VISVESVARAYA TECHNOLOGICAL UNIVERSITY
"JnanaSangama", Belgavi-590018

A Project Report On

"DESIGN, FABRICATION AND EVALUATION OF STATIC AND DYNAMIC BALANCING APPARATUS"

Submitted in partial fulfillment of the requirements for the degree of

BACHELOR OF ENGINEERING
IN
"MECHANICAL ENGINEERING"

Submitted by

DEEPU D K	4BW14ME013
SACHIN	4BW14ME041
SUJATHA K M	4BW14ME048
MANJUNATH L S	4BW15ME408

Under the guidance of
Dr. HEMARAJU
Associate Professor

Department of Mechanical Engineering
B.G.S Institute of Technology
B G Nagar - 571448
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DEPARTMENT OF MECHANICAL ENGINEERING

CERTIFICATE

Certified that the project work entitled **"DESIGN, FABRICATION AND EVALUATION OF STATIC AND DYNAMIC BALANCING APPARATUS"** carried out by **Deepu DK** (4BW14ME013), **Sachin** (4BW14ME041), **Sujatha KM** (4BW14ME048) and **Manjunath LS** (4BW15ME408), are the bonafide students of BGS Institute of Technology, B G Nagar in partial fulfillment for the award of Bachelor of Mechanical Engineering Visvesvaraya Technological University, Belgaum during the year 2017-18. It is certified that all corrections/suggestions indicated for Internal Assessment have been incorporated in the Report deposited in the departmental library. The project report has been approved as it satisfies the academic requirements in respect of project work prescribed for the said Degree.

Signature of the Internal Guide	Signature of the HOD	Signature of the Principal
Dr.Hemaraju Associate Professor	**Prof. M. Rajaram** HOD	**Dr. B K Narendra** Principal

EXTERNAL VIVA

Name of the Examiners	Signature of Examiners
1. M. RAJA RAM	1.
2. Dr. Gomatesh M Ravanavar	2.

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DEEPU DK
SACHIN
SUJATHA K M
MANJUNATH L S

DECLARATION

We, Deepu DK, Sachin, Sujatha KM, Manjunath LS, hereby declare that this project work entitled **"DESIGN, FABRICATION AND EVALUATION OF STATIC AND DYNAMIC BALANCING APPARATUS"** was independently carried out by us under the guidance and supervision of Hemaraju, Asst. Prof. B.G.S Institute of Technology, B G Nagar. This project work is submitted to Visvesvaraya Technological University in partial fulfillment of the requirement for the award of degree of Bachelor of Mechanical engineering during the academic year 2017-2018.

DATE:
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ABSTRACT

The time of the study is to design, fabricate and make a performance studies on balancing of rotating parts machine apparatus. The present work focus on the design modification of the existing equipment. The complete design as well as fabrication concept are presented and discussed in detail. The details of the experimental procedure as well as theoretical method and graphical method find the unbalanced masses are presented in details. The result indicates that there is a good agreement between present experimental results with that of from the theoretical approach showing perfect balance condition.

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CHAPTER 1

INTRODUCTION

Production tolerances used in the manufacture of rotors are adjusted as closely as possible without running up the cost of manufacturing prohibitively. In general, it is more economical to produce parts, which are not quite true, and then to subject them to a balancing produce than to produce such perfect parts that no correction is needed. Typical example of such machinery is crankshafts, electric armatures, turbo-machinery, printing rollers, centrifuges, flywheels, and gear wheels. Some common cause of irregularity during production are machining error, cumulative assembly tolerance, distortion due to heat treatment, blow holes or inclusion in casting , and materials non homogeneity. Because of these irregularities are actual axis of rotation does not coincide with the one of the principle axis of inertia of the body, and variable disturbing forces are produced which results in vibrations. In order to remove these vibrations and establish proper operation, balancing becomes necessary. The forces generated due to an unbalance are proportional due to the rotating speed of the rotor squared. Therefore, the balancing of high speed equipment is especially important.

Frequently, the machine already in operation will need rebalancing or a new machine when assembled at its permanent location will need balancing. In some cases, the cost of disassembly, shipping to a balancing machine, and delay, are prohibitive and the machine must be rebalancing on the field in its bearings. The system of balancing discuss in this experiment was developed to satisfy the need to perform field balancing of equipment easily and accurately.

Although there are many possible cases of vibration in rotating equipment, this technique will deal only with that component of vibration, which occurs at running speed, and is caused by a mass unbalance in the rotor. This component of vibration which can be eliminated by the addition or removal of weight from the rotor.

The condition of unbalance of a rotating body may be classified as static or dynamic unbalance. In the case of static unbalance, the unbalance appears in single planes. In the case of dynamic unbalance, the unbalance can be in different axial planes. As a result, while in rotation, the two unbalanced forces form a couple, which rocks the axis of rotation and causes undesirable vibration of the rotor, mounted in its bearings.

Let us now considering a single rigid rotating mass mounted in two supporting bearings and assume that the axis of rotation in horizontal. It can be shown that for the correct balance of such a rotor, two weights placed in different radial planes of the rotor are necessary and sufficient to balance the rotor. The vibratory motion of either bearing may be represented by three components, the horizontal and vertical radial components and the axial components. The purpose of balancing at running speed of the rotor is to reduce the greatest of these three components to a practical minimum. The other two components will be reducing to negligible amounts from their original magnitudes by this technique. Assume in this example that the radial components is the greatest.

Therefore, only this component will be measured and analysed in this technique. It follows that if the vertical components of vibration of two points, one chosen on each bearing, are reduce to zero(or near zero) the purpose of balancing has been accomplished and no vibration will be transmitted to the support structure.

These are four variables to be deals with when balancing any rigid rotor. They are the amount and position of the two correction weights required to balance the rotor. Each correction weight is located in one of the arbitrary chosen radial reference planes on the rotor. These reference planes are usually placed near the support bearings. In general, the farther apart the radial reference planes are located, the smaller are required correction weight. This technique deals with these four variables simultaneously as the amount and position of the correction weight in the other reference plane.

The data necessary to determine the magnitudes and position (angle) of the two correction weights are obtained by test runs, all at the same speed by measuring the vibration amplitude and phase angle at each bearing. Some commercial equipment allows measurement of the vibration amplitude and phase relative to a geometric trigger reference point on the rotor. Lacking the instrumentation to measure the phase angle, this technique will obtain data to allow calculation of the phase angle.

The high speed of engines and other machines is a common phenomenon now-a-days. It is therefore, very essential that all the rotating and reciprocating parts should be completely balanced as for as possible. If these parts are not properly balanced, the dynamic forces are set up. These forces not only increase the loads on bearings and stresses in the various members, but also produce unpleasant and even dangerous vibrations. Here we shall see the balancing of unbalanced forces caused by rotating masses.

If the mass centre of a component of mass 'm' is rotating at an angular velocity 'ω' at a distance 'r' from the axis of rotation, then the component is subjected to force of $m*r*\omega^2$. The out of balance forces increases the bearing loads and introduce stresses in the rotor and frame work of a machine. These so called 'inertia forces' may introduce dangerous vibration, structural failure or unacceptable noise and may limit the operating speed of a machine. The magnitude of these forces may be reduced or eliminated in the design stage by balancing the effects of the various mass elements of the device. Additionally, extra balance masses may deliberately added to a rotating system in order to cancel out the residual design imbalance. This experiment involves balancing a number of known out of balance masses on a shaft.

The apparatus is constructed so that unbalanced rotating masses may be set up in 5 parallel planes. A variable speed electric motor is used to drive the shaft. Out of balance is indicated by excessive vibration when the frequency of rotation passes through one of the frequencies of the suspension system. The effectiveness of the balance is then demonstrated by rotating the balanced mass system in a suspended framework.

This application note will demonstrate with the aid of several worked examples, how easy it is to balance rotating machinery. Straight forward methods will be presented that make use of simple portable balancing instrumentation to measure on rotating parts running at normal operating speeds. The balancing machine accepts balancing masses at different positions at any suitable radius and show whether the system is completely balanced or not.

1.2 Concept of Balancing

Balancing is the technique of correcting or eliminating unwanted inertia forces and moments in rotating or reciprocating masses. The balancing of the system may be in static or dynamic balancing. The objectives of balancing an engine are to ensure.

1. That the centre of gravity of the system remains stationary during a complete revolution of the crankshaft and
2. That the couples involved in acceleration of different moving parts balance each other

1.3 Types of Balancing

1.3.1 Primary Balancing

Primary Balancing describes the process where primary forces cause by unbalanced mass components in a rotating object may be resolved into one plane and balanced by adding a mass in that plane only. As the object would now completely balanced in the static condition (but not necessarily in dynamic) this is often known as static balancing. A router is set to be statically balanced if the vectors sum of centrifugal force is zero.

1.3.2 Secondary Balancing

Secondary Balancing describes the process where primary forces and secondary force couples caused by unbalanced mass components in a rotating object may be resolved into two or more planes and balanced by adding mass increments in those planes. This balancing process is often known as dynamic balancing because the unbalanced only becomes apparent when the object is rotating. Dynamic balance is a balance due to the action of inertia forces. After being balanced dynamically, the object would be completely balanced in both static and dynamic condition. However, it is not true that if a system is statically balanced, it is also dynamically balanced. A system of rotating mass is in dynamic balanced when there does not exist any resultant centrifugal force as well as couple.

1.4 Causes of Rotor Unbalance

1.4.1 Manufacturing Causes

Many causes are listed as contributing to an unbalanced condition, including material problems such as density, porosity, voids and blowholes. Fabrication problems such as misshapen castings, eccentric machining and poor assembly. Distortion problems such as rotational stresses, aerodynamics and temperature changes. Even inherent rotor design criteria that cannot be avoided. Many of these occur during manufacture, others during the operational life of the machine. While some corrections for an eccentricity can be counteracted by balancing. Dynamic balancing should not be a substitute for poor machining or other compromise manufacturing practices. In the manufacture process, if proper care is taken to ensure that castings are sound and machining is concentric, then it is follows that the two axis will coincide and the assemble rotor will be in a state of balance.

1.4.2 Assembly Causes

As previously stated, there are many reasons why unbalanced occurs when a rotor is being fabricated. Principle among these is a stack up of tolerances. When a well-balanced shaft and well balanced rotor are united, the necessary assembly tolerances can permit redial displacement, which will produce an out of balanced condition. The additions of keys and keyways add to the problem. Although an ISO standard does exist for shaft and fitment key conventions, in practice different manufactures follow their own procedures. Some use a full key, some a half key and some no key at all. Thus, when a unit is assembled and the permanent key is added, unbalanced will often be the result. The modern balancing tolerances specified by ISO, ANSI and others make it imperative that the conventions listed in the ISO standard be followed. Failure to do so will mean that the low level balance tolerances specified by in these standards will be impossible to achieve.

1.4.3 Installed Machines Causes

When a rotor has been in service for some time, various other factors can contribute to the balanced condition. These include corrosion; wear, distortion and deposit build up. Deposits can also break off unevenly, which can lead to severe unbalance. This particularly applies to fans, blowers, compresses and other rotating devices handling process variables. Routing inspection and cleaning can minimize the effect, but eventually the machine will have to be removed from service for balancing. Large unbalances will of course require large weight corrections and unless care is taken, this can have a detrimental effect on the integrity of the rotor. Concentrating a weight adjustment (whether adding or taking away) at given point can weaken the rotor.

1.4.4 Other Causes

Another cause of unbalanced is not readily apparent, is the difference between types of rotors. There are two distinct types – rigid and flexible.

- If a rotor is operating within 70% - 75% of its critical speed (the speed at which resonance occurs, i.e. its natural frequency) it can be considered to be a flexible rotor.
- If it is operating below that speed it is considered rigid. A rigid rotor can be balanced at the two end planes and will stay in balance when in service. A flexible rotor will require multi-plane balancing. If a rotor is balanced on a low speed balancing machine assuming it is a rigid and then in service becomes flexible, then unbalanced and thus high vibrations will be the result.

