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Design and Development of Dynamic Balancing Machine

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Abstract:-*This work is an attempt to evolve a means for statically and dynamical balancing of mechanical components. This will no doubt evolve greater success in ensuring sound fabrication of our local machine and ensuring that life span of such machine is prolonged since vibration will be reduced to the barest minimum.*
Every wheel and its shaft have to be in a state of balanced, two mating shaft in rotation have to be in a state of static and dynamic balance. This is because unbalance condition produce centrifugal force which increase as the speed of the machine increase, causing damage to machine parts.
To achieve the aim of static and dynamic balancing, various design alternatives for achieving the design solution were synthesized and a choice of economic method which will satisfy the objective was made. Based on the principle guiding the performance of the machine, the dimension and size of the various components were established and correctly selected and the basic static and dynamic apparatus was fabricated and after testing it was found to satisfy the objective.
Keywords: *dynamic, Static, Rotors*

I. Introduction

Power transmission in engineering is accomplished by the use of shafts bearings, pulleys, belts, rollers, gears, couplings, etc. Most machine members are linked or interconnected with one another before this power transmission can be effective.

Misalignment or out of balance of shafts or any of these machine members results in excessive vibration which is transmitted to other members and down to the foundation bolt and therefore in fatigue stress on the members. Fatigue, probably, is responsible for over 70% of failure of engineering component in operation. Misalignment in shafts and bearing causes excessive load. High speed bearing of aircrafts that may operate at over 20,000 rpm may become catastrophic if there be any unbalance of the members. Out of balance of machine members creates serious problems on the functionality and durability and machine members and on the machine in general.

Since rotating machinery is omni present right from household machinery, automotives, marines to space applications and hence understanding dynamic behavior of rotating machinery is crucial for long life of machinery and safety of humans. Rotor dynamics deals with these aspects and hence its study is very important for design engineers and field engineers. As compared to structural vibrations the rotor dynamics differs in several ways as rotating machineries have inherent forces and moments due to dynamics of various machine elements or faults occur in them also gyroscopic effect which is predominant at higher speeds makes the natural frequency speed dependent. Bearings and seals also make natural frequency of the rotor system speed dependent, moreover, it also makes system unstable. An asymmetry in rotors due to operational requirements such as keyways or slots in rotors causes the rotor instability. The internal damping hysteric and friction between two mating parts in rotors makes the system unstable and there are several other reasons for the instability due to working fluid interaction with rotor components. Above mentioned reasons makes the rotor dynamics more challenging as compared to the structural dynamics. [2]

The most important and fundamental procedure to unfavorable vibrations is to eliminate geometric unbalance present in the rotor. The balancing procedure for rigid rotor was established relatively early. The arrival of high speed rotating machines made it necessary to develop balancing technique for flexible rotors. Two representative theories were proposed for flexible rotors. One was modal balancing method and the other was influence coefficient method. Vibration response measurements yield a great deal of information concerning any faults within rotating machines. Cracks in shaft have long been identified as factors limiting safe and reliable operation of turbo machines. They can sometimes result in catastrophic failure of equipment and more often, in costly process upsets, repairs and premature scrapping and replacement of equipment. It was pointed out that transverse vibrations are very common in rotor system due to residual unbalance, which is the most inherent fault in rotor. Self- excited vibrations are disturbances belong to a fundamentally different class as compared to free or forced vibrations. In self excited vibrations, the excitation force sustains the motion is created or controlled by motion itself, when the motions stops the excitation force disappears. On forced vibration sustaining excitation force exists independent of motion and persists even when the vibratory motion is stopped. An unbalanced disc mounted on flexible shaft running in two bearings executes an ordinary transverse forced vibration. [3]

Miklos T. Koncz explained in the paper entitled 'Design of low cost balancing machine for gas turbine of UAV' that Balancing of the rotor is one of the most important crucial key elements of a gas turbine construction. It is true for amateur, semi-professional and professional field as well. In this article the author presented a new method for balancing small-scale gas turbine. During the experimental phase his main aim was to implement a sophisticated balancing machine for homebuilders and semi-professionals without large investment, complicated electronics and mechanics. Every balancing machine consists of two parts: mechanics and electronics and/or software. The design concept of the mechanical part is to provide easy balancing of complete turbine. This in situ, soft suspension, balancing system (balancing in own support) helps to correct the unbalance problems coming from bearings and shaft misalignments (angular and parallel). The other advantage of this system was implementing the balancing instrumentation virtually without electronics. Only an electronic switch provides the selection between left and right signal of the sensors. All measurement functions are implemented by software with digital signal processing method, average workshop personal computer and low cost soundcard.

