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Design Development and analysis of portable waist belt mounted flexible shaft multi-function power tool.

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ABSTRACT

In case of conventional portable drilling machine while working on ceiling or roofs standing in up-right position with power tool in both the hands, the operator has to balance himself while performing the operation. This awkward position of working leads to cramps, back ache, discomfort leading to fatigue and health disorders. Also for performing different operations like drilling, hole sawing, Jig sawing we need separate machines. This portable waist belt mounted power tool is a special purpose machine in which weight of tooling is reduced so that it can be operated with single hand, while other hand holds on to the support. Also it can perform drilling, hole sawing, jig sawing operations. In this machine there is a waist belt on which motor is mounted which fits on operator's waist. There is a flexible shaft which can transmit torque from motor up to the tooling handle. In the tooling handle there is a small gear box for torque amplification and further power is given to the tool for performing different operations. As the motor is mounted on waist belt, there is a need of isolating the vibrations from operator's body. So to isolate vibrations there is elliptical leaf spring mounted between motor body and belt. There is also a hydraulic viscous fluid damper at the spinal cord location of mounting to isolate and eliminate any machine vibrations reaching the body

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I. INTRODUCTION

a a. Drill bits are cutting tools used to create cylindrical holes. Bits are held in a tool called a chuck, which rotates them and provides torque and axial force to create a hole. Drilling machines are used in variety of applications for drilling operation. These machines are hand held in which vibrations transmitted to workers' hands and arms. Hand-arm vibration can cause a range of conditions collectively known as hand-arm vibration syndrome (HAVS), as well as specific diseases such as white finger or Raynaud's syndrome, carpel tunnel syndrome and tendinitis. Vibration syndrome has adverse circulatory and neural effects in the fingers.

Work with hand-held power tools can be found in most industries all over the world. This type of work exposes the operators to different kind of loads like gripping-forces, feed-forces, exposure to vibration and noise, holding hot or cold surfaces and the exposure to dust. Designing a power tool with good ergonomics is a matter of finding the best compromise. As a simple example, increasing the mass is not acceptable because it will increase the forces needed to handle the tool. At the same time increased mass will in most cases reduce the vibrations.

Tools for industrial use must be of very robust design to withstand the vibrations they are exposed to. Industrial tools are therefore normally designed with the main parts made of metal .From a vibration point of view this means that most of the tools can be treated as rigid bodies, especially because the dominating frequency normally is equal to the rotational frequency of the tool spindle or the blow frequency for a percussive tool. There are several examples of weak suspensions designed to reduce vibration transmitted to the hands of the operator. There are also examples of designs where the handles just happened to be non-rigidly connected and in some cases even in resonance within the frequency region of interest. Oscillating forces act on the tool and the result is vibration.

Elliptical leaf spring is the type of vibration and shock isolator that was designed specifically for mobile applications. Their basic design employs two or higher tensile stainless steel "U" formed leaves, situated at each end, forming an elliptical shape when joined together in the center portion with face plates.

1.1 Problem Statement

In many applications the operator has to work on ceiling or roofs standing in up-right position with the power tool in hand, majority of the times the power tool needs to be supported by both hands, thus the operator has to balance himself while performing the operation, this awkward position of working further leads to cramps, back ache, discomfort leading to fatigue and health disorders. Thus there is a need for special purpose machine that address to the problem above by reducing the weight of the tooling so that it can be operated with single hand , while other hand holds on to the support, the machine be multi-functional i.e., should perform drilling , hole sawing ,jig sawing operation.

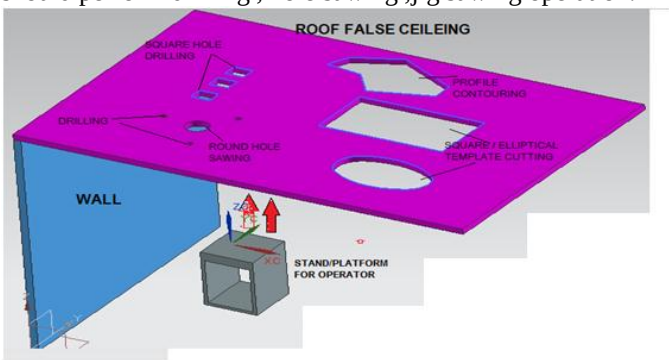


Fig.1 The false ceiling of plaster of Paris or plywood

1.2 Objective

- Design of flexible shaft drive and reduction gear box to perform multiple functions like drilling, hole sawing and jig sawing operations.
- Design of Elliptical leaf spring mounts and hydraulic viscous fluid damper to isolate and reduce the vibrations generated during cutting.
- Testing of the developed power tool cutter with and without the vibration reduction mechanism to determine the Overall damping coefficient for three operations namely drilling, hole-sawing and jig-sawing.