CHAPTER 2

LITERATURE REVIEW

The primary cause of vibration in rotating machinery is mass imbalance. Which occurs in the principle axis of moment of inertia is not coincide with the axis of rotation. As we know that usually balancing procedure involving adding or subtracting correction masses in different planes. So that principle axis is re-centred and realigned. Carrying out of balancing with different planes is strong need in order to avoid vibrations, noise, which results with unpleasant stresses and breakage. Several scientists done too much work and developed many models to determine magnitude and direction of the force and couples from testing. Various models with different test procedure and methods which was adapted to measures unbalanced force and couples are studied. Among the methods some of the models are discussed below.

SEVERAL CASES OF BALANCING

2.1 Balancing of Single Rotating Mass By a Single Mass Rotating in the Same Plane [1]

Consider a shaft rotating between two bearings and shaft is perfectly machined i.e., no eccentricity. Hence the system is said to be a balanced one. Now if the mass is attached to the shaft, the system will be disturbed i.e., unbalanced. In order to balance this system attaches one more mass m_2 diagonally opposite to the mass m_1. Now consider a disturbing mass m_1 attached to a shaft rotating at ω rad/sec as shown in fig. Let r_1 be the radius of rotation of the mass m_1 (i.e. distance between the axis of rotation of the shaft and the centre of gravity of the mass m_1). We know that the centrifugal force exerted by the mass m_1 on the shaft.

$$FC_1=m_1\omega^2r_1 \text{................} \quad (i)$$

This centrifugal force acts radially outwards and thus produces bending moment on the shaft. In order to counteract the effect of this force, a balancing mass (m_2) may be attached in the same plane of rotation as that of disturbing mass (m_1) such that the centrifugal forces due to the masses are equal and opposite.

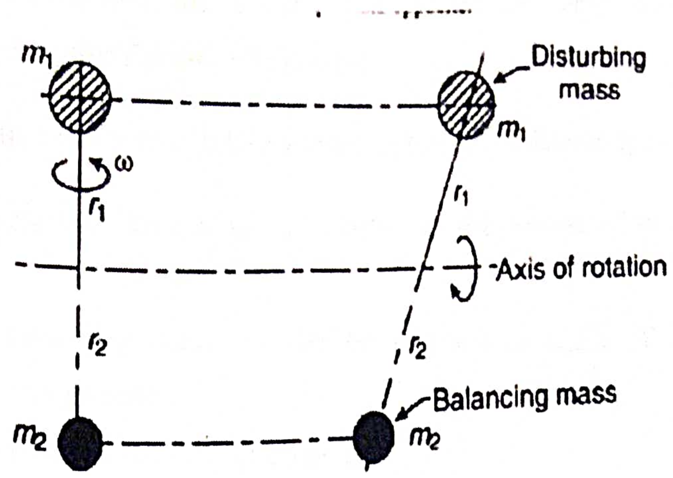


Fig 2.1: Balancing of a single rotating mass by a single mass rotating in the same plane

Let r_2= Radius of rotation of the balancing mass m_2 (i.e. distance between the axis of rotation of the shaft and the centre of gravity of mass m_2).

Centrifugal force due to mass m_2,

$$F_{C2}=m_2\omega^2 r_2 \text{............. (ii)}$$

For perfect balancing, equating equations (i) and (ii),

$$m_1\omega^2 r_1=m_2\omega^2 r_2$$

2.2 Balancing of a Single Rotating Mass By Two Masses Rotating in Different Planes [1]

We have already discussed that by introducing a single balancing mass in the same plane of rotation as that of disturbing mass, the centrifugal forces are balanced. In other words, the two forces are equal in magnitude and opposite in direction. But this type of arrangement for balancing gives rise to a couple which tends to rock the shaft in its bearings. Therefore in order to put the system in complete balance, two balancing masses are placed in different planes, parallel to the plane o rotation of the disturbing mass, in such a way that they satisfy the following two conditions of equilibrium.

1. The net dynamic force acting on the shaft is equal to zero. This requires that the line of action of three centrifugal forces must be same. In other words, the centre of masses of the system must lie on the axis of rotation. This is the condition for static balancing.

2. The net couple due to the dynamic forces acting on the shaft is equal to zero. In other words, the algebraic sum of the moments about any point in the plane must be zero. The conditions (1) and (2) together give dynamic balancing.

The following two possibilities may arise while attaching the two balancing masses:

a. The plane of the disturbing mass may be in between the planes of the two balancing masses, and
b. The plane of the disturbing mass may lie on the left or right of the two planes containing the balancing masses.

We shall discuss both the above cases one by one.

2.2.1 When the plane of the disturbing mass lies in between the planes of the two balancing masses [2]

Consider a disturbing mass m lying in a plane A to be balanced by two rotating masses m_1 and m_2 lying in two different planes L and M as shown in fig. Let r, r_1, and r_2 be the radii of rotation of the masses in planes A, L and M respectively.

Let l_1=distance between the planes A and L,

l_2=distance between the planes A and M, and

l=distance between the planes L and M

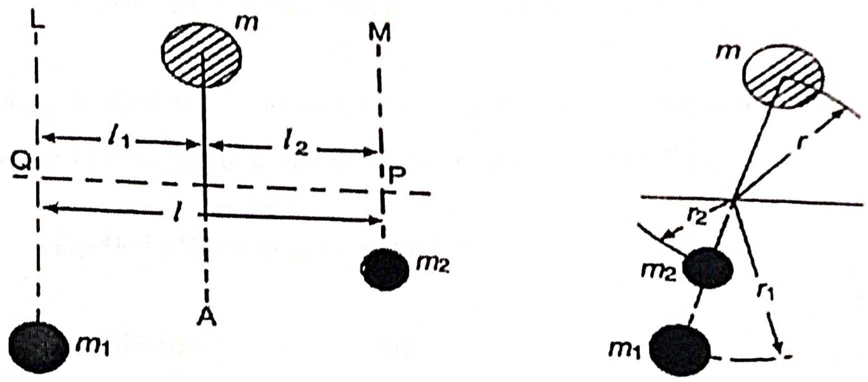


Fig 2.2(a): Balancing of a single rotating mass by two rotating masses in different planes [2]

When the plane of single rotating mass lies in between the planes of two balancing masses.

We know that centrifugal force exerted by the mass m in the plane A,

$$Fc = m\,\omega^2\,r$$

Similarly, the centrifugal force exerted by the mass m_1 in the plane L

$$Fc_1 = m_1 \omega^2 r_1$$

And, the centrifugal force exerted by the mass m_2 in the plane M,

$$Fc = m^2 \omega^2 r_2$$

Since the net force acting on the shaft must be equal to zero, therefore the centrifugal force on the disturbing mass must be equal to the sum of the centrifugal forces on the balancing masses, therefore

$$Fc = Fc_1 + Fc_2 \text{ or } m\omega^2 r = m_1\omega^2 r_1 + m_2\omega^2 r_2$$

$$mr = m_1 r_1 + m_2 r_2 \ldots\ldots\ldots\ldots\ldots \text{(i)}$$

Now, in order to find the balancing force in the plane L (or the dynamic force at the bearing Q of a shaft), take moments about P which is the point of intersection of the plane M and the axis of rotation. Therefore

$$Fc_1 * l = Fc * l_2 \quad \text{or} \quad m_1\omega^2 r_1 * l = m\omega^2 r * l_2$$

$$m_1 r l_1 = m r l_2 \text{ or } m_1 r_1 = m.r * \frac{l2}{1} \ldots\ldots\ldots \text{(ii)}$$

Similarly, in order to find the balancing force in plane M take moments about Q this is the point of intersection of the plane L and the axis of rotation. Therefore

$$Fc_2 * l = Fc * l_1 \text{ or } m_2\omega^2 r_2 * l = m\omega^2 r * l_1$$

$$m_2 r_2 l = m_2 r_2 \frac{l1}{1} \ldots\ldots\ldots\ldots \text{(iii)}$$

It may be noted that equation (i) represent the condition for static balance, but in order to achieve dynamic balance, equation (ii) or (iii) must also be satisfied.

2.2.2 When the plane of the disturbing mass lies on one end of the planes of the balancing masses [2]

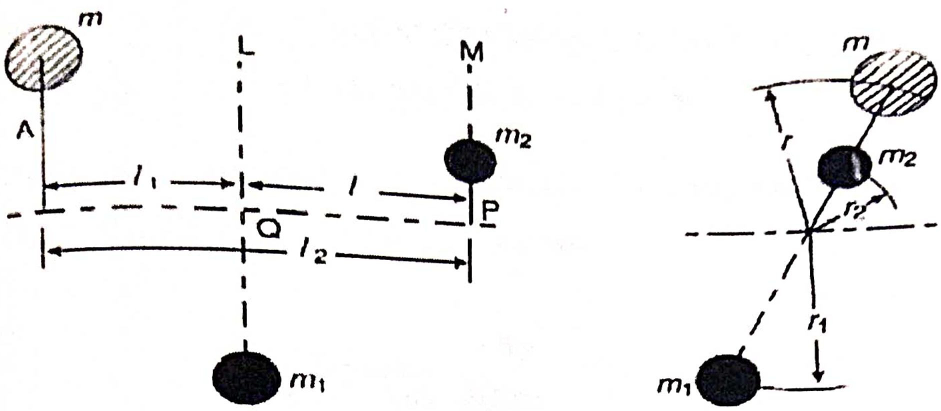


Fig 2.2(b): Balancing of a single rotating mass by two rotating masses in different planes [2]

In this case, the mass m lies in the plane A and the balancing masses lie in the planes L and M, as shown in fig. As discussed above, the following conditions must be satisfied in order to balance the system i.e.

$$F_c + F_{c2} = F_{c1} \quad \text{or}$$

$$m\omega^2 r + m_2\omega^2 r_2 = m_1\omega^2 r_1$$

$$mr + m_2 r_2 = m_1.r_1 \ldots\ldots\ldots\ldots\ldots\ldots \text{(iv)}$$

Now, to find the balancing force in the plane L (or the dynamic force at the bearing Q of a shaft), take moments about P which is the point of intersection of the plane M and the axis of rotation. Therefore

$$F_{c1} * l = F_c * l_2 \text{ or } m_1\omega^2 r_1 * l = m\omega^2 r * l_2$$

$$m_1 r l_1 = m r l_2 \quad \text{or } m_1 r_1 = mr\frac{l2}{1} \ldots \text{(v)} \ldots\ldots \text{[Same equation (ii)]}$$

Similarly, to find the balancing force in the plane M (or the dynamic force at the bearing P of a shaft), take moments about Q which is the point of intersection of the plane L and the axis of rotation. Therefore

$$F_{c2} * l = F_c * l_1 \quad \text{or} \quad m_2\omega^2 r_2 * l = m\omega^2 r * l_1$$

$$m_2 r_2 l = m r_1 l \text{ or } m_2.r_2 = m.r*\frac{l1}{1} \ldots\ldots\ldots \text{(vi)} \ldots\ldots \text{[Same as equation (iii)]}$$

2.3 Balancing of several masses rotating in the same plane [2]

Consider any number of masses (say four) of magnitude m_1, m_2, m_3 and m_4 at distances of r_1, r_2, r_3 and r_4 from the axis of the rotating shaft. Let θ_1, θ_2, θ_3 and θ_4 be the angles of these masses with the horizontal line OX, as shown in fig.

Let these masses rotate, about an axis through O and perpendicular to the plane of paper, with a constant angular velocity of ω rad/sec.