Generally speaking unbalance is one but not only source of the vibration of the rotating engine. Unbalance force generates vibration, but the resonance can gain it at a representative revolution speed. The resonance can be characterized by amplitude emphasis and 180° phase changing around resonance frequency. If resonance exists in the system the balancing decreases its effects, but only for temporary time, because small mechanical changing can cause large vibrated amplitude amplification. In this case the resonating part must be enforced (stronger spring force) to eliminate the resonance. The program and mechanics proved to be useable for balancing Phoenix Mk-4. There are some disadvantages of this balancing solution, but these can be corrected by enhancing the program and rebuilding the mechanics.[4]

Rajeev Chaturvedi and Shree Niwas Sahu explained in their paper entitled 'Design and analysis of vertical dynamic balancing machine flexure for satellite balancing' that Mass properties of the satellite refer to properties of a body like Mass, Centre of gravity (C.G.), moment of inertia and product of inertia. Chobotov (1991) in his book has explained the significance of Mass properties of satellite. Further the test shows how Mass properties of satellite are used during attitude and orbital control. In the life cycle of development of satellite it is essential to know and control these mass properties for desired performance. Satellite mass properties viz. Mass, C.G. are the deciding parameter in selection of launch vehicle and hence affect the overall budget of the project. Satellite mass properties are also vital parameter for the design of its interface on launch vehicle, thrusters specifications and there location. Accurate measurements of mass properties enhance the mission life of the project by effective utilization of propellants.

A satellite consists of various subsystems and payloads having unsymmetrical shape that create complicacy in theoretical assessment of mass properties, also satellite harness, thermal elements viz. multilayer insulation, heater etc contribute to mass properties. Owing to irregular shape it is not possible to get exact theoretical assessment of mass properties of the satellite and therefore arises the need to measure mass properties of fully configured satellite. Mass properties of satellite are measured in-house in ISRO satellite centre, Bangalore. Measurement of static and dynamic unbalance present in satellite is done on in-house developed Vertical dynamic balancing machine (VDBM). Vertical dynamic balancing machine (VDBM) has key components comprising of Hydrostatic bearing, DC drive, Flexures, Velocity pick up transducer and Data acquisition System. Fig-1 shows the satellite balancing on VDBM. The Spacecraft to be balanced is mounted on the VDBM using suitable interface fixture. When the spacecraft is spun about its axis, the static and dynamic unbalance present in the spacecraft causes oscillatory motion of the static and dynamic flexures which are simple harmonic in nature. These harmonic motions of the static and dynamic flexures are sensed by the respective sensors and they proportionately convert the mechanical input of harmonic motion into electrical voltage output. These readings are used to compute the actual static and dynamic unbalance present in the satellite with the help of calibration constants This paper discusses the design of flexure for measurement of static unbalance (centre of gravity) and dynamic unbalance (Product of inertia) in the body. Liberalized displacement relations are derived to measure the force and moment through the flexure. Analytical studies based on kinematic modeling and computational modeling, as well as experimental study was conducted to investigate working of the flexure. Experimental and computational analysis result was found to be in close match with analytical result.

Static flexure is basically a parallelogram mechanism which is designed to produce a linear translation motion on the application of centrifugal load F due to static unbalance. It consists of set of fixed guided beam connected in parallel with mechanical amplifier. The mechanical amplifier amplifies the harmonic motion of static flexure and improves the measurement accuracy. Dynamic flexure is designed to measure the dynamic unbalance in satellite. It consists of two Tie rod separated by distance W and a flexure pivot. During spinning the forces generates moment about the flexure pivot which causes the harmonic oscillation about the flexure pivot.Fig-6 shows the schematic of the dynamic flexure. For small oscillation θ about the flexure pivot. Strain δy /LD in tie rod related to axial load P in the tie rod as P /δy □ AE/LD where A is are of cross section of tie rod , LD is length and E is young Modulus.[5]

