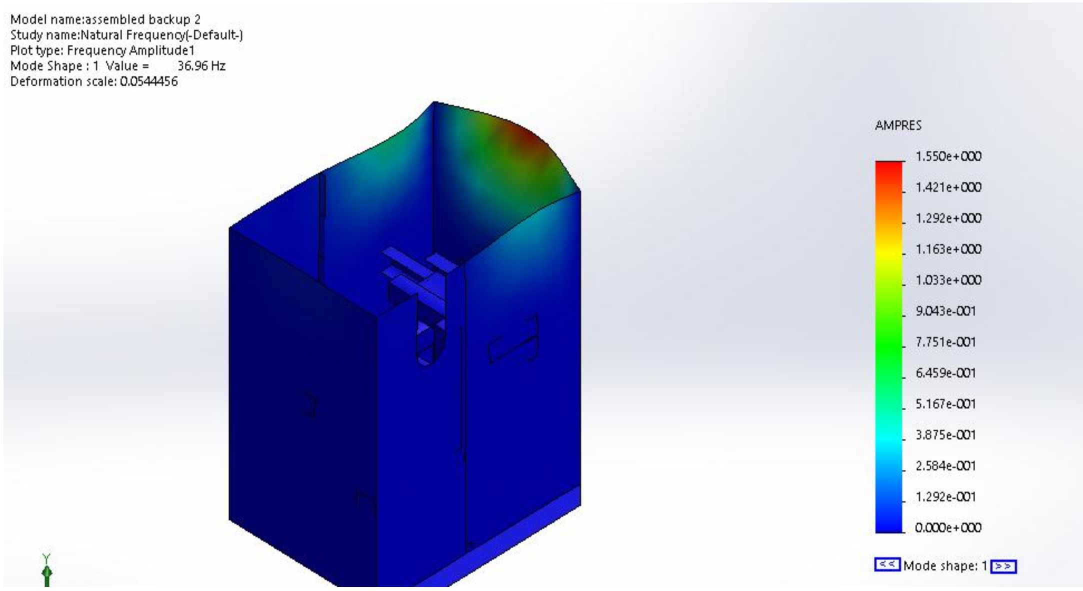

Figure 29: Natural Frequency-Mode 1

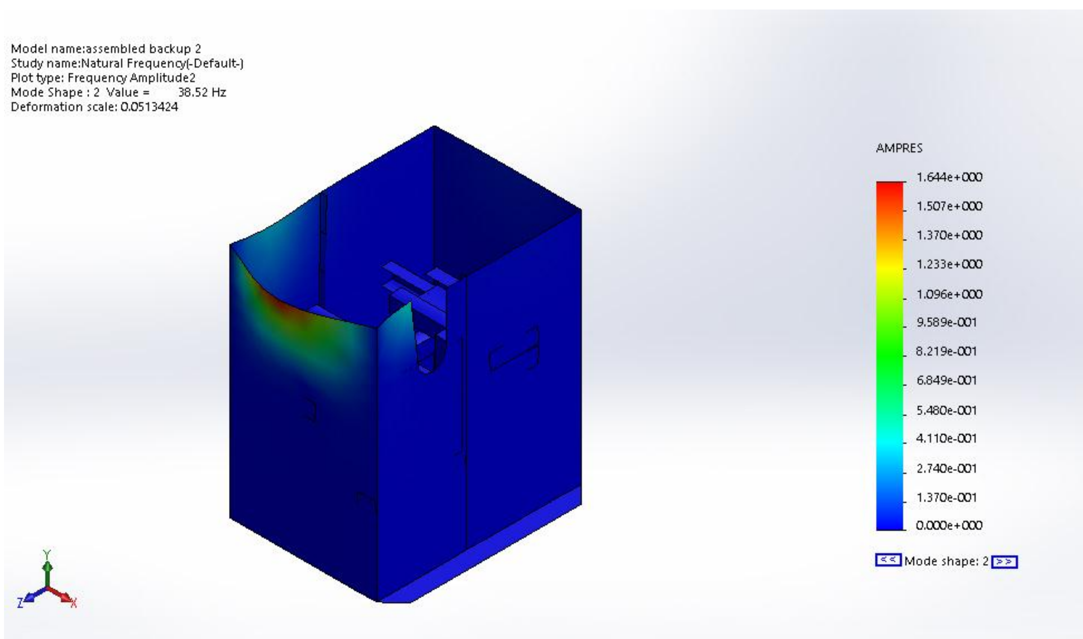

Figure 29: Natural Frequency-Mode 2

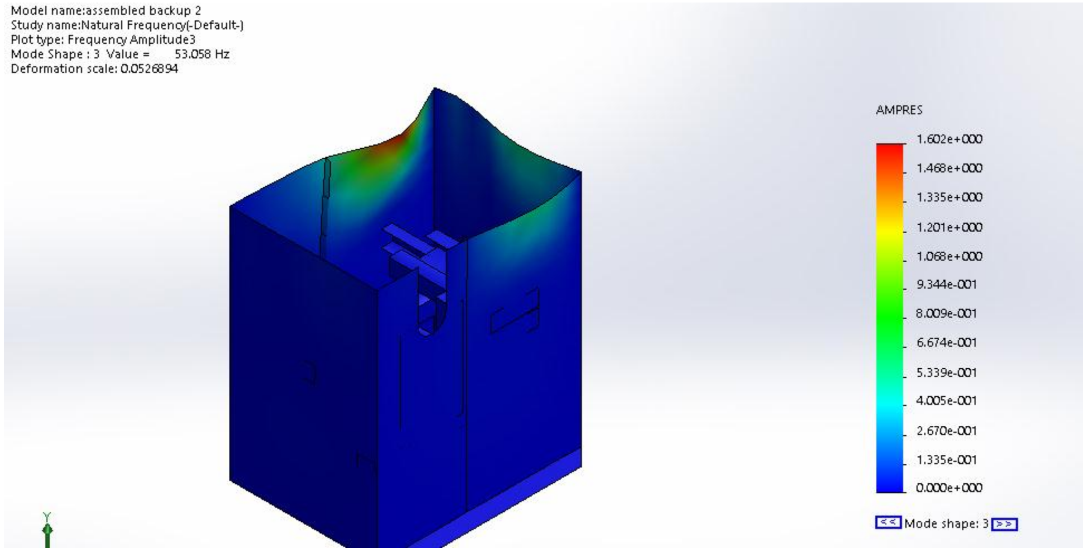

Figure 30: Natural Frequency-Mode 3

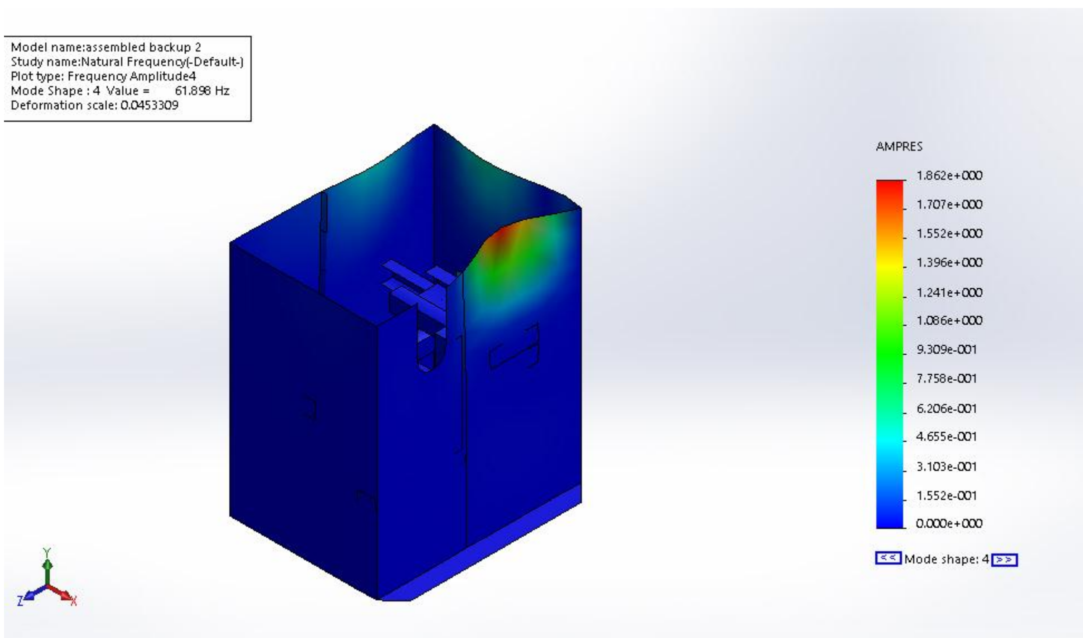

Figure 31: Natural Frequency-Mode 4

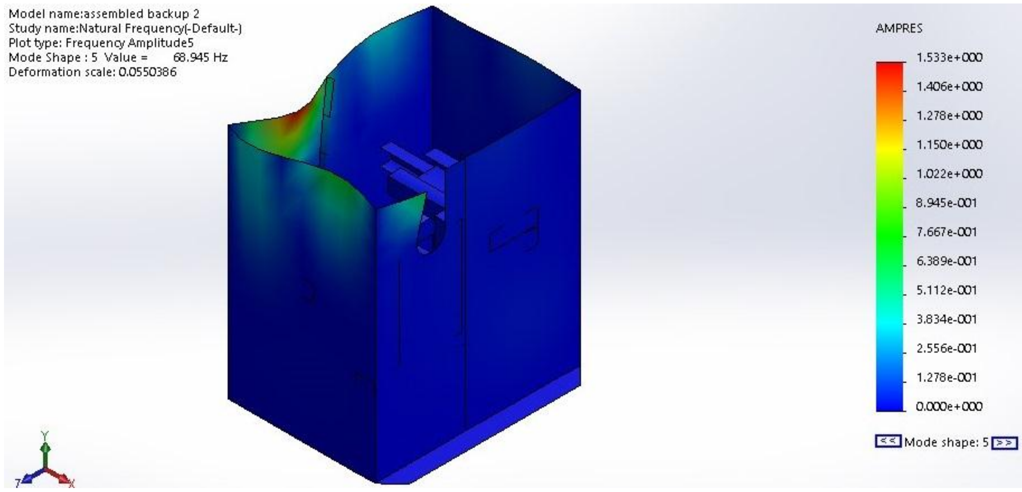

Figure 32: Natural Frequency-Mode 5

4.5.5. Factor of Safety (FOS)

The FOS analysis shows that it ranges from a minimum of 47 to a maximum of 800. The minimum FOS is on the surface on which the shaft is mounted, while the maximum is on the outer body structure. The FOS of the inner beams are relatively lower because of the fluctuating forces due to the motor and the rotating shaft.