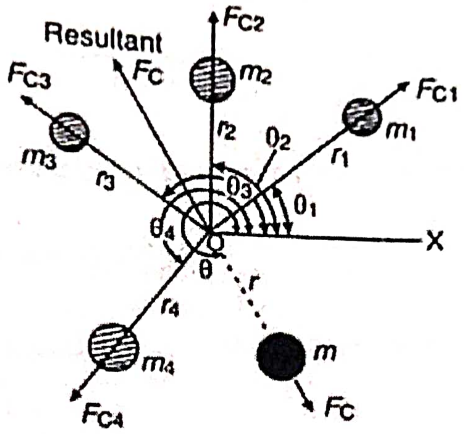


Fig 2.3(a) Space diagram

The magnitude and position of the balancing mass may be found out analytically, or graphically as discussed below:

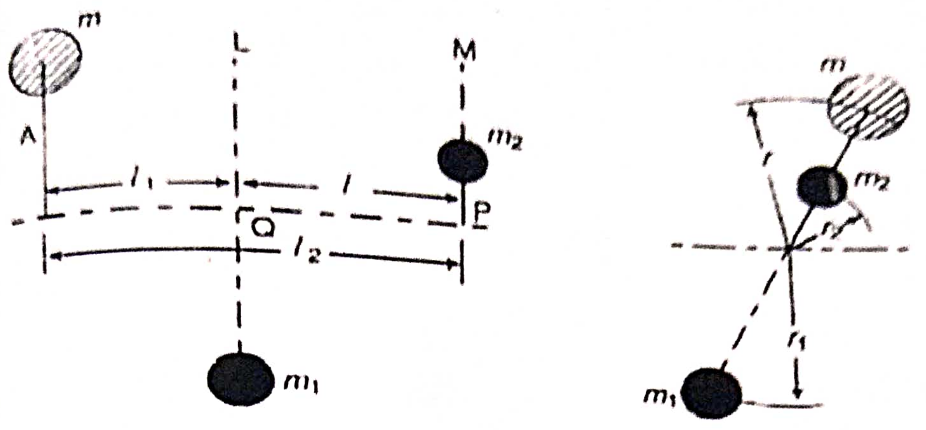
2.3.1 Analytical method [1]

The magnitude and direction of the balancing mass may be obtained analytically, as discussed below:

I. First of all, find out the centrifugal force (or the product of the mass and its radius of rotation) exerted by each mass on the rotating shaft. Since ω2 is same for each mass, therefore the magnitude of the centrifugal force for each mass is proportional to the product of the respective mass and its radius of rotation.

II. Resolve the centrifugal forces horizontally and vertically and find their sums, i.e. $\sum H$ and $\sum V\theta$. We know that

2.2.2 When the plane of the disturbing mass lies on one end of the planes of the balancing masses [2]



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$$F_c + F_{c2} = F_{c1} \quad \text{or}$$

$$m\omega^2 r + m_2\omega^2 r_2 = m_1\omega^2 r_1$$

$$mr + m_2 r_2 = m_1.r_1 \text{.................. (iv)}$$

Now, to find the balancing force in the plane L (or the dynamic force at the bearing Q of a shaft), take moments about P which is the point of intersection of the plane M and the axis of rotation. Therefore

$$F_{c1}*l = F_c*l_2 \text{ or } m_1\omega^2 r_1*l = m\omega^2 r*l_2$$

$$m_1 r l_1 = m r l_2 \quad \text{or } m_1 r_1 = mr\frac{l2}{1} \text{.... (v)..... [Same equation (ii)]}$$

Similarly, to find the balancing force in the plane M (or the dynamic force at the bearing P of a shaft), take moments about Q which is the point of intersection of the plane L and the axis of rotation. Therefore

$$F_{c2}*l = F_c*l_1 \quad \text{or } m_2\omega^2 r_2* l = m\omega^2 r*l_1$$

$$m_2 r_2 l = m r_1 l \text{ or } m_2.r_2 = m.r*\frac{l1}{1} \text{....... (vi)...... [Same as equation (iii)]}$$

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Consider any number of masses (say four) of magnitude m_1, m_2, m_3 and m_4 at distances of r_1, r_2, r_3 and r_4 from the axis of the rotating shaft. Let θ_1, θ_2, θ_3 and θ_4 be the angles of these masses with the horizontal line OX, as shown in fig.

Let these masses rotate, about an axis through O and perpendicular to the plane of paper, with a constant angular velocity of ω rad/sec.

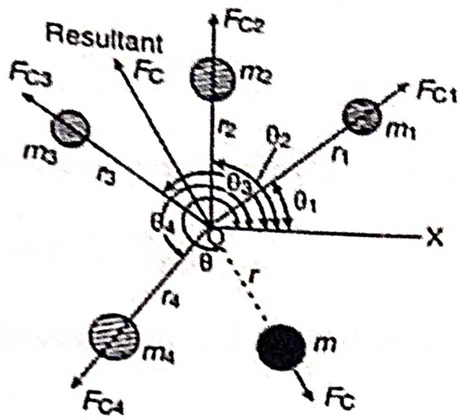


Fig 2.3(a) Space diagram

The magnitude and position of the balancing mass may be found out analytically, or graphically as discussed below:

2.3.1 Analytical method [1]

The magnitude and direction of the balancing mass may be obtained analytically, as discussed below:

I. First of all, find out the centrifugal force (or the product of the mass and its radius of rotation) exerted by each mass on the rotating shaft. Since ω_2 is same for each mass, therefore the magnitude of the centrifugal force for each mass is proportional to the product of the respective mass and its radius of rotation.

II. Resolve the centrifugal forces horizontally and vertically and find their sums, i.e. $\sum H$ and $\sum V\theta$. We know that

Sum of horizontal components of the centrifugal forces,

$\sum H = m_1 r_1 \cos\theta_1 + m_2 r_2 \cos\theta_2 + \ldots\ldots$ and

Sum of vertical components of the centrifugal forces,

$\sum V = m_1 r_1 \sin\theta_1 + m_2 r_2 \sin\theta_2 + \ldots\ldots\ldots\ldots$

III. Magnitude of the resultant centrifugal force,

$$F_C = \sqrt{(\sum H^2 + \sum V^2)}$$

IV. If e is the angle, which the resultant force makes with the horizontal, then

$$\tan\theta = \sum V / \sum H$$

V. The balancing force is then equal to the resultant force, but in opposite direction.

VI. Now find out the magnitude of the balancing mass, such that

$$F_C = m.r$$

Where: m= Balancing mass, and r= radius of rotation.

2.3.2 Graphical method

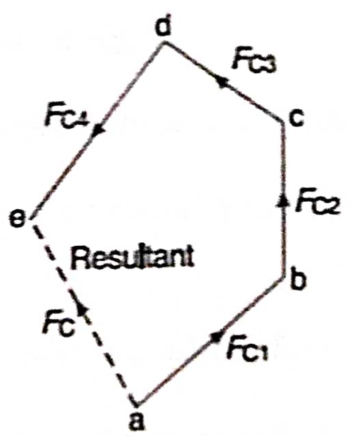


Fig 2.3(b): Vector diagram

Table 2.1: Tabular Column for several masses in a single plane

Mass	Radius	Force
m_1	r_1	$m_1.r_1$
m_2	r_2	$m_2.r_2$
m_3	r_3	$m_3.r_3$
m_4	r_4	$m_4.r_4$
M	R	m.r

The magnitude and position of the balancing mass may also be obtained graphically as discussed below:

1. First of all, draw the space diagram with the position of the several masses, as shown in Fig. (a).

2. Find out the centrifugal force (or product of the mass and radius of rotation) exerted by each mass on the rotating shaft and tabulate is as shown in the tabular column.

3. Now draw the vector diagram with the obtained centrifugal force (or the product of the masses and their radii of rotation), such that AB represent centrifugal force exerted by the mass m_1 (or $m_1.r_1$) in magnitude and direction to some suitable scale. Similarly, draw bc, cd and de too represented centrifugal forces of other masses m_2, m_3 and m_4 (or m_2. $r_2 m_3.r_3$ and $m_4.r_4$).

4. Now, as per polygon law of forces, the closing side a represents the resultant force in magnitude and direction, as shown in Fig. (b).

5 .The balancing force is, then, equal to the resultant force, but in opposite direction.

6. Now find out the magnitude of the balancing mass (m) at a given radius of rotation (r), such that,

$$m.\omega^2.r = \text{Resultant centrifugal force}$$

$$m.r = \text{Resultant of } m_1.r_1,\ m_2.r_2,\ m_3.r_3 \text{ and } m_4.r_4.$$

2.4 Balancing of several masses rotating in different planes [2]

When several masses revolve in different planes, they may be transferred to a reference plane (briefly written as P.P.), which may be defined as the plane passing through a point on the axis of rotation and perpendicular to it. The effect of transferring a revolving mass (in one plane) to a reference plane, together with a couple of magnitude equal to the product of the force and the distance between the plane of rotation and the reference plane. In order to have a complete balance of the several resolving masses in different planes, the following two conditions must be satisfied:

1. Sum of centrifugal force must be equal to zero i.e., Resultant force must be equal to zero.

i.e., Sum of horizontal component of the centrifugal forces,

$$\sum H = m_1 r_1 \cos\theta_1 + m_2 r_2 \cos\theta_2 + \ldots\ldots\ldots\ldots = 0 \text{ and}$$

Sum of vertical component of the centrifugal forces,

$$\sum V = m_1 r_1 \sin\theta_1 + m_2 r_2 \sin\theta_2 + \ldots\ldots\ldots = 0$$

In other words force polygon must be close

2. Sum of the couples about any plane must be equal to be zero i.e., Resultant couple must be zero.

$$\sum H = m_1 r_1 l_1 \cos\theta_1 + m_2 r_2 l_2 \cos\theta_2 \ldots\ldots\ldots\ldots = 0$$

$$\sum V = m_1 r_1 \sin\theta_1 + m_2 r_2 \sin\theta_2 \ldots\ldots\ldots\ldots\ldots\ldots = 0$$

Steps to draw couple polygon and force polygon

1. Represent four masses m_1, m_2, m_3 and m_4 revolving in planes 1, 2, 3 and 4 respectively as shown in figure (a) and the relative angular positions of these masses as shown in the end view [Fig. (b)].

2. Take any one of the planes, say L as the reference plane (R P). The distance of all the other planes to the left of the reference plane may be regarded as negative ('-1'), and those to the right as positive ('+1').

3. Tabulate the forces and couples as shown in the tabular column

Table 2.2: Tabular column for several masses in a several planes.

plane	Mass(M)	Radius(r)	Centrifugal force÷ω2 (m.r)	distance from ref plane L	couple÷ω2 (m.r.l)
1	m_1	r_1	$m_1.r_1$	$-l_1$	$-m_1.r_1.l_1$
L(R.P)	m_l	r_L	$m_l.r_l$	0	0
2	m_3	r_2	$m_2.r_2$	l_2	$m_2.r_2.l_2$
3	m_4	r_3	$m_3.r_3$	l_3	$m_3.r3.l_3$
m	m_m	r_m	$m_m.r_m$	l_m	$m_m.r_m.l_m$
4	m_4	r_4	$m_4.r_4$	l_4	$m_4.r_4.l_4$

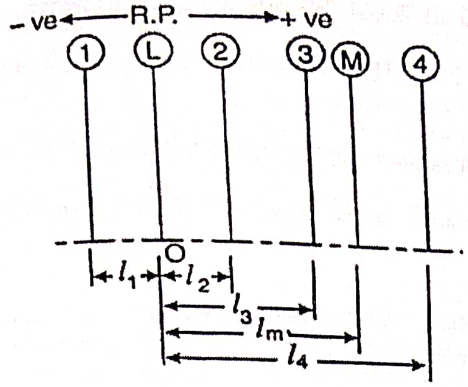


Fig 2.4(a): Position of planes of the masses

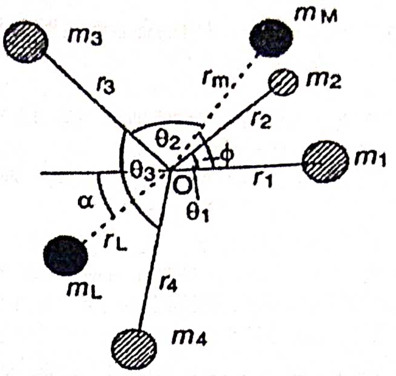


Fig 2.4(b): Angular position of the masses

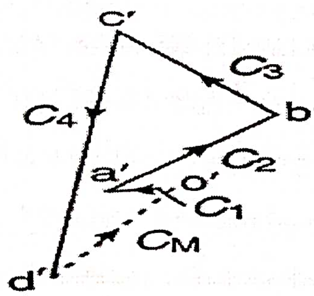


Fig 2.4(c): Couple polygon

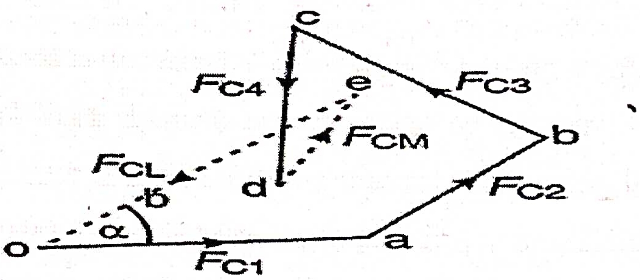


Fig 2.4(d): Force polygon

4. A couple may be represented by a vector drawn perpendicular to the plane of the couple. The couple C_1 introduced by transferring m1 to the reference plane through O is proportional to $m_1.r_1.l_1$ and act in a plane OM_1 and perpendicular to the paper. The vector representing this couple is drawn in the plane of the paper and perpendicular to OM_1 as shown by OC_1 in Fig. similarly the vectors OC_2, OC_3 and OC_4 are drawn perpendicular to OM_2, OM_3 and OM_4 respectively and in the plane of the paper.