Jim Lyons explained in paper entitled ‘Dynamic Balancing- Causes, Corrections and Consequences’ that when man invented he very quickly if it wasn’t completely round and it didn’t rotate evenly its central axis, then he had a problem. The wheel would vibrate, causing damage to itself and it’s support mechanism and in severe cases, be unusable. As the task of manufacturing mechanism and in severe cases, be unusable. As the task of manufacturing to minimize the problem. Research showed that the wheel and its shaft had be in a state of balance, i.e. the mass had to be evenly distributed about the rotating centerline so that the resultant vibration was at a minimum. This had to be achieved during the manufacturing process (and perhaps just as importantly, as wear occurred) so that maximum service life could be achieved from the system. Modern man still suffers from the same problem – only now the problem is amplified. As machines get bigger and go faster, the effect of the unbalance is much more severe. Many causes are listed as contributing to an unbalance 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. Whilst some corrections for eccentricity can be counteracted by balancing, it is a compromise. Dynamic balancing should not be a substitute for poor machining or other compromise manufacturing practices. In the manufacturing process, if proper care is taken to ensure that castings are sound and machining is concentric, then it follows that the two axis will coincide and the assembled rotor will be in a state of balance.[3]

A level of unbalance that is acceptable at a low speed is completely unacceptable at a higher speed. This is because the unbalance condition produces centrifugal force, which increases as the speed increases. In fact, the force formula shows that the force caused by unbalance increases by the square of the speed. If the speed is doubled, the force quadruples; if the speed is tripled the force increases by a factor of nine! It is the force that causes vibration of the bearings and surrounding structure. Prolonged exposure to the vibration results in damage and increased downtime of the machine. Vibration can also be transmitted to adjacent machinery, affecting their accuracy or performance. The key phrase being “rotating centerline” as opposed to “geometric centerline”. The rotating centerline being defined as the axis about which the rotor would rotate if not constrained by its bearings. The geometric centerline being the physical centerline of the rotor.[4]

When the two centerlines are coincident, then the rotor will be in a state of balance. The geometric centerline being the physical centerline of the rotor. When the two centerlines are coincident, then the rotor will be in a state of balance. Even inherent rotor design criteria that cannot be avoided. Many of these occur during manufacture, others during the operational life of the machine. When unbalance has been identified and quantified, the correction is straightforward. Weight has to be either added or removed from the rotating element. The ultimate aim being to reduce the uneven mass distribution so that the centrifugal forces and hence the vibrations induced in the supporting structures are at an acceptable level.

Everything that rotates needs to be in a state of balance to ensure smooth running when in operation. Precision balancing is essential to the manufacture of rotating equipment and to the repair and renovation of installed machines. As machine speeds increase, the effects of unbalance become more detrimental. Modern technology allows for accurate balancing to be performed both in the field and in the workshop. Increased time between outages and availability for production is the prime benefit.[6]

II. Problem Statement And Objectives

Power transmission devices mainly consist of rotating elements such as shafts, bearings, pulleys. If there is any misalignment in shaft or any unbalance in rotating elements or both of these causes excessive vibration which causes fatigue stresses in machine element which leads to failure of machine components in operation

The objective of this project is to design, construct and test a basic static and dynamic balancing apparatus as one of the ways to prevent premature failure of engineering components due to fatigue loading

WHAT IS BALANCING

Balancing is the processing of minimizing vibration, noise and bearing wear of rotating bodies as well as the adding or removal of material from non – rotating parts to change the centre of gravity and effect bending moments. The goal of balancing is to reduce vibration. The reduce vibration minimize noise, increase bearing life, decreases operating stress, consume less energy, increases product quality etc.

