VIBRATING TABLE FOR EQUIPMENT TESTING

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***Abstract*- This paper presents the modelling and control design of a vibratory table which could actually test the components by applying multiple of G-force using undamped frequency. The vibrating machine will test the components of an AFV, whether it would withstand the vibrational forces along with the G-force applied which is one of cause of vibration. The equipment of Armored fighting vehicle (AFV) which will undergo vibrational forces during working condition on the battle field are to be tested before installing them into the AFV. A mathematical model of the system is developed to evaluate the effect of each of the design parameters. The CATIA V5R19 software is used for cad modelling and validation of the vibrating machine. The vibratory table is designed systematically to optimize the performance of the system as well as improve its robustness.** **KEYWORDS – AFV, G-force, undamped vibration, frequency.**

I. INTRODUCTION

The vibration table unlike the common applications like paving concrete, it will be mainly used for checking and analyzing the equipments or components that are produced and supplied to the armed forces. This Vibration table will provide variable frequency keeping the amplitude constant which in turn will impart variable G-forces on the components due to changing vertical linear acceleration. The G-force test will define the components permissible limits and ensure whether the component is safe enough to be installed in the AFV. The components of the vibrating table consists of a table top which will be supported by four vertical spring which will give the table top one degree of freedom to move in vertically linear direction due to spring effect. This vibration will be caused by an unbalanced mass mounted on a shaft which will be driven by belt and pulley drive with the help of an AC motor.

II. LITERATURE REVIEW[1]

Thomas Bevan and Matthew Laurino - The currently used vibration table (VT3) has several disadvantages associated with it, which the new table (VT4) eliminates. Compared with the VT3, the VT4 has a more rigid tabletop and improved suspension system in order to vibrate the concrete within the wooden mold frames and maintain the integrity of concrete molds. Additionally, with the use of a single Damping Mass of 10 kg, the VT4 outputs the required amplitude of 0.3 to 0.4 mm and frequency of 3900±200 RPM to correctly consolidate concrete ranging from 2 to 40 kg. Furthermore, as desired, the VT4 uses the same motor and support frame as the VT3. See Figure 1 for a drawing of the final VT4 design.

III. DESIGN APPROACH

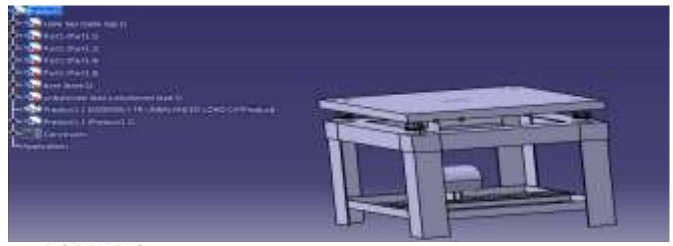

Figure 1 : Full assembly of vibratory table

A. TABLE TOP

The following section summarizes the analysis performed on the reinforced tabletop design that is common between the three four corner suspension type design options. The tabletop for our design has to be very rigid because the amplitudes of vibration are between 4.5 mm. Since the amplitudes are so small, the flex of table due to the forces from the eccentricity coupled with the weight of component and acceleration forces must be limited to less than 4.5 mm. This is to ensure that the tabletop does not vibrate only the center of the table. To accomplish this required rigidity and to minimize the cost we kept the table to thickness to 5 mm. Different clamping devices can be used to clamp the components such as Claw machine clamp GRS, Hold down table clamp BS, Hold down table clamp BSG.

Material selection and properties

The material selected for the table top should sustain high vibrational and a range of G- forces. The table top will undergo a uniformly distributed load that will cause bending effect if the material chosen is inappropriate. This table top will also undergo fatigue stress due to variation in frequency due to gradual change in angular velocity. The clamping system used to mount the components on the table top will produce scratches and indents on the surface of the table top resulting in faster wearing off of the table top and leading to crack propagation.