5. Now draw the couple polygon as shown in Fig. (c). The vector d 'o' represented the couple. Since the balanced couple CM is proportional to $m_m.r_4.l_m$, therefore

$c_m = m_m.r_m.l_m = d$ 'o'*scale.

Balancing mass in the plane, $m_m = \dfrac{d'o' * scale}{rm.lm}$

From the expression, the value of the balancing mass m_4 in the plane M may be obtained, and the angle of inclination ϕ of this mass may be measured from Fig. (c). [w. r. t m_l (ccw)].

6. Now draw the force polygon as shown in Fig. (d). the vector eo (in the direction from e to o) represents the balanced force. Since the balanced force is proportional to $m_l.r_l$, therefore,

$$m_l.r_m = eo*scale \quad \text{or} \quad m_l = \frac{eo*scale}{rl}$$

From this expression, the value of the balancing mass m_l in the plane L may be obtained and the angle of inclination α of this mass with the horizontal may be measured from Fig. (d) [w.r.t m_l (ccw)].

Gideon A. Almas [6] was attempted to studied principles of static balancing and dynamic balancing. He develop theoretical linear dynamic balance model by assuming various variables with these model residual error of static MER(measurement, error, reconciliation) due to dynamic effect can be estimated. Dynamic balancing is proposed as an estimation of flow and relevant variables by filtering method was applied to the model to balance. The error estimated and obtained by this way is significantly than that of static MER estimates when the process is nearly in steady state. The main problem of balancing is that it requires not only the redundant observation of force but also the observation of couples.

D.J.Rodriguez et al [7] was tried to prevent an analysis of two planes automatic balancing device for of the device depends on non-tribal parameters of the system. Due to this method explained rigid rotor. His use legrangial equation in terms of theoretical model to identify kinetic and potential energy of the rotor cause an elastic and plastic deflection. Ball bearing which are free to around the bearing rear are used to eliminate imbalance due to shaft eccentricity are misalignment. They clearly found that automatic balancing is highly nonlinear device and hence stability him will more requires design requirement of the line (ABB).

P.Kalinichenko and A. Suprun, b [8] attempted to work on effective modes of axial balancing of centrifugal pump rotor, they developed of a node o axial balancing of a centrifugal pump rotor are in process stated. Constructional solutions of equilibrating devices on the basis of hydrostatical seal are observed. Comparison with existing balancing devices is held. They observed modes of axial balancing of a rotor of the centrifugal pump are more effective and reliable in comparison with the balance piston, a hydro bearing and look alike devices. The application in the system of axial relief of a self-aligning collar allows to diminish an axial gabarit of the pump and to slash metal consumption of a product.

Anil. B chaudry and Klaus Jürgen bathe [9] presented a method for the analysis of contact between two or more three dimensional bodies. The surfaces of the container bodies are discredited using quadrilateral surface segments. A Lagrange multiplier technique is employed to impose that in the contact area, the surface displacement of the contacting bodies is compatible with each other. Distributed contact traction over the surface segments are calculated from the externally applied forced, inertia forces and internal element stresses. Using the segments traction coulombs law of friction is enforced in a global sense over each surface segment. The time integration of dynamic response is performed using the new mark method with parameters delta=1/2 and alpha=1/2 using these parameters the energy and momentum balance inertia for the contacting bodies are satisfied accurately when a reasonably small time step is used. The applicability of the algorithms is illustrated by selecting sample numerical solution to static and dynamic contact problems.

CHAPTER 3

SCOPE OF PRESENT WORK

Due to the problem involved in the current balancing testing apparatus which will there in the lab and other type apparatus, it is necessary to design, develop and fabricate new apparatus for conducting reliable tests.

Many types of balancing apparatus are available in the market as well as also in the laboratories. All types of balancing apparatus do not give reliable results. It is found that while conducting the balancing tests in the design laboratory an appreciable remarks was observed and facing so many complexities. The complexities observed are.

- Time required to conduct the test is more.
- Inadequacy while taking the readings.
- Putting of small masses to get balance is risky.
- Difficulty while lifting up and lifting down the balancing frame.

Balancing apparatus possesses the above problems is shown in the figure 3.1

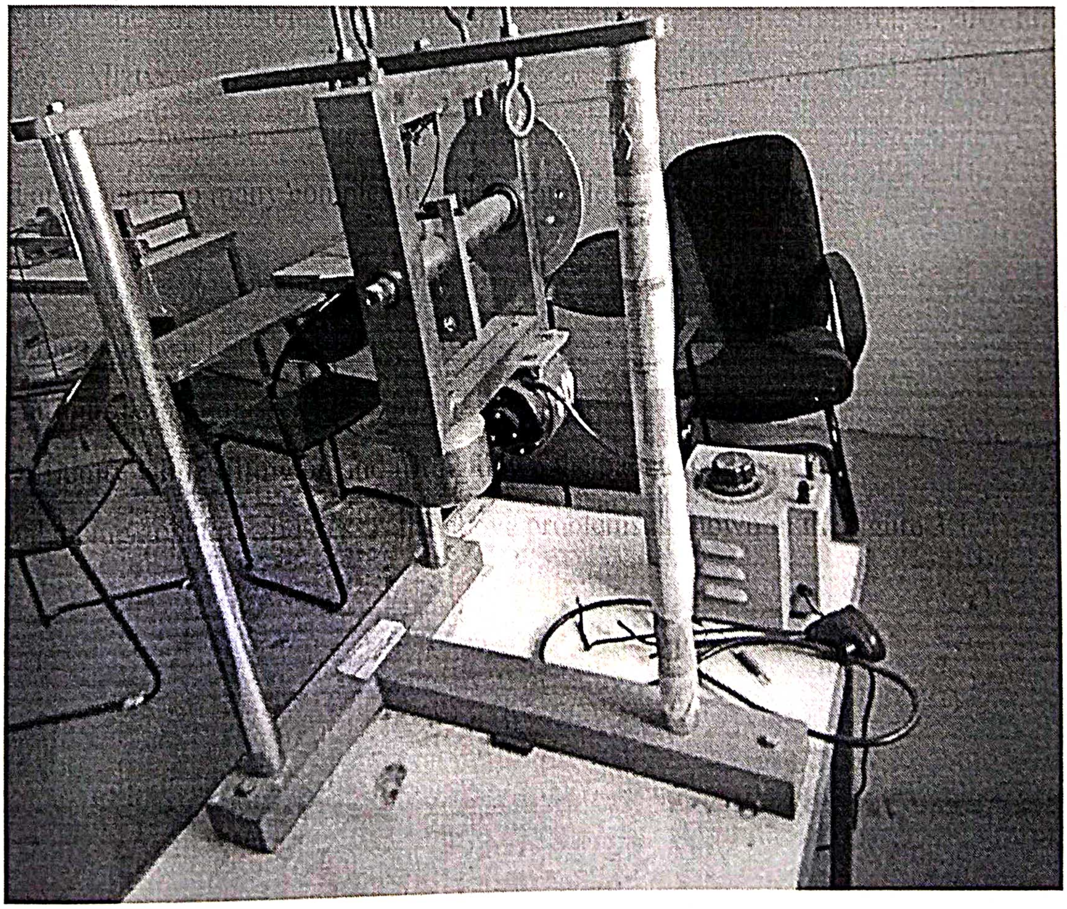


Fig 3.1: Portable Type Balancing Machine

Hence designers and fabricators of laboratory testing equipments suppliers adapting conventional method of balancing. In conventional method also many problems were found out while conducting the test. Problems found in conventional method of balancing are, all discs of balancing machine, weights, bearings, pulleys are carried by single plat form along with motor. This entire assembly was carried by four helical coiled springs, this may cause large amount of tension on the springs and may create excess vibrations which leads to failure of springs. Also while conducting dynamic testing an eccentricity of discs produce large amount of loads on the apparatus.

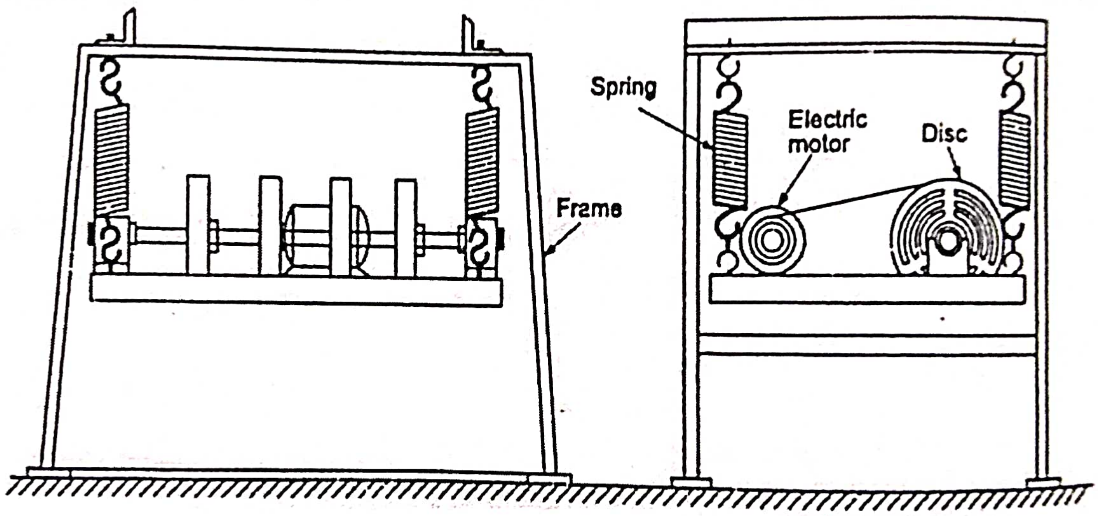


Fig 3.2: Conventional Balancing Machine

Hence we are planning to design, develop and fabricate new type of balancing apparatus and tried to avoid difficulties facing in present available apparatus. Present study involves

- Identification of problems involved in the current test apparatus
- Design of parts of apparatus
- Modelling of parts using standard CAED tool.
- Fabrication and assembling of parts
- Calibration of apparatus
- Conducting of test
-

CHAPTER-4

DESIGN OF THE PARTS

In the proposed work of project there are several elements which works together in order to achieve desired output. Every work or project belongs to engineering is starts with ideal assumption followed by theoretical approach. Design is the base requirement for mechanical engineers for when they are solving any type of problem. Design helps us to find suitable dimensions and of parts and also they are working under safe zone or not. Hence in order to follow the basic phenomena of the work, some major parts of the machine are designed and discussed below. The parts designed are springs, Shaft, V-belt, Pulley, Bearings etc.

4.1 DESIGN OF SPRING [3, 4, and 5]

Weight of the frame=15kg

i.e. W =F=147.15 N

τ_y=690Mpa=690N/mm^2 (from DDHB for Chrome vanadium steel)

G=79340Mpa=79.34GPa FOS = 2

$\tau = \frac{\tau y}{FOS} = \frac{690}{2}$=**345N/mm^2**

1) Diameter of Wire

Shear stress $\tau = \frac{8FDK}{\pi d^3}$

Wahl's stress factor (K) is given by,

$K=\frac{4c-1}{4c-4}+\frac{0.615}{c}$

$K=\frac{4*6-1}{4*6-4} + \frac{0.615}{6}$

K=1.2525

Spring index $c=\frac{D}{d}$

Assume spring index as c=6

$6=\frac{D}{d}$; D=6d

$345=\frac{8*147.15*6d*1.2525}{\pi*d^3}$

d=2.49mm

Select standard diameter of wire from DDHB

d= 5mm

2) Diameter of coil.