1.4 Scope

i. Increases operator comfort
ii. Prevents damages to hands, joints etc as minimum vibrations are transmistted
iii. Fewer vibrations lead to lesser audible noise.
iv. Simple system to implement.
v. Increases operator efficiency
vi. Increases dimensional accuracy

II. LITERATURE REVIEW

a NERC health & safety procedure [3] seeks to ensure the risks from exposure to vibration, whether to hands and arms or to the whole body, are adequately controlled. Where employees are likely to be exposed to vibration at or above the relevant exposure action or limit values, measures and adequate controls to ensure the risk of persons suffering harm from vibration is eliminated, minimized or adequately controlled must be implemented. Hand arm vibration (HAV) is a potential hazard for employees who work with hand held tools, hand guided machinery or feed work by hand to a machine where this exposes their hands and arms to high levels of vibration. Prolonged and regular exposure to excessive levels of HAV can affect the operator's health in particular causing Hand Arm Vibration Syndrome (HAVS), of which the best known condition is Vibration White Finger (VWF). Some persons who suffer from certain medical conditions such as diabetes, circulatory or nervous disorders are at increased risk of developing HAVS. An individual's health status and any medication must be taken into account when considering the adverse effects of HAV.
Exposure Limit Values and Exposure Action Values for Hand arm vibrations

- Daily exposure limit value (ELV) = 5 m/s^2
- Daily exposure action value (EAV) = 2.5 m/s^2

No-one within NERC may be exposed to HAV at or above the relevant Exposure Limit Values (ELV) as specified in the Control of Vibration at Work Regulations 2005.
Mirta Widia et.al [4] have conducted an experiment on effect of handheld vibrating equipment on human body. The aim of the study is to identify the effect of hand held vibrating tools on muscle activity and grip strength. The study was conducted on seven subjects (three male and four female). The experiments were performed with two kinds of exposure time, 5 and 15 minutes. Subjects were required to drill wood material using electric drill. Electromyography (EMG) and Vernier Labpro with 3 axis accelerometer used in the experiment.
The results showed that mean vibration level for electric drill was 10.53 m/s^2 for 15 minutes and 10.39 m/s^2 for 5 minutes duration. The most affected muscle by vibration factor was found to be the extensor Carpi radials muscle. Extensor Carpi radials is one of the muscle at the forearm. Muscle activity and grip strength increasing as the vibration level increasing.

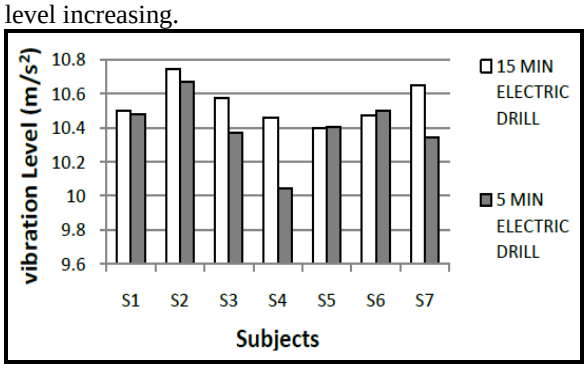


Fig.2 Vibration Level Drilling Wood Material [4]

Table .1 Vibration level effect on Grip strength [4]

Subjects	5 min electric drill			15 min electric drill		
	GS (Before)	GS (After)	Decrease	GS (Before)	GS (After)	Decrease
	(N)	(N)	Ratio	(N)	(N)	Ratio
S1	147.69	114.51	33.18	101.91	91.66	10.25
S2	121.50	52.01	69.50	119.32	36.73	82.59
S3	138.90	128.89	10.01	159.76	141.03	18.73
S4	72.36	34.65	37.71	75.24	20.78	54.46
S5	106.02	75.91	30.11	103.49	73.71	29.78
S6	90.01	58.49	31.52	103.65	59.78	43.88
S7	75.09	68.10	6.98	67.19	46.68	20.51
Mean	107.37	76.08	31.29	104.37	67.20	37.17

So from fig we conclude that vibration level increases as interval time of operating tool increases & from table it is shown that grip strength decreases as the as interval time of operating tool increases.

S. S. Rao [2], illustrated the basic features of a vibration measurement scheme. In this figure, the motion (or dynamic force) of the vibrating body is converted into an electrical signal by the vibration transducer or pickup. In general, a transducer is a device that transforms changes in mechanical quantities (such as displacement, velocity, acceleration, or force) into changes in electrical quantities (such as voltage or current). Since the output signal (voltage or current) of a transducer is too small to be recorded directly, a signal conversion instrument is used to amplify the signal to the required value. The output from the signal conversion instrument can be presented on a display unit for visual inspection, or recorded by a recording unit, or stored in a computer for later use. The data can then be analyzed to determine the desired vibration characteristics of the machine or structure.