Balancing is accomplished by reducing the centrifugal forces by aligning the principal initial axis with the geometric axis of the rotation through the adding or removal of materials. [1]

III. dynamic Unbalance

This is also known as tow plane unbalance, it is the Victoria summation of force and couple unbalance. It is the most common condition found in virtually all rotors. The specification of unbalance is only complete if the serial location of the correction plane is known. Dynamic unbalance or two – plane unbalance specifies all the unbalance which exists in the work – piece. These types of unbalance can only be measured on a spring.[1]

OUT OF BALANCE MASSES IN COMMON TRANSVERSE PLANE

Consider masses …M1, M2, M3 and M4 arranged at radii r1, r2, r3 and r4 at angles θ1, θ2, θ3, and θ4 in a common plane. As the shaft rotate out angular speed ω, each mass will pull on the shaft by a force m1 w2r12, m2w222, m3w32r3, m4w2r4 and directed outward from the centre of the shaft to the centre of the respective masses. The resultant of the centrifugal force can be obtained either by calculation, resolution of forces or by force polygon. If the resultant is mw2r, th4e balancing force is – mw2r. Since w2 is common to all the forces, the computation can be carried out with m1r1, m2r2, m4r4.

By resolution of force

E4 $= m1r1\cos\theta1 + m2r2\cos\theta2 + m3r3\cos\theta3 + m4r4\cos\theta4$ …………….1 or

$\sum V$ $= m1r1\cos\theta1 + m2r2\cos\theta2 + m3r3\cos\theta3 + m4r4\cos\theta4$ …………….1

:. The magnitude of the resultant force RF $= \sqrt{(\sum H)^2 + (\sum V)^2}$

The angle which the resultant makes with the horizontal is

$$\theta = \tan^{-1}\left(\frac{\sum H}{\sum V}\right)$$

By polygon of forces, the magnitude and direction of the various forces are represented with a line, the balancing force will be the completing line. [3]

DESIGN CALCULATIONS

APPROXIMATE WEIGHT OF THE BALANCE

Considering the size of the machine, the dimension of the maximum balancing weights is chosen. The volume of the cylindrical balancing weight is

$= \pi h\,(\theta2 - r2)$……………………….2

$= \pi \times 0.094\,(0.03152 - 0.012)$

$= 0.295309\,(8.9225 \times 10\text{-}4)$

$= 2.634 \times 10\ m^3$

Since the density of steel = 7680 kg/m^3

Density $= \dfrac{\text{mass}}{\text{Vol.}}$ …………………………….3

:. Mass =density x vol.

$= 7680\ kg/m^3 \times 2.634 \times 10\text{-}4\ m^3$

=2.20 kg

2 kg weight =2 x 9.8 = 19.6N = 20N

FORCE ANALYSIS ON SHAFT

FC = centripetal force

FC=mω2r =

Where, m=total mass in rotation

ω=angular velocity of rotation

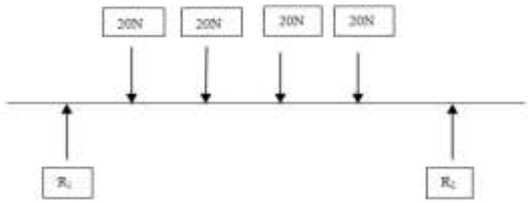

Fig 1 Force Analysis on shaft

For this design, the mass of the mass of the shaft and its content (pulley) is 5kg. The radius of the pulley of centre = 75mm =0.075m

Hence, FC = 5 x (274.925)2 x 0.075

=2.8, 343.9N

Based on this calculation, balancing weight of this design is taken to be the free body diagram of the balanced shaft with the forces acting on it is shown below. R1 and R2 are the bearing reactions while force of 20N due to the balanced weights at each end of the shaft at the respective distance.

For static equilibrium

$$\sum Fy = 0$$

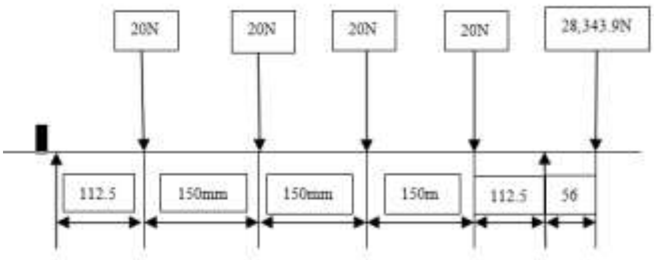

Fig. 2 Force Analysis of Shaft

:. R1 + R2 – 20 – 20 – 28, 343.9 – 0

R1 + R2=28, 383.9N

Taking moment about bearing (1)