$C=\frac{D}{d}$; $6=\frac{D}{5}$

Mean diameter of coil D=30mm

Outer diameter of coil D_0=30+5=35mm

Inner diameter of coil D_i=30-5=25mm

3) Number of coil or turns

Deflection $y = \frac{8FD^3i}{d^4G}$

Assume a deflection 5-10mm

$5 = \frac{8*147.15*30^3*i}{5^4*79.34*10^3}$ = i=7.42

Number of active turns i=7

4) Free length

$l_0 \geq (i+n)\,d + y + a$

Clearance a =25% of maximum deflection=$\frac{25}{100}$*4.48=1.121mm

Assume squared and ground end, n=2

Total number of turns, i′=i+n=7+2=9

$l_0 \geq (7+2)*5+5+1.121=$ **51.121mm**

Actual maximum deflection $y = \frac{8*147.15*30^3*7}{5^4*79340} =$ **4.48mm**

Actual free length $l_0 \geq (7+2)*5+5+1.121+\frac{25}{100}*4.48=$ **52.24mm**

5) Pitch

$P = \frac{l_0 - 2d}{i} = \frac{52.24-2*5}{7} =$ **6mm**

6) Required stiffness or required rate of spring

$F_0 = \frac{F}{Y} = \frac{147.15}{5} =$ **30N/mm**

7) Actual stiffness or actual rate of spring

$F_0 = \frac{d^4 G}{8iD^3} = \frac{5^4*79340}{8*7*30^3} =$ **32.79N/mm**

8) Total length of wire

$l = \pi D i'$

Where i′=i+1=7+2=9

$l = \pi*30*9 =$ **848.23mm**

4.2 DESIGN OF V-BELT [3, 4, and 5]

1) Materiel of v-belt

V-Belts are made of polyester fabric with rubber reinforcement.

Allowable tensile stress = 20.25 MPa

$\sigma_{allowable}$= 20.25Mpa P= 0.166HP

Induced Tensile stress, $\sigma_{induced}$ P=0.166*0.746 KW

W.K.T, $P=\frac{2\pi*N*T}{60000}N$

$T=\frac{0.166*0.746*60*1000}{2\pi*350}$

T= 3378.7 N-m

ALSO, Torque=Force*Radius

3378.7 =Force*0.0375

Radius=0.75/2

=0.375

Force= 90.09 KN

Induced Tensile Stress,

$\sigma_{induced}$=Force/Area

= $(90.09*10^3)/(8.9125*10^{-3})$

=10.10Mpa

For Design to be safer,

$\sigma_{allowable}>\sigma_{induced}$

20.6>10.10

Hence the design of belt is on the safer side.

2) Assuming ''A" cross section of the V-belt

Table 4.1: Standard dimension of "A" c/s v-belt

Sl. No	Power Range (kw)	Top Width, W (mm)	Thickness, T (mm)	Min diameter of pulley, d(mm)
1	Up to 3.5	13	8	75

Centre Distance =C

C_{max}=max centre distance

C_{min}=min centre distance

$C_{max}=2\{D+d\}$

$=2\{310+75\}$

=770mm

$C_{min}= 0.55\{D+d\}-T$

=0.55{310+75}-8

=203.75mm

Approximated centre distance= **500mm,**

Correct centre distance, $C= \frac{L}{4} - \frac{\pi(D+d)}{8} + \sqrt{\left\{\frac{L}{4} - \frac{\pi(D+d)}{8}\right\}2 - \frac{(D-d)^2}{8}}$

=409-151.189+245.709

=462.37mm

Length of the belt, $L = 2C + \pi/2(D+d) + \frac{(D-d)2}{4C}$

=1632.36mm

For c/s "A" standard length of v-belt is 1560 mm.

4.3 DESIGN OF PULLEY [3, 4, and 5]

Materiel for V-grooved pulley cast iron grade FG-250.

Table; The standard dimension of V-grooved pulley for "A" c/s of the belt

Sl no	c/s	l_p	b	h	α^0	d_p	g	Outside diameter	f
1	A	11	2.75	11	34	Up to 118	13	dp+5.5	9.12

All dimensions are in mm

b= min height of groove above pitch line.

h = min depth of groove below pitch line.

l_p = pitch width of pulley groove.

α^o = groove angle.

d_p= pitch diameter.

4.4 DESIGN OF SHAFT [3, 4, and 5]

Material used for the shaft is Mild steel

Assuming overall length of the shaft=600mm

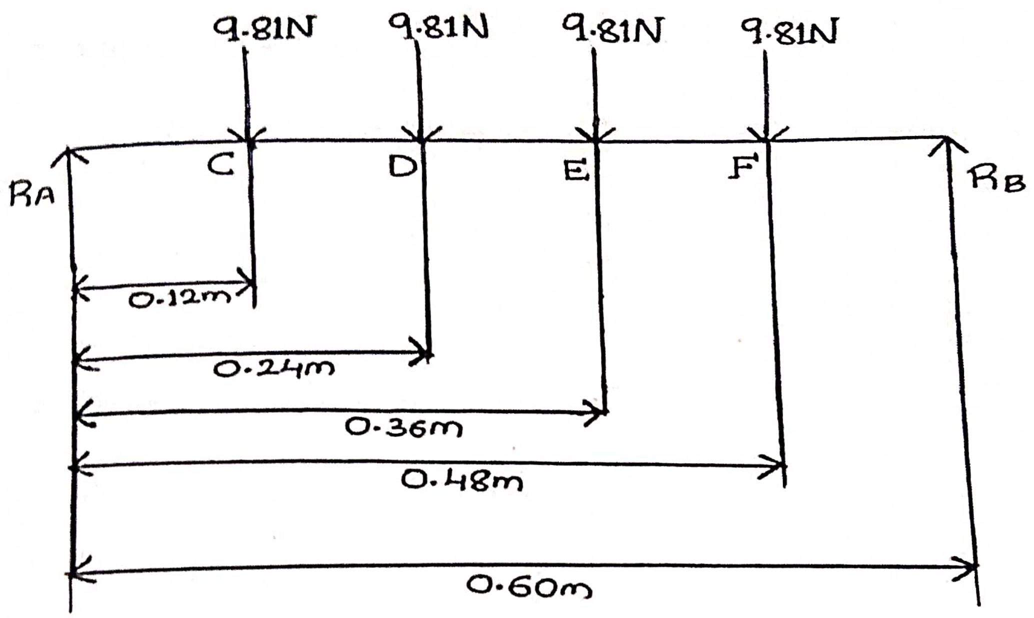


Fig 4.1: Vertical Load Diagram

$R_A + R_B = 39.24N$

1) **Bending moment at point B**

$(9.81*0.12) + (9.81*0.24) + (9.81*0.36)+(9.81*0.48) - (R_A*0.60) = 0$

$R_A = 19.62N$

2) **Bending moment at point A**

$(9.81*0.12) + (9.81*0.24) + (9.81*0.36) + (9.81*0.48) - (R_B*0.60) = 0$

$R_B = 19.62N$

3) **Bending moment at point C**

$M_C = 19.62*0.12$

M_C= 2.354N-m

4) **Bending moment at point D**

M_D= (19.62*0.24) – (9.81*0.12)

M_D= 3.5316N-m

5) **Bending moment at point E**

M_E= (19.62*0.36) – (9.81*0.24) – (9.81*0.12)

M_E= 3.5316N-m

6) **Bending moment at point F**

M_F= (19.62*0.48) – (9.81*0.36) – (9.81*0.24) – (9.81*0.12)

M_F= 2.354N-m

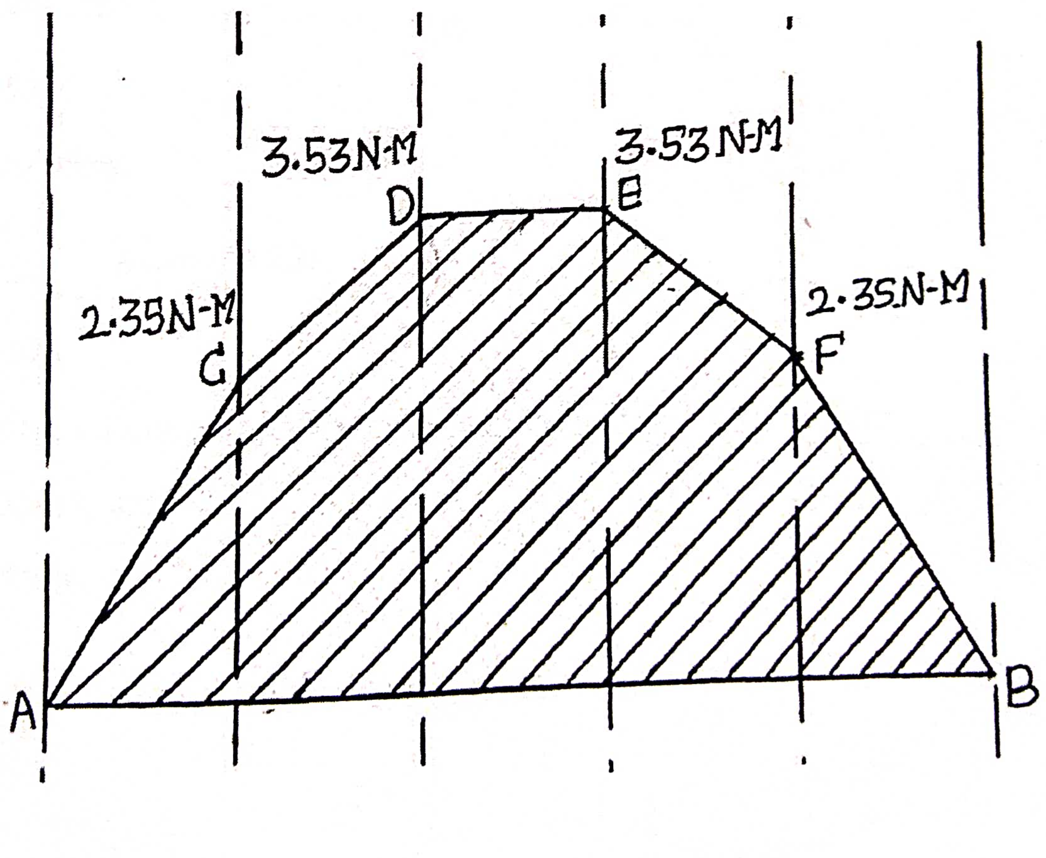


Fig 4.2: Vertical bending moment diagram

Calculate the Tensions

W.K.T $T_1=\sigma_1*A$ Assume $\sigma_1=2.26N/mm^2$

Assume A cross section belt from DDHB for V-belt

Nominal top width (b) =13mm

Nominal thickness (t) =8mm

W.K.T $\tan=\frac{h}{t}=\tan(20)=\frac{h}{8}$

h= 2.91mm

$A=\frac{1}{2}(13+7.18)*8=80.72mm$

$T_1=2.26*80.72$

=182.4272 N

W.K.T D=50.28mm, d=25.40mm, C=150mm.