Depending on the quantity measured, a vibration measuring instrument is called a vibrometer, a velocity meter, an accelerometer, a phase meter, or a frequency meter. If the instrument is designed to record the measured quantity, then the suffix meter is to be replaced by graph. In some application, we need to vibrate a machine or structure to find its resonance characteristics. For this, electrodynamic vibrators, electrohydraulic vibrators, and signal generators (oscillators) are used.

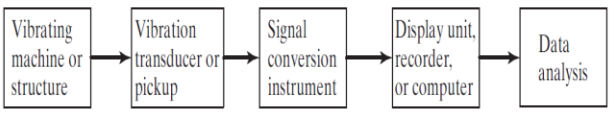


Fig. 3 Basic vibration measurement scheme [2].

M. A. Salim et al [1] Study determines the vibration occurs in handheld tools using Fast Fourier Transform and Operational Deflection Shape methods. The experimental results show the point, which occur higher vibration level. Fast Fourier Transform (FFT) and Operational Deflection Shape (ODS) experiments show that the vibration at the rear handle is not the same for each point. But between all points, there is one point same in both method, which shows the highest level of vibration. With Fast Fourier Transform, frequency responses of a structure can be computed from the measurement of given inputs and resultant responses. The equipment used are accelerometer, impact hammer with green tip, analysis software, PC data acquisition system, amplifier (4 channels), stabilize table, and rubber support.

Andrew K. Costain et.al[5] have given five methods of control vibrations. The generally accepted methods for vibration control of industrial equipment include; Force Reduction, Mass Addition, Tuning, Isolation and Damping.

i. Force Reduction of excitation inputs due to, for example, unbalances or misalignment will decrease the corresponding vibration response of the system.
ii. Mass Addition will reduce the effect (system response) of a constant excitation force.
iii. Tuning (changing) the natural frequency of a system or component will reduce or eliminate amplification due to resonance.
iv. Isolation rearranges the excitation forces to achieve some reduction or cancellation.
v. Damping is the conversion of mechanical energy (vibrations) into heat.

So we conclude that effectively controls vibration at or near resonance through energy dispersion, usually as heat. Three common types of damping forces are viscous, coulomb, and structural. Viscous damping forces are generated by masses moving through a fluid (e.g. dashpot or shock absorber). Coulomb damping forces are a result of sliding motion between two dry surfaces. Structural or material damping, the category most commonly applied for industrial vibration control, is caused by internal friction within a material. Open cell (polyurethane or butyl rubber) foams encompass inherent damping (and elastic) properties that make them suitable for shock and vibration control of systems with low frequency vibration, high deflection and large mass.

Rene Granado [8] studied the "reciprocating saw attachment for electric drill. A universal, quick connect, reciprocating saw attachment for electric drills is disclosed, designed as an adapter that converts a power rotary drill into a reciprocating saw. The present invention converts the rotary action of a drill into the necessary reciprocating action to power a saw blade. It is therefore an object of the present invention to provide an improved universal quick connect and reciprocating saw attachment for all electric drills that allows a power rotary drill to be used as a reciprocating saw. It is another object of the present invention to provide a device that permits a drill to be used as a reciprocating saw, thereby saving time, money and space. Also to provide a device that is universal in design, being capable of attachment to all models of drills. Another main object of the present invention to provide a device that is durable, long lasting, and requires little or no maintenance.

Rao V. Dukkipati & J. Srinivas [6] have given Forced vibration of damped system in their book Mechanical vibrations. We have adopted this theory due to our machine induces continuous forced vibrations while in working.

Steady state solution can be expressed as:

$$X_p = \frac{X_0}{\sqrt{(1-r^2)^2 + (2r\xi)^2}} \sin(\omega t - \phi)$$

Where,

X_0 – Static deflection $=\frac{F_0}{K}$,

r – frequency ratio $=\frac{\omega}{\omega_n}$,

$$\phi = \tan^{-1}\left(\frac{2r\xi}{1-r^2}\right)$$

$$\xi - \text{dapng ratio} = \frac{c}{c_c}$$

Also the magnification factor $\beta = \frac{1}{(1-r^2)^2 + (2r\xi)^2}$

The effect of the frequency ratio r, & damping factor ξ on magnification factor and the phase angle ϕ are shown in fig.2.2 & 2.3 From fig.2.2 we can say that the magnification decrease with increased damping. Also resonance the magnification factor β does not have the maximum value.