The material medium carbon steel being selected from various of material options available in market due to its good and cheaper availability in the market. The medium carbon steel also has the desirable properties as required for the tabletop. It came out to be highly resistive to the bending load due to its high bending stress capacity while designing the tabletop using the medium carbon steel.[5]

B. SPRING

The table below shows the dimensions and properties and a schematic representation of the spring designed which is suitable for the vibrating table in terms of bearing load and enough vibration for implying g-forces at different frequency.

Rates & Loads[2]

Spring Rate (or Spring constant), k: 7.4414 N/mm True

Maximum Load, True Fmax: 491.0182 N Maximum Load Considering Solid Height, Solid Height Fmax: 491.0182 N

Safe travel[2]

True Maximum Travel, True $Travel_{max}$: 65.984 mm

Physical Dimensions[2]

Diameter of spring wire, d: 5.000 mm
Outer diameter of spring, D_{outer}: 60.000 mm
Inner diameter of spring, D_{inner}: 50.000 mm
Mean diameter of spring, D_{mean}: 55.000 mm
Free length of spring, L_{free}: 120.000 mm
Number of active coils, na: 5.000
Number of total coils, n_T: 7.000
Solid height, L_{solid}: 35.000 mm
Type of ends: closed & ground
Spring index, C: 11.0000
Distance between coils, $C_{oilpitch}$: 22.0000 mm
Rise angle of coils: 7.2561 Degrees
Material Type
Material type: Hard Drawn MB ASTM A227

Weights & Measures[2]

Weight of one spring, M: 0.1906 Kgs
Weight per one thousand springs, M: 190.6352 Kgs
Length of wire required to make one spring, L_{wire}: 1,209.5132 mm

Stress Factors[2]

Material shear modulus, G: 79,236,434,108.5271 Pa
Maximum shear stress possible, tmax: 622,182,898.1357
Wahl correction factor, W: 1.1309

Possible Loads[2]

63.252 N @ 111.500 Loaded Height
253.009 N @ 86.000 Loaded Height
Max Load For This Spring: 491.01819281858 N

C. L-SECTION BEAM FOR BASE SUPPORT

L-section beam will be used for forming the base support of the vibrating table which will support the whole system. The springs will be sandwiched between the base support and the tabletop. This base support will act as a rigid support and stand still while supporting the whole system. It will be generally under the axial compressive stress due to the combined load of tabletop, equipment to be tested and springs. This component of the machine will also bear the fatigue load produced due to the vibration as it will be rigidly fixed and non-movable part.

It will generally consist of the L-section beam or angled beams of 100x100x5 mm standard dimension which will be welded altogether to produce a rigid frame a CAD model design in CATIA is shown below in the figure to illustrate the structure of the frame of the base support.

MATERIAL FOR BASE SUPPORT

The general material used for the manufacturing of the L-section beam is medium carbon steel similar to the tabletop which is C45 grade medium carbon steel.[5] Hence the property of the material of tabletop as well as base support is one and the same. This material is indeed chosen because of its good weldability property as it is the only machining operation performed to manufacture the frame of the base support i.e. by welding the L-section beams as shown in the above figure.

D. BEARING

SKF6410

Dimensions [5]

D= 50 mm D= 130 mm
B= 31 mm d_1=75.46 mm
D1= 104.25 mm $r_{1,2}$ min.= 2.1mm

Abutment dimensions

d_{amin}=64mm. D_{amax}=116mm
$r_{a\,max}$=. 2 mm

Calculation data

Basic dynamic load rating (C) 87.1 kN Basic static load rating (C_0) =52 kN Fatigue load limit (P_u) =2.2 kN Reference speed =12000 r/min Limiting speed =7500 r/min Calculation factor (k_r)= 0.035 Calculation factor (f_0)= 12.2

E. PULLEY AND BELT

PULLEY[5]

A PULLEY is a wheel on an axle or shaft that is designed to support movement and change of direction of a taut cable or belt, or transfer of power between the shaft and cable or belt. In the case of a pulley supported by a frame or shell which does not transfer power to a shaft, but is used to guide the cable or exert a force, the supporting shell is called a block, and the pulley may be called a sheave.