To find angle of contact on smaller pulley

$\theta s=\pi-2\sin^{-1}\frac{D-d}{2C}$

θs= 175.14°

θs= 3.04radians

$\frac{T1}{T2}=\frac{\mu\theta}{e^{\sin\alpha}}$ Assume $\mu=0.25$

T_2= 454.97N

From the two calculations substitute maximum term as T1 and minimum as T2

T=454.97+182.427

T = 637.39N

To find horizontal force bending momentum

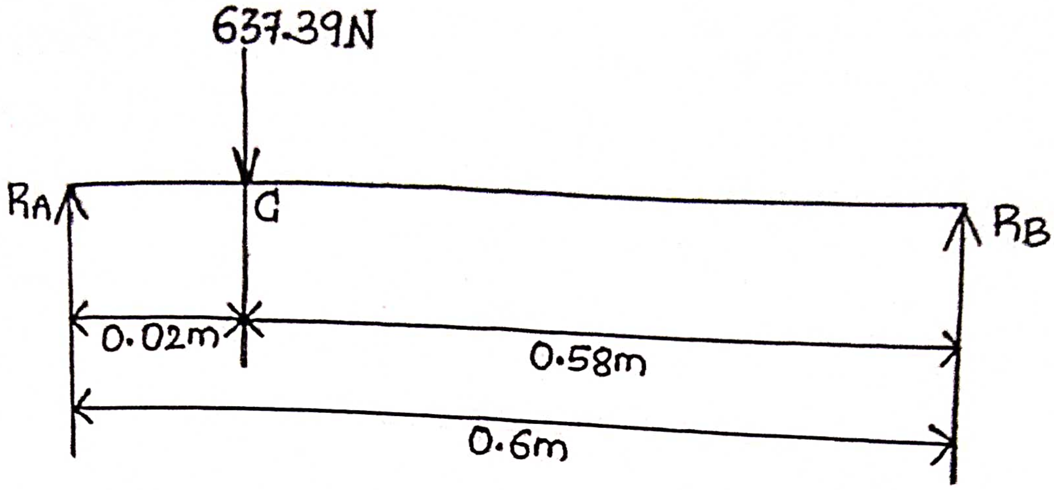


Fig 4.3: vertical load diagram

$R_A + R_B = 637.39N$

$R_A = 637.39 - R_B$

Taking bending moment at A

$637.39*0.02 - R_B*0.58 = 0$

$R_B = 21.97N$

$R_A = 637.59 - 21.97$

$R_A = 615.41N$

Taking bending moment at B=0

Taking bending moment at C

$M_C = (21.97*0.58)$

$M_C = 12.74N\text{-}m$

Taking bending moment at A

M_A= (21.97*0.6)-(637.39*0.02)

M_A= 0.0N-m

Resultant bending moment at C

Bending moment at C=$\sqrt{12.74^2 + 2.35^2}$

C=12.95N-m

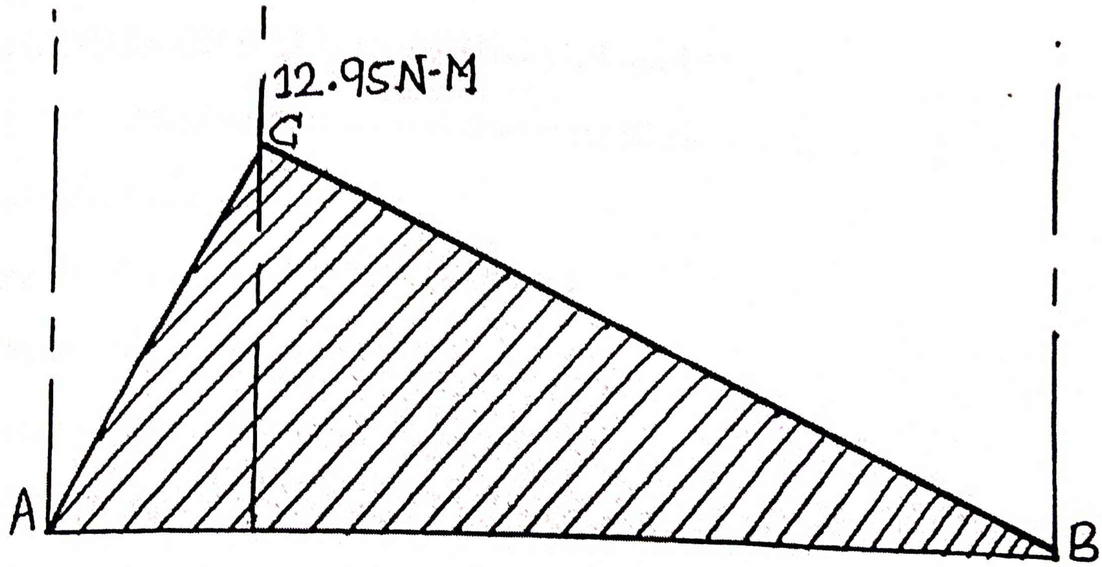


Fig 4.4: Horizontal Bending Moment Diagram

Diameter of solid shaft subjected to combined bending and torsion according to maximum to maximum shear stress theory.

$$D = \left[\frac{16}{\Pi \tau cd}\{(K_b M_b)^2 + (K_t M_t)^2\}^{1/2}\right]^{1/3}$$

$\tau_{cd} = \frac{\tau y}{FOS}$ select c30 steel

$= \frac{\sigma y}{2*FOS}$

$= \frac{294.2}{2*3}$

τ_{cd} = 49.03N/mm

Assume K_b= 1.5, K_t = 1.25 from DDHB

$$P = \frac{2\pi NT}{60,000}$$

$$0.125 = \frac{2\pi * 50 * T}{60,000}$$

T = 23.87*10³N-mm

$$D = [\frac{16}{\pi * 49.03} * \{(1.5*12.79)^2 + (1.25*13.87*10^3)^2\}^{1/2}]^{1/3}$$

D = 15.41mm.

Based on shaft basis 'A' the preferred shaft size is 20mm.

4.4 DESIGN OF BALL BEARING [3, 4, and 5]

For radial ball bearing, the basic static load rating (C_0) is given by

$C_0 = f_0 * Z * D^2 * \cos\alpha$

I=Number of rows of rolls of balls in any one bearing.

Z=Number of ball per row.

D=diameter of ball, in mm.

α=Nominal angle of contact.

f_0=A factor depending upon the type of bearing.

Value of factor (f_0) for bearing made of hardened steel.

For radial ball bearing (f_0)=12. 3, Assuming, $\alpha = 15^0$

$C_0 = 12.3*1*10^2*9*\cos 15$

C_0=10.9KN

The standard dynamic load rating is 11200N.

For the above dynamic load rating the bearing number is 6205.

The bearing designated by the number 6205 which means that the bearing is of light series(2) & the last two digits when multiplied by 05 gives the bore diameter of bearing, 05*05, i.e. 25mm.

d=25mm=inner diameter of the bearing or bore.

D=52mm=outer diameter of the bearing.

B=15mm=Axial width of bearing.

Inner race is rotating & outer race is stationary.

4.5 MOTOR SPECIFICATIONS [3, 4, and 5]

Single phase AC induction motor

Power=1/6HP

=0.16HP

Power =125watts.

Motor speed=6000 R.P.M

220/230 volts,

1 phase 50 HZ, 0.95A.

CHAPTER 5

FABRICATION OF THE APPARATUS

In the present work few elements of apparatus are purchased and remaining parts are fabricated. The parts which are purchased and fabricated are designed theoretically using fundamental equations and found out the shapes and sizes. Parts which were purchased are Closed coiled helical springs, Electric motor, Switches, Bolts, bearings etc. These parts were purchased based on the specification and results obtained from the designed calculations. In the same manner parts which are not readily available are fabricated by various manufacturing methods for pre designed parts. Fabricated parts are Shaft, Discs, Outer frame, Base frame etc. These parts were manufactured based on the specification and results obtained from the designed calculations. In present work conventional manufacturing methods adopted are welding, casting, turning, powder coating, polishing, grinding etc.

5.1 DISCRIPTION OF PARTS USED

5.1.1 Radial Ball Bearing

In our project we have selected radial ball bearing. It consist of four main elements an inner race, an outer race, the rolling elements of balls and a cage to keep the rolling element apart. The inner and outer race contains hardened tracks in which the rolling elements roll.

This bearing is good at with standing radial loads. The main load is transferred from the rotating shaft to its support by rolling contact.

5.1.2 Shaft

Shaft is 450mm in length and diameter of 20mm is used for higher stabilities. The ends of the shaft are supported by two ball bearings.

It usually carry disk, four mass attaching element two pro circle arrangements.

5.1.3 Disc

This disc is usually placed at the centre of shaft which is used to solve problems in single planned and also problems in several planes. The masses are usually attached inside the slots /grooves provided in the disc.

It even acts as larger pulley at one end to which V-belt is fixed, which helps in the running in the shaft.

5.1.4 Mass Attaching Elements

These are specially design elements which mainly reduce the cost of disc. Few other purposes are for the provision of variation of length between the shaft and also for the variation of the radius of rotation which is not possible in case of the disc.

5.1.5 V-Belts

In this apparatus a cross section belt is used to transmit power from the smaller pulley in motor to the larger pulley in the shaft in vertical direction.

5.1.6 Motor

As per the calculation and the requirements a single phase AC induction motor is used which is having a power of 125 watt running at 6000 rpm operating at voltage of 230 volts and 50 cycles/sec.

5.1.7 Pro-Circles

Pro-circles are used to provide an angular reference for the disc as well as for the mass attaching elements.

5.1.8 Springs

Springs rather than for hanging purpose it is mainly used to indicate balanced condition and the vibrations present in the apparatus during the unbalanced condition.

5.2 FABRICATED PARTS

5.2.1 Disc:

The material selected for the disk is aluminium because of its less weight density. The designed disc was modelled from Solid edge and pattern has been made according to modelled dimensions. The pattern has moulded, in the metallic mould and molten aluminium metal was poured into the prepared cavity. After solidification disc has been removed from the mould and then cleaned with suitable solution. Unnecessary elements such as gates, raisers, pouring basin, runner and gates were removed from disc by parting operation. Then the disc was sent to grinding operation. As shown in figure 5.1.

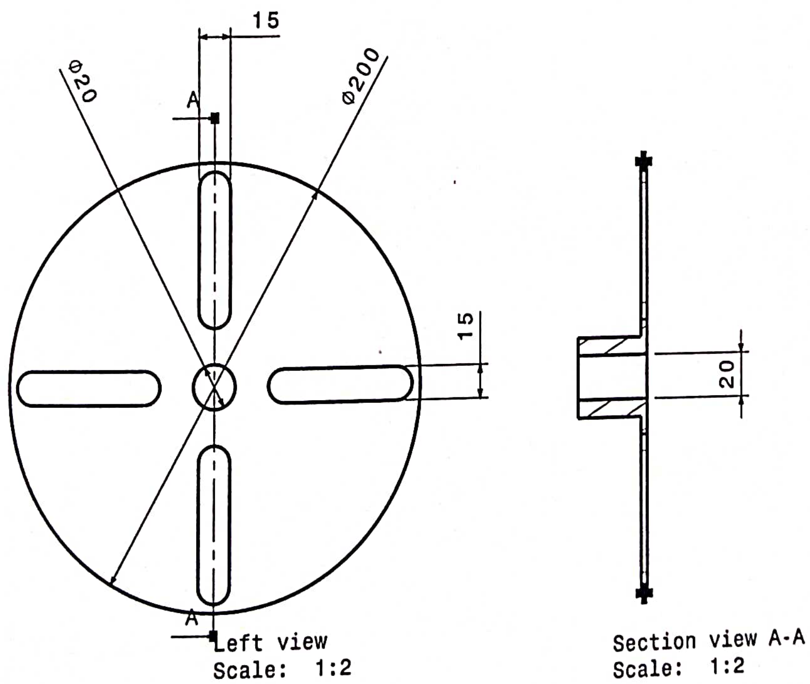


Figure 5.1: Disc

5.2.2 Shaft:

The material selected for shaft is Mild steel. The shaft was designed and solid round raw material of diameter 40 mm has been purchased. Unwanted material of purchased solid round raw material is removed using conventional lathe. The raw material was held between two revolving centres and material was removed using two operations. These operations are rough turning and smooth turning. Facing was done to two sides and drilled the two holes each at opposite to the diameter of 20 Rough turning was done by using 2 mm depth of cut and smooth turning was done by 1 mm depth of cut. As shown in figure 5.2.