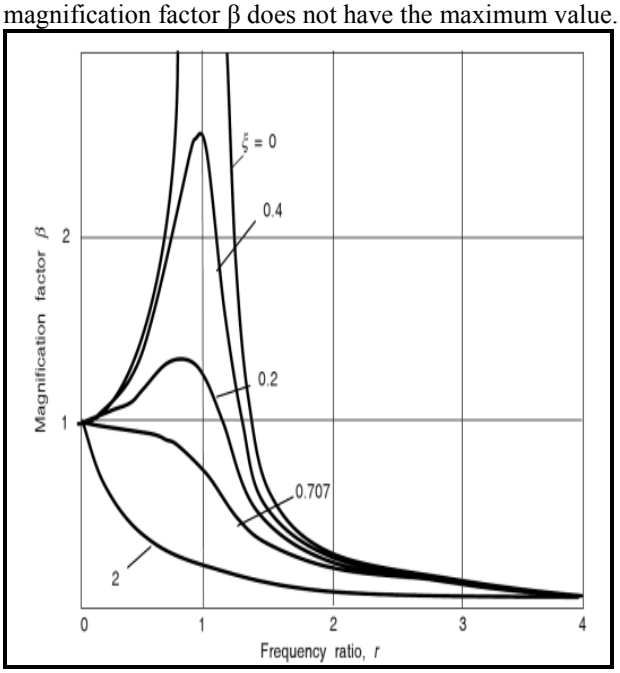


Fig. 4 Magnification factor [6]

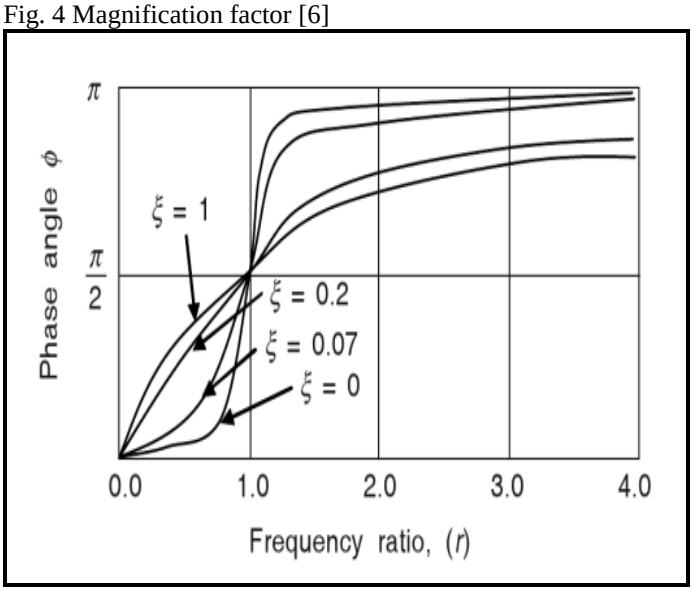


Fig. 5 Phase angle [6]

Philippe DUFLOT et.al & D. LEE et.al [7] have explained the introduction to the fluid damper, they have given overview of output force characteristics & they have given list of components we have to design while designing the damper. A damper can be globally defined as an element which can be added to a system to provide forces which are resistive to motion, thus providing a means of energy dissipation.

The most convenient and common functional output equation for a damper can be characterized as:

$$F = C \times V^{\alpha}$$

Where,

F is the output force,

V is the relative velocity across the damper,

C is the damping coefficient and

α is a constant exponent which is usually a value between 0.3 and 2.

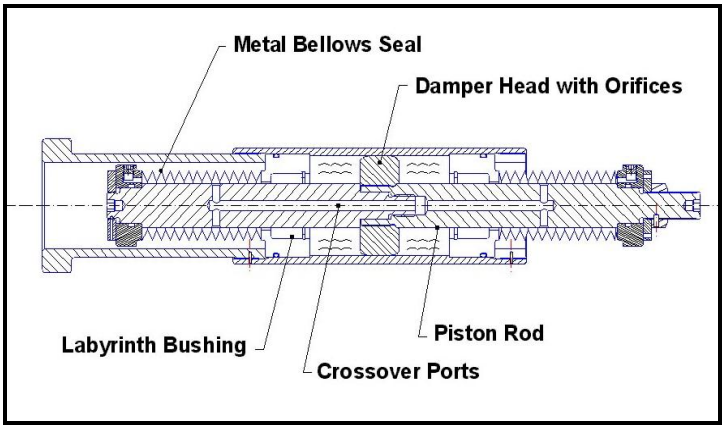


Fig. 6 Viscous Damper [7]

Fluid viscous dampers operate on the principle of fluid flow through orifices. A stainless steel piston travels through chambers that are filled with silicone oil. The silicone oil is inert, nonflammable, nontoxic and stable for extremely long periods of time. The pressure difference between the two chambers cause silicone oil to flow through an orifice in the piston head and input energy is transformed into heat, which dissipates into the atmosphere.