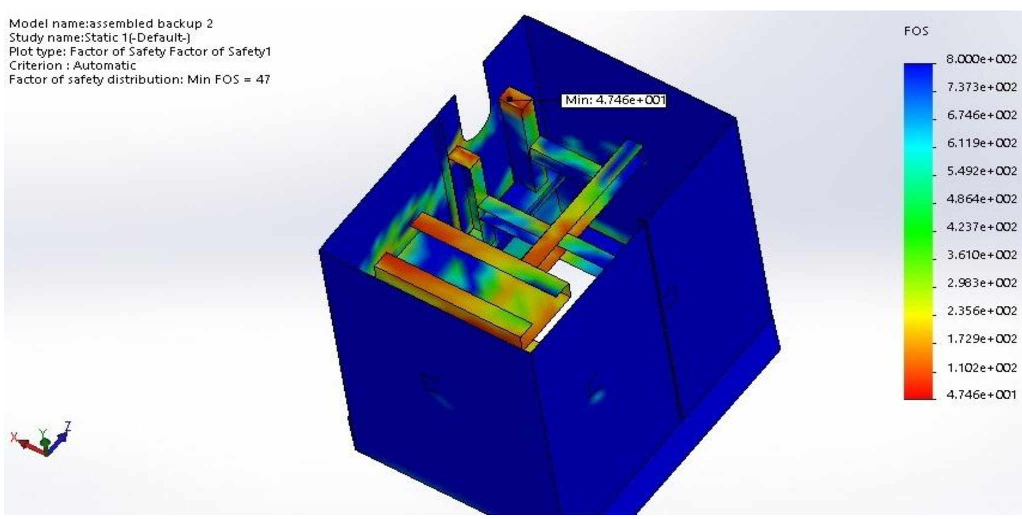

Figure 33: Factor of Safety

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1. Conclusion:

In light of the above-mentioned results, the following can be concluded about each part of the machine.

5.1.2. Drive Mechanism:

The drive mechanism designed by us, which includes motor, Variable Frequency Drive, pulley, belt and shaft shows us the complete reliability of the machine. The belts proposed by us will have the life expectancy of 20 years. On the other hand, the results of stress analysis on shaft shows us that the Von-Mises stress, which shows us the maximum principal stress is less than the yield stress of the material of the shaft. This shows that under any circumstances the shaft will not experience stress failure. Furthermore, the total deformation in both the cases of torsional analysis and bending moment analysis is less than 1 mm, which is very minimal clearly showing us the integrity of the shaft. In addition to this, the Factor of Safety results convey that with minimum FOS being equal to 7, the shaft design is good enough to sustain unexpectedly large forces. Lastly, the small frequency ration of natural frequency and the frequency of the rotating shaft suggests us that the amplitude of vibration in a rotating shaft will be small enough to cause any problems, both to the results and the integrity of the structure.