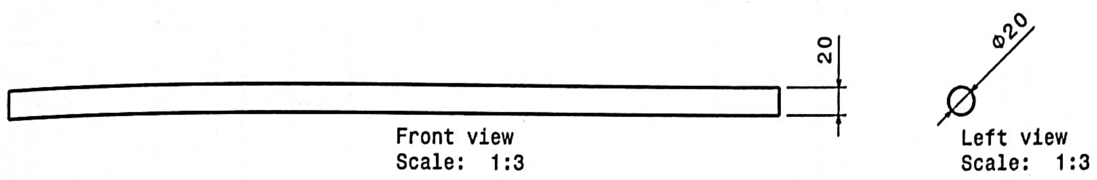


Figure 5.2: shaft

5.2.3 Outer frame:

The material selected for outer frame is mild steel sheet metal of 2 mm thick. The total surface area of outer frame has been calculated and sheet metal was cut into the calculated area. Bending operation's at all edges was done using CNC bending machines. Perforated and blanking operations are done at the required edges. Then they obtained part has been sent to galvanisation and powder coating. Galvanizing is done by etching the frame in the chemical solution and powder coating is done by impinging the powder at the rated velocity. Then the frame is allowed to dry for few minutes and used as outer frame. As shown in figure 5.3.

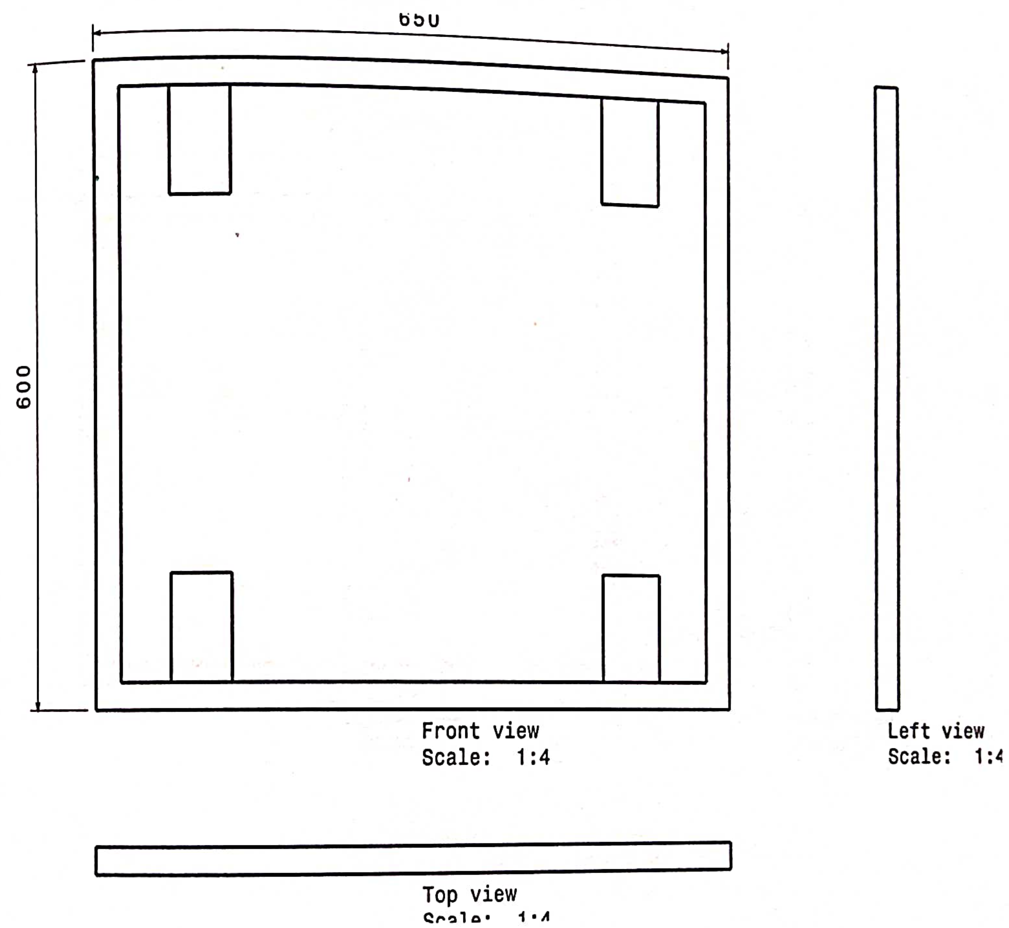


Figure 5.3: Outer frame

5.2.4 Inner frame:

The material selected for inner frame is mild steel sheet metal of 2 mm thick. The total surface area of inner frame has been calculated and sheet metal was cut into the calculated area. Bending operation's at all edges was done using CNC bending machines. Perforated and blanking operations are done at the required edges. Then they obtained part has been sent to galvanisation and powder coating. Galvanizing is done by etching the frame in the chemical solution and powder coating is done by impinging the powder at the rated velocity. Then the frame is allowed to dry for few minutes and used as inner frame. As shown in figure 5.4.

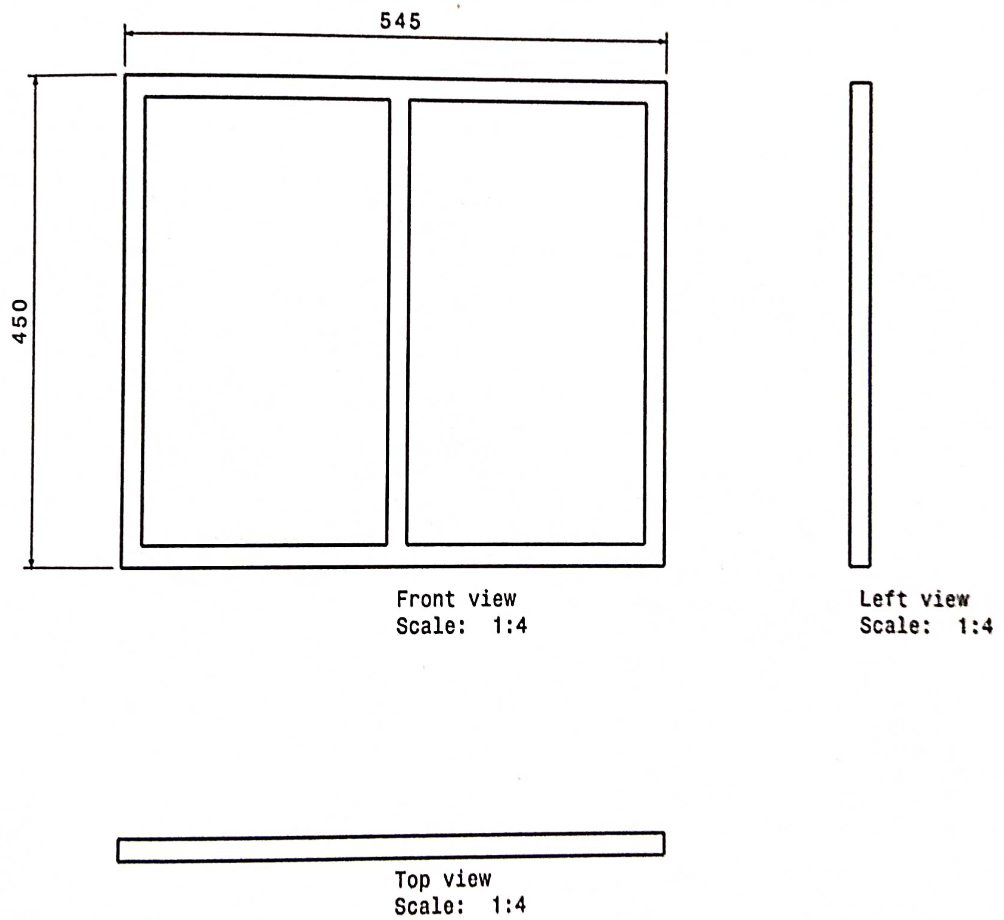


Figure 5.4: Disc

CHAPTER-6

CONCLUSIONS

- It is finally conclude that after implementing this design and fabricating the equipment it provides a provision for variation in length between the planes and the radius also can be varied.
- The set-up is designed for higher reliability, lesser maintenance and safe operation. Hence it can be used as laboratory equipment in the 7^{th} semester design lab.
- Errors caused from old type of test rigs are avoided to some limits.
- Eccentric effects caused during dynamic balancing will be reduced.

CHAPTER-7

SCOPE OF FUTURE WORK

- It is possible to increase number of spring in order to give better damping effect.
- It is possible to replace mild steel disc by aluminium disc in order to decrease the weight.
- It is possible to provide digital angle measurement by mounting digital angle measurement indicator.
- It is possible to use non-conventional tachometer instead of proximity sensor.

APPENDIX-I

A1: Disc (Front view)

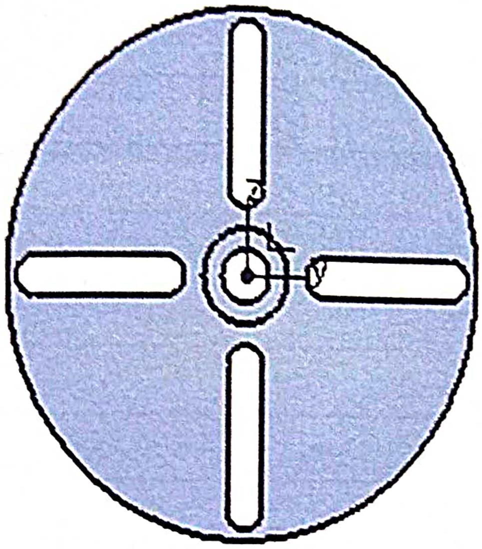


Fig.A1. Disc

A2: Disc (side view)