A pulley may have a groove or grooves between flanges around its circumference to locate the cable or belt. The drive element of a pulley system can be a rope, cable, belt, or chain.

Hero of Alexandria identified the pulley as one of six simple machines used to lift weights. Pulleys are assembled to form a block and tackle in order to provide mechanical advantage to apply large forces. Pulleys are also assembled as part of belt and chain drives in order to transmit power from one rotating shaft to another.

Dimension

Inner diameter = 50 mm

Outer diameter = 80 mm

Width = 50 mm

Type = Single race grooved pulley.

Material = AISI TYPE 304 Stainless Steel

BELT[4]

A belt is a loop of flexible material used to link two or more rotating shafts mechanically, most often parallel. Belts may be used as a source of motion, to transmit power efficiently or to track relative movement. Belts are looped over pulleys and may have a twist between the pulleys, and the shafts need not be parallel.

In a two pulley system, the belt can either drive the pulleys normally in one direction (the same if on parallel shafts), or the belt may be crossed, so that the direction of the driven shaft is reversed (the opposite direction to the driver if on parallel shafts). As a source of motion, a conveyor belt is one application where the belt is adapted to carry a load continuously between two points.

DIMENSION
Length of belt = 1.1 m
Width of belt = 25 mm
Material = Rubber belt with fibre enforced nylon cords impregnated in core

F. SHAFT

A drive shaft, driving shaft, propeller shaft (prop shaft), or Cardan shaft is a mechanical component for transmitting torque and rotation, usually used to connect other components of a drive train that cannot be connected directly because of distance or the need to allow for relative movement between them.

As torque carriers, drive shafts are subject to torsion and shear stress, equivalent to the difference between the input torque and the load. They must therefore be strong enough to bear the stress, whilst avoiding too much additional weight as that would in turn increase their inertia.

To allow for variations in the alignment and distance between the driving and driven components, drive shafts frequently incorporate one or more universal joints, jaw couplings, or rag joints, and sometimes a splined joint or prismatic joint.

CALCULATION OF SHAFT[2]

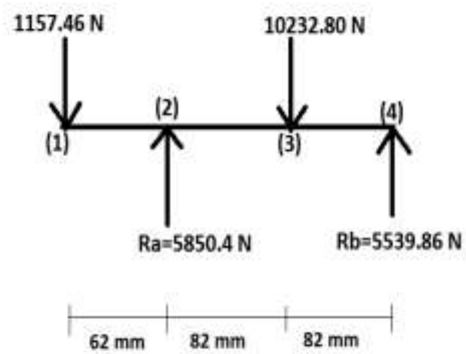

Figure 2 : Schematic representation of the shaft

Calculation of reaction forces,[5]

$\Sigma F_x=0$

$-1157.46+ R_a - 10232.80+R_b =0$

$R_a + R_b =11390.26$ …………(4)

$\Sigma M_{@2} = 0$

$(-1157.46*60) - (10232.80*82)+(R_b*164)=0$

Therefore R_b =5539.86N

Put R_b in equation (4)

Ra=5850.4N

Considering the maximum reactive force as the force applied on bearing .i.e. R_a

Therefore $F_r = R_a = 5850.4$N

Taking F_r= 6000N

Critical location on the shaft is (3) as it is highly loaded point on the shaft.

Case (1):

$[\sigma_b] =(M . y)/I$

Where, $[\sigma_b] = 630$ N/mm^2 from PSG 1.9

M = 10232.80*82=839089.6

Y= d/2

$I = (\pi * d^4)/64$

∴ d=23.25 mm

Taking d= 25 mm

Case (2):

$T_e=(\pi*[T]*d^3)/16$ ………(5)

Where, [T] = 80 N/mm^2 (assuming shaft with keyway)

P*1.5= (2.π.N.T)/60

∴T_e= 21.94 N-m

$T_e=21.94*10^3$ N-mm

Increasing torque by 50% for dynamic vibration.