Solving moment

Clock noise moment = anti clock noise moment

Clock noise moment = anti clock noise moment

20 x 60 + 28,343.9 x 150 $=R_2$ x 300 x 20 x 300N

R_2 x 200=28343.9 x 150 + 20 x 600 – 20 x 300

=4251585 + 12000 – 6000

R_2 =4257585 300

= 14191.95

BOT $R_1 + R_2$ =28383.9

:.R_1 =28383.9 – 14191.95

=14191.95

Shear Force (SF) Calculations

F_1= SF at 1 – 1 = - 20N (same through AB)

F_2= SF at 2 – 2= - 20N + 14191.95

= 14171.95N same for BC

F_3= SF at 3 – 3 = 14171.95 – 28343.9

= - 14171.95N (Same through CD)

F_4= SF at 4 – 4 = 14171.95 + 14191.95

= 20N (same through DE)

Bending Moment (BM) Calculations

BM at 1 – 1 =M1= - 20 x1

MA =M. z0 = 0

MB = (M1) x1=- 20 x 0.30=- 6Nm

BM at 2 – 2 = M2= - 20x2 + 14191.95 (x2 – 0.3)

MC = (M2) x 2= 0.6= - 20 x 0.45 + 14191.95 (0.45 – 0.30)

90 + 2128.7925 = 2119.7925NM

BM at 3 – 3 = m3

=-20 x 3 +14191.95 (x3 - 030)-28343.9 (x3 – 0.45)

$M_0 = (M_3)$ x3= 0.00= -20 x 0.6 + 14191.95 x 0.3 – 28343.9 x 0.15

= - 12 + 4257585 – 4251.585

=- 6NM

BM at 4 – 4 $=M_4$= -20x4 + 14191.95 x (x4 – 0.3) – 28343.9 (x4 – 0.45) + 14191.95 (x4 – 0.6)

= - 20 x 0.90 + 14191 x 0.6 – 28343.9 x (0.9 – 0.45) + 14191.95 (0.9 – 0.6)

= - 18 + 85146 – 12754.755 + 4257. 585

= 12772.755 – 12772.185

:. Mass bending moment = 2119.7925N

X SHAFT DESIGN[6]

Using maximum shear stress theory

The mass shear stress on the shaft is

E max =1/2(δ6)2 + 4c2

Substituting for δb and c

T max =1/2(32m)2+4 (16T)2
(πd3) (πd3)

$$= \frac{16\ M2 + T2}{\pi d^3}$$

T= 348022.08 Nm

Since T = 2.08 Nm, d = 0.02m

=2.08 x 16

=3.142 x 0.023

=1323997.454

d =3

3.2.6 Selection of Pulley [6]

Required r.p.m. = 1000

Motor r.p.m. = 1440

Pulley Ratio $= \frac{1440}{1000}$

= 1.44

As we have found pulley ratio, now we can select standard pulley and according to our pulley ratio we can select motor shaft pulley.

Shaft pulley diameter = 100 mm

Motor pulley diameter = 100 = 70 mm

Design of Key [6]

Material used for key is 40C8

Now,

For design to be safe it is required to take larger length. Therefore taking larger length 1.39 mm.

IV. Manufacturing

Rotors

Material used for. The circular grooves on rotors are for accommodation of balancing weights. These grooves are machined on vertical machining center.

Fig. 3 Rotor

Shaft

The shaft is machined from medium carbon steel (40C8) round bar. This is due to its strength, ability to withstand combined bending and torsional stress. There are slots are machined to accommodate key.

Fig.4 Shaft

PULLEY:

For the selection of pulley we first calculated pulley ratio and according to that we selected shaft pulley and motor pulley. Shaft pulley has diameter 100 mm and motor pulley has diameter 70 mm. Pulleys are made of cast iron because they are easily available and due to its strength.

Fig.5 Pulley

STRUCTURE:

The material used for the structural stand is mild steel angle iron 40m x 40mn. This is due to its strength and affordable.

Fig.6. Structure

V. Conclusion

Generally all power transmission devices consist of rotating parts have unbalance present in them. Due to this unbalance excessive vibration is produced and these vibrations are transmitted to structure and foundation which leads to failure of machine parts and collapse of foundation. With the use of dynamic Balancing Machine we are able to estimate the unbalance in rotating parts and can be removed with the help of balancing weights ensuring smooth running of machine.

Acknowledgements

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