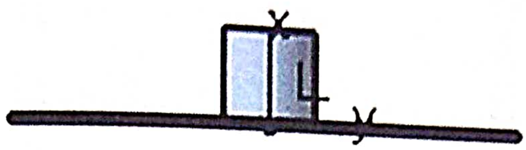


Fig.A2.Disc

A3: Inner Frame

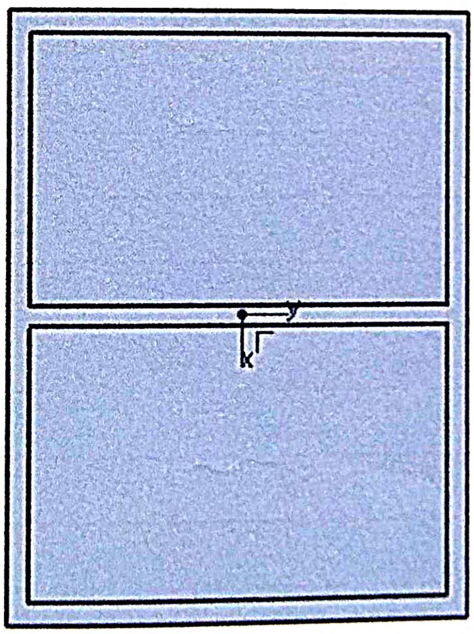


Fig.A3.Inner frame

A4: Outer frame

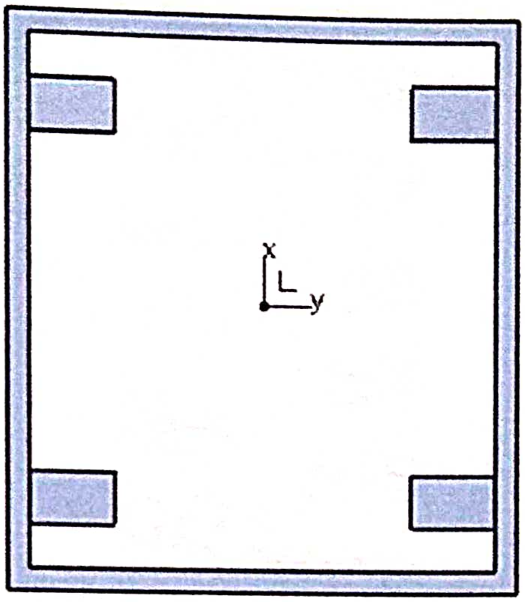


Fig.A4.Outer frame

A5: Spring

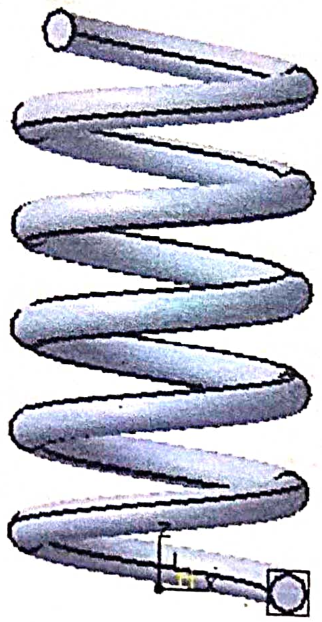


Fig.A5. Spring

A6: Shaft

Fig.A6. Shaft

Fig.A7: photograph of the complete fabricated model

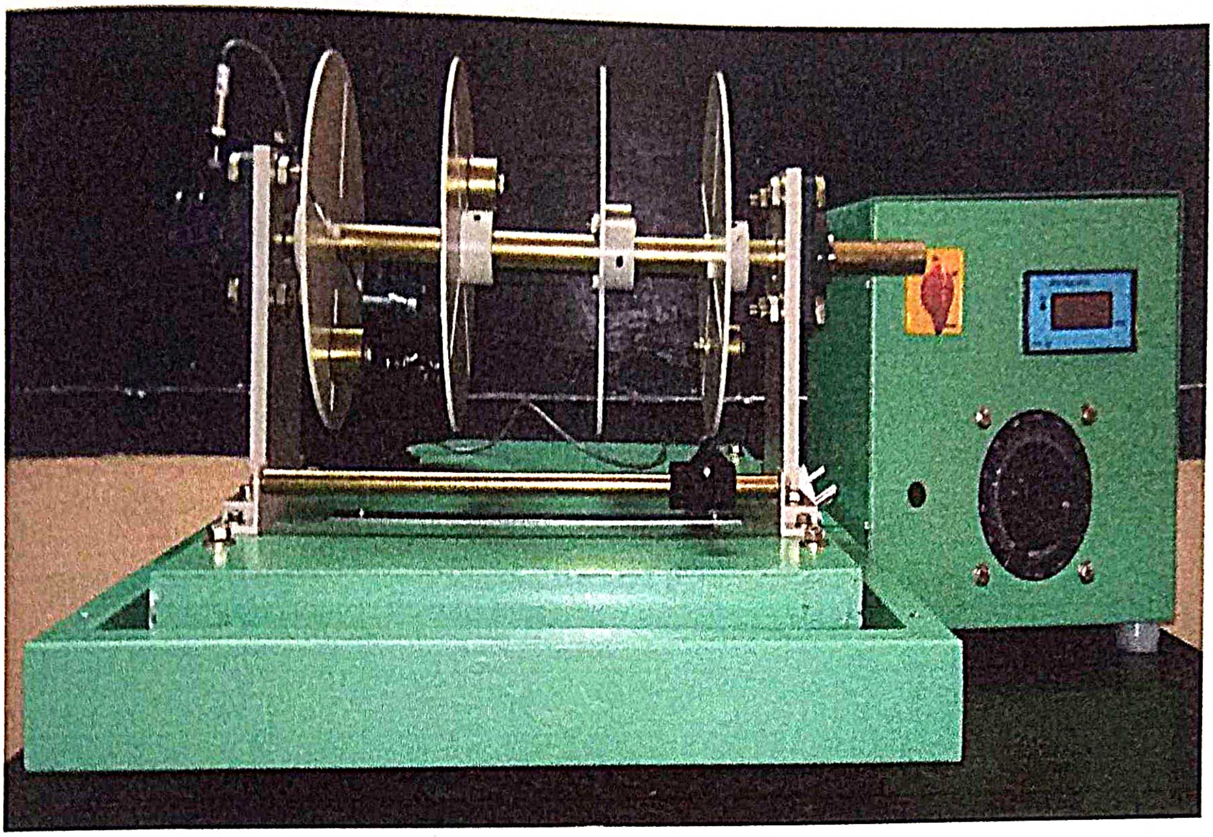


Fig A7.Fabricated model

APPENDIX-II

Case 1;

Balancing of Single Rotating Mass by a Single Mass Rotating in the same Plane.

1. **The mass M_1 of 100 gm revolves in a plane with a radii of 60mm. Angular position of the mass M_1 is 70°. Determine the angular position, radii and mass M_2 required to balance the system.**

Solution;

Given; M_1=100gm, r_1=60mm, θ_1=70°

M_2=? , r_2=? , θ_2=?

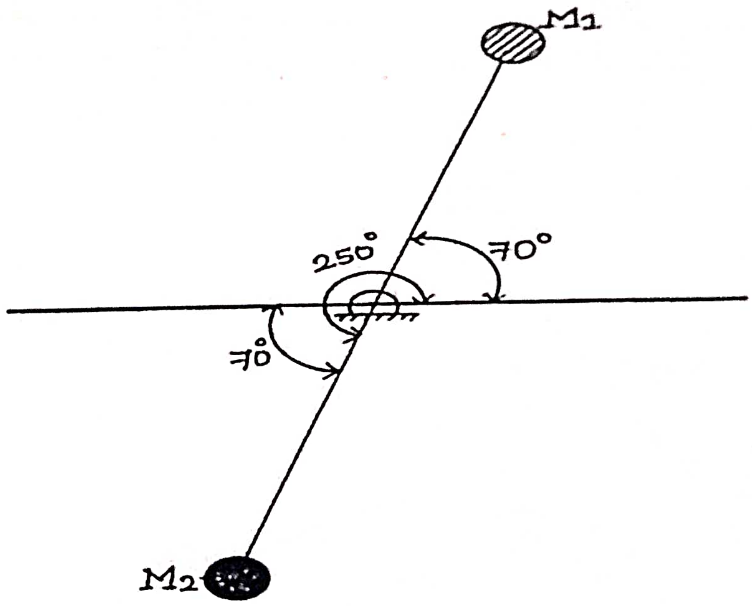


Fig (a): Balancing of single rotating mass by a single mass rotating in the same plane

Centrifugal force due to mass $M_1 = M_1\omega^2 r_1$(at an angle 40^0)

Centrifugal force due to mass $M_2 = M_2\omega^2 r_2$(at an angle 220^0)

For perfect balancing $M_1\omega^2 r_1 = M_2\omega^2 r_2$

i.e., $M_1 r_1 = M_2 r_2$

100*0.06=100*0.06

Case 2;

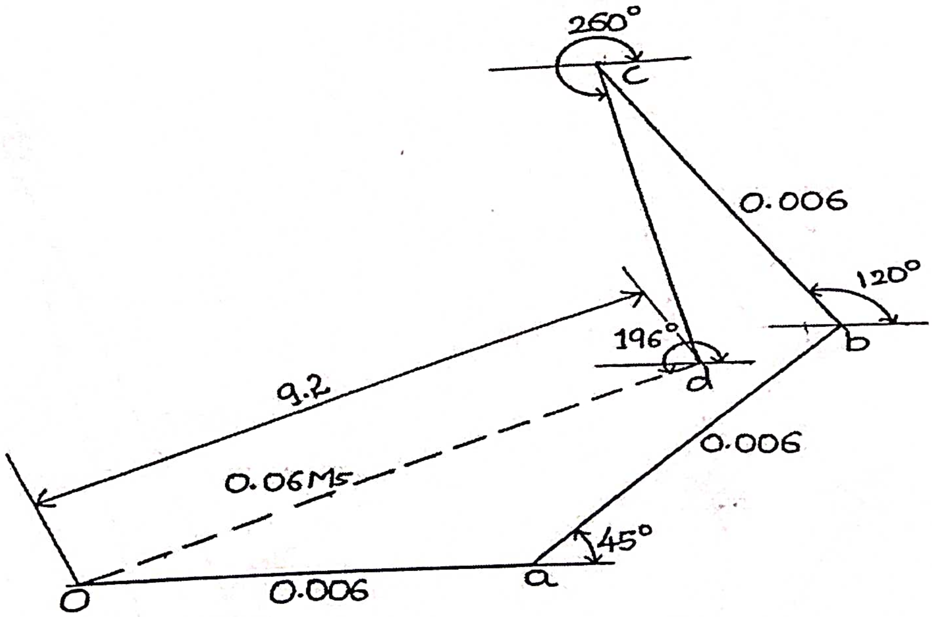
Balancing of Several Masses Rotating in the Same Plane.

1. **Four masses M_1, M_2 and M_3, M_4 with 0.1kg respectively revolve in the same plane with radii 60mm. Angular position of masses M_2 M_3 and M_4 are 45^0, 120^0 and 260^0 from M_1. Determine the position and magnitude of mass M_5 at radius 60mm to balance the system.**

Solution:-

Tabular column 1:

Mass, 'M' (kg)	Radius, 'r'(m)	Force $\div\omega^2$ Kg-m
M1=0.1	0.06	0.006
M2=0.1	0.06	0.006
M3=0.1	0.06	0.006
M4=0.1	0.06	0.006
M5=?	0.06	0.06* M5



Force polygon

od*scale=0.06*M_5

9.2/1000=0.06*M_5

M_5 =0.153kg, θ =196°

Case 3;

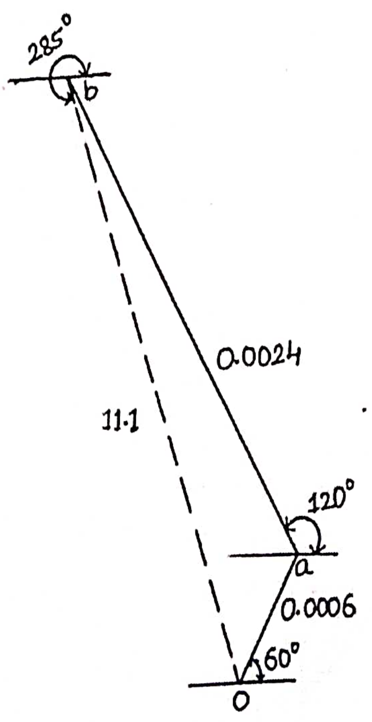
Balancing of several Masses Rotating in the Several Plane

1. **A shaft carries 4 masses M_1 M_2 M_3 M_4, where M_2=0.1kg, M_3=0.2kg, the masses revolve at radii 60mm. The distance from the plane M_1 are 0.1mm, 0.2mm, 0.3mm respectively. The relative angular positions of the mass M_2 M_3 are 60^0 and120^0 respectively. Determine the position and magnitude of unbalanced masses M_1 and M_4 revolve at radii 60mm.**

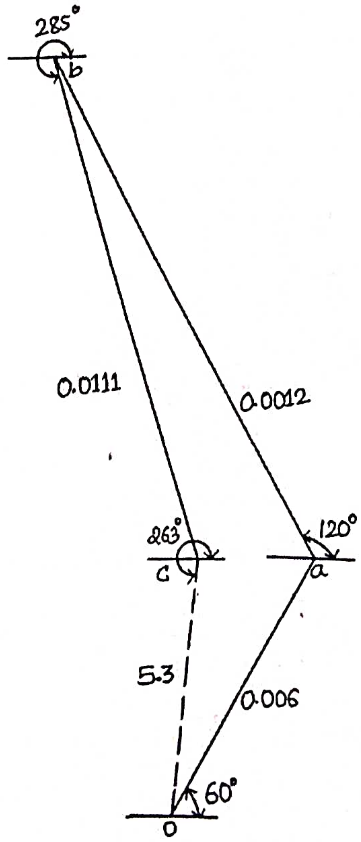
Solution:

Tabular column

Plane	**Mass 'M' (kg)**	**Radius 'r' (m)**	**Force÷ω^2 Kg-m**	**Distance From reference plane 'L'**	**couple÷ω^2**
M_1	?	0.06	0.06*M_1	0	0
M_2	0.1	0.06	0.006	0.1	0.0006
M_3	0.2	0.06	0.012	0.2	0.0024
M_4	?	0.06	0.06*M_4	0.3	0.018



Couple polygon



Force polygon

Ans;

From couple polygon;

Ob*scale=0.06*M_4

11.1/1000=0.06*M_4

M_4=0.185kg

θ =285°

From force polygon;

oc*scale=0.06*M_1

5.3/1000=0.06*M_1

M_1=0.088kg

θ =263°

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