5.1.3. Sensor Instrumentation:

The combination of amplifier, RC filtration low pass and digital low pass filtration gave us a raw peak-to-peak value which is than converted to the mass and corresponding angle. The final accuracy of the machine was 7g and the angle accuracy was 10 degrees. This combination is minimal due to two main reasons. Firstly as far as the accuracy was mass is concerned, the force generated by such mass is nigiligible. Furthermore another important factor for the smooth running of the car is wheel alignment hence the combination of both only results in a smooth drive without wobbling. As far as the angle is concerned, the 10 degrees difference is minute because the mass components used to balance the tire is has enough width that it covers this much angle. The example of this is given in the following figure.

Figure 34: Mass Component

5.1.4. Solid Structure:

The results of the analysis carried out on the solid structure is also acceptable. The maximum displacement analysis is satisfactory as the maximum displacement is around 0.1mm, which is not significant at all. This ensures that the structure is strong enough when it is in a static condition under application of all possible loads. The Von-Mises stress analysis shows that the maximum principal stress is less than the yield strength of the material by the factor of 100. This ensures that the structure will not experience structural failure. As far as the analysis of reaction force is concerned, it shows that the maximum force of 40 N being applied on the surface satisfies the width of the base sheet which is taken by us. The width is enough to evenly distribute the total weight of around 1000 N of the machine. The results of the natural frequency of the structure shows that the maximum frequency ratio is equal to 0.67. This shows that under any circumstances, resonance will not occur which ensures that the structural integrity of the machine is good enough to sustain the damage caused by the rotating objects. Lastly, and most importantly, the Factor of Safety analysis of any machine is very important. Studies have shown that FOS should be greater than 10 in such machines to account for any unexpected force. Our results completely satisfy this result which shows that the minimum FOS is 47. Whereas the FOS on the outer body, which is most likely to experience an unexpected impact, is 800. These results suggest that the solid structure is capable of withstanding any impact and is least likely to experience structural failure.

5.2. Future Recommendations:

During the course of instrumentation in this project, significant problems were faced due to the limitations of the Arduino when it came to reading and processing data at high rate. Furthermore the inability of Arduino to define an array of undefined size also created huge problems when making code to save the sensor values. Even though the use of interrupts did solved the problem but that came at the cost of intrinsic programming and compromise on the results. This is because when the code was further modified to enhance the results, the Arduino again faced lag, limiting out accuracy. In recommendation to these problems, a Programmable Logic Controller (PLC) can be used as instructed to us by the evaluator in open-house. Even though PLC is a bit expensive than Arduino, it is more suitable for industrial usage with higher processing capability and easy programming.

In addition to this, the accuracy of the position of imbalance can also be increased by using the encoder plate of 64 teeth rather than 32 teeth. This would be able to improve the position accuracy to 1.4 degrees. But then again, this will only be viable if it is used with PLC rather than Arduino or else Arduino will become the limiting factor in improvement of the results.

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APPENDIX I: PROPERTY TABLES

Table 2: 1023 Carbon Steel

Mechanical Properties	Metric	Imperial
Tensile strength	425 MPa	61600 psi
Yield strength	360 MPa	52200 psi
Shear modulus	80.0 GPa	11600 ksi
Bulk modulus	140 GPa	20300 ksi
Elastic modulus	190-210 GPa	27557-30458 ksi
Poisson’s ratio	0.27-0.30	0.27-0.30
Elongation at break	15%	15%
Reduction of area	40%	40%
Hardness Brinell	121	121
Hardness, Knoop	140	140
Hardness, Rockwell B	68	68
Hardness, Vickers	126	126
Machinability	65	65

Table 3: Mild Steel

Mechanical Properties	Metric	Imperial
Hardness, Brinell	126	126
Hardness, Knoop	145	145
Hardness, Rockwell B	71	71
Hardness, Vickers	131	131
Tensile Strength, Ultimate	440 MPa	63800 psi
Tensile Strength, Yield	370 MPa	53700 psi
Elongation at Break (In 50 mm)	15.0 %	15.0 %
Reduction of Area	40.0 %	40.0 %
Modulus of Elasticity	205 GPa	29700 ksi
Bulk Modulus (Typical for steel)	140 GPa	20300 ksi
Poisson’s Ratio (Typical for Steel)	0.290	0.290
Machinability	70 %	70 %
Shear Modulus (Typical for steel)	80.0 GPa	11600 ksi