$T_e=32913*10^3$ N-mm

Putting the above value in equation (5)

We get,

d = 12.79 mm

If considering just to be safe design so that it will always be on verge of getting fatigue crack,

Taking d=25 mm

But if we want shaft to be safe for longer time so that along with reversible loading on it fatigue crack which will appear on shaft will take considerably after many years of million revolution.

Therefore diameter of shaft is considered to be 50 mm.

G. UNBALANCED ROTATING DISC

Rotating unbalance is the uneven distribution of mass around an axis of rotation. A rotating mass, or rotor, is said to be out of balance when its center of mass (inertia axis) is out of alignment with the center of rotation (geometric axis). Unbalance causes a moment which gives the rotor a wobbling movement characteristic of vibration of rotating structures.[2]

CALCULATION–

Mass that we desire so that we get desired amplitude of vibration is 15 kg.

i.e. m = 15 kg.

material selected = AISI type 304 Stainless Steel.[5]

Density = 8030 kg/m^3.[5]

We know,

Density = mass/ volume

We know mass and density and volume will be $(3.14 * d^2 * b) / 4$

Where, d= diameter of disc

b = width of rotating mass

considering diameter of disc = 16 cm . (as per CAD design)

therefore,

8030 = 15 / (0.02010 * b)

Hence, b= 9.4 cm.

Hence final dimension,

Diameter of disc = 16 cm.

Eccentricity = 3mm.

This disc will be covered in a casing of MILD STEEL grade C45 of diameter.

IV. ANALYTICAL APPROACH

Vibration may be desirable or undesirable in situations according to the condition.. For e.g. a rotating fan, Engine of automobile Vibration is undesirable but for a digger machine or vibrating table Vibration is desirable. In our project Vibration is desirable as equipment's are to be checked under some specific frequency of Vibration. In vibrating table Vibration is produced with rotation of unbalanced disc mounted on shaft with some minor eccentricity. This eccentricity gives rise to centrifugal force of magnitude mew^2

Where m= mass of the disc (kg)

e = Eccentricity of the disc mounted on driven shaft in mm.

w = angular velocity in radian/sec

Taking references from book of mechanical Vibration by SIR. GROVER published under McGraw hill publication we know,

$(XM / me) = (r^2 /r^2 - 1)$[7]

Where X= amplitude of Vibration.

M=mass of whole body.

m= mass of unbalanced disc.

e= eccentricity.

r= ratio of w to w_n .

w= angular velocity (r/s).

w_n= natural frequency of body.

Here numerical values of damping effect are neglected as we don't have dampers inculcated in system and only damping resistance offered to Vibration is of surrounding air which is considered as zero as is it negligibly small.

sr.no	Angular velocity rpm(1*10^2)	ratio of w/wn	Amplitude of vibration (mm)	accelaration m/s^2	G force
1	3	1.82	4.5	4.44	0.45
2	4	2.428	4.5	7.85	0.8
3	5	3.03	4.5	12.33	1.25
4	6	3.64	4.5	17.76	1.81
5	7	4.25	4.5	24.18	2.46
6	8	4.86	4.5	31.59	3.22
7	9	5.46	4.5	39.97	4.07
8	10	6.07	4.5	49.34	5.029
9	12	7.28	4.5	71.06	7.24
10	14	8.49	4.5	96.72	9.85
11	14.4	8.74	4.5	102.32	10.43

Table which shows G-force impressed on equipment at approximate range of speed.

G-Force	Speed Range (RPM)
1	400 – 500
2	600 – 700
3	700 – 800
4	900
5	1000
7	1100 – 1200
8	1200 – 1300
9	1300 - 1400
10	1400 – 1440

Following are the outputs from the calculations in terms of graph-

Graph 1- Ratio v/s Amplitude

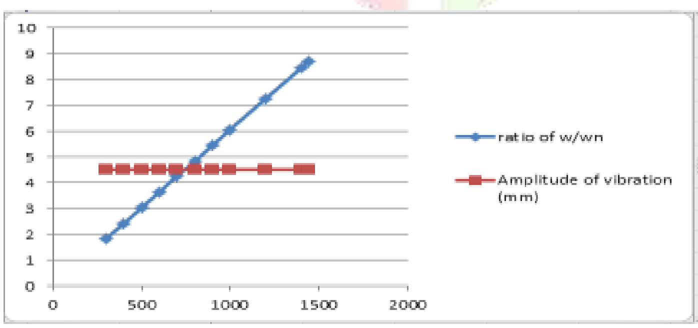

Figure 3 : Graph Of Amplitude And Ratio OF w/w$_n$

Graph 2 - Integrated graph with speed, amplitude, acceleration, G-force

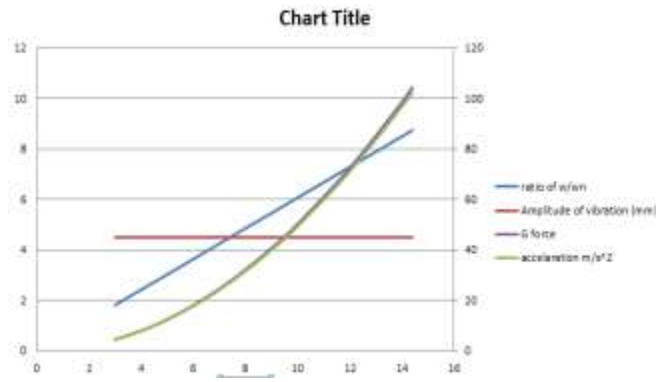

Figure 4 : Overall G-force Analysis Graph

V. *APPLICATIONS*

Vibrating Tables are custom engineered to fit your specific application requirements. Our tables are available with several different drive options, depending on the power source available and the application requirements. Each table is designed with a vibration isolation system that allows the energy input to the material to be maximized while minimizing the vibration transmitted to the frame and supporting foundation. Our effort is to seamlessly integrate custom vibratory tables into existing processes. Vibrating tables have been successfully utilized in a diverse set of applications through a wide range of industries including Densification of bulk materials, Unloading Materials, Shakeout Tables, Compaction Table. But unlike the above mentioned application our intentions are to use this sophisticated vibrating table to test the components manufactured by particular company which will check whether the components pass the safety criteria under vibration analysis as well as G-force analysis at different frequency. The vibrating table is used in many industrial domains, including Aerospace, Aggregate, Concrete, Chemical, Consumer products, Food processing, Plastic, Steel.

VI. *FUTURE WORK*

The product is designed by considering the financial constraint as per the consumer class but the quality and life of the machine can be improved by taking material that can withstand corrosion and indents caused by sharp edges more effectively than the current material used for table-top i.e. C-45 grade steel. To make the product more affordable Solid rubber mounts in compression were less widely used, as they did not allow as much movement as the other two. However they are generally the least expensive, and may be used if low amplitude vibrations are desired.

VII. *CONCLUSION*

This paper presents the modelling and control design of a vibratory table. The "G-Force" is identified as the key element for testing the components using undamped frequency. The vibrating machine will test the components of an AFV, whether it would withstand the vibrational forces along with the G-force applied which is one of cause of vibration. The equipment of Armored fighting vehicle (AFV) which will undergo vibrational forces during working condition on the battle field are to be tested before installing them into the AFV. A mathematical model of the system is developed to evaluate the effect of each of the design parameters. The CATIA V5R19 software is used for cad modelling and validation of the vibrating machine. The vibratory table is designed systematically to optimize the performance of the system as well as improve its robustness.

VIII. *APPENDIX*

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