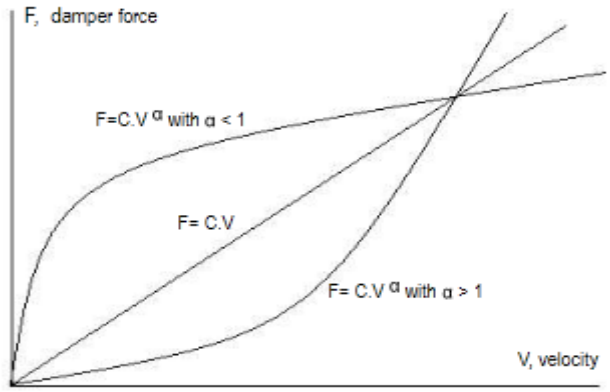


Fig. 7 Force against velocity for different exponent values [7]

From this paper we conclude that there is no spring force in this equation. Damper force varies only with velocity. For a given velocity the force will be the same at any point in the stroke. The essential design elements of a fluid damper are relatively few. However, the detailing of these elements varies greatly and can, in some cases, become both difficult and complex. So in case of design of damper we have to design the piston rod, piston head orifices, Cylinder, Seal, Piston head, Orifice etc.

J. Ehyaei et all [9] In this paper, dynamic stability and time responses are analyzed for a system of unbalanced flexible rotating shaft equipped with n automatic balancers, where the unbalanced masses are distributed in the length of the shaft. Also, the system lies on two linear elastic supports. This study adopts the Stodola–Green rotor model to consider the rigid-body rotations due to shaft flexibility instead of Jeffcott rotor model. The nonlinear equations of motion are derived for an autonomous system considering the ball-balancer of Stodola–Green rotor, utilizing the Lagrange's method, an equilibrium position and the linearized equations are obtained. Furthermore, the stability analysis is performed using the Routh–Hurwitz criteria.

Moreover, time responses are investigated for the nonlinear equations of motion using the generalized - α method. The study shows that for the angular velocities more than the first natural frequency and selecting the suitable values for the parameters of the automatic ball-balancers, which are in the stability region, the auto ball-balancers tend to improve the vibration behavior of the system, i.e., the partial balancing, but the complete balancing was achieved in a special case, where the imbalances are in the planes of the auto ball-balancers. Furthermore, it is shown that if the auto ball-balancers are closer to the unbalanced masses, a better vibration reduction is achieved. The fluid damping coefficient is one of the essential parameters to gain balancing.

E. Mahdi, et al [10] this study introduces a new composite semi-elliptical suspension spring by utilizing fiber reinforced composite strength in principal direction instead of shear direction. Three types of composites were tested, namely, carbon/epoxy, glass/epoxy and glass/carbon/epoxy. A comprehensive experimental investigation of composite semi-elliptical suspension springs has been carried out to find out typical behaviors of their compression, tension, torsion and cyclic tests. The results showed that the ellipticity ratio significantly affected the resilience energy absorption capability of composite elliptical tubes. For all load direction the fiber and laminate stacking sequences were designed to withstand any resulting shear stress by employing the cross ply (±45) laminates. The relaxation of the composite elliptic spring found to be very sensitive to the rate of compression. After 1.15 million fatigue cycles, composite semi elliptical suspension spring's useful stroke is reduced by only 2%. No hysteresis is observed under compression 50% of composite elliptical spring's useful stroke. The carbon-glass/epoxy elliptical springs exhibited higher spring rate but poor ride quality compared with the non-hybrid one.

III. DESIGN & ANALYSIS

a A. POWER REQUIREMENT IN DRILLING.

By calculations Power at the Motor is

N_{el} = 0.11 KW

Torque, T_s = 0.119 N.m

Thrust, T_h = 79.56 N

B. MOTOR SELECTION

Dongcheng- Trimmer

Single phase AC motor

Rated power input = 350 watt

Speed= 0-3000 rpm (variable)

Weight-1.8kg

Torque T = 1.11 Nm

C. DESIGN OF ELLIPTICAL LEAF SPRING

Material selected is spring steel EN 42J hardened & tempered

By calculations the dimensions are

Height, H = 80 mm

Length, L = 140 mm

Breadth, B = 30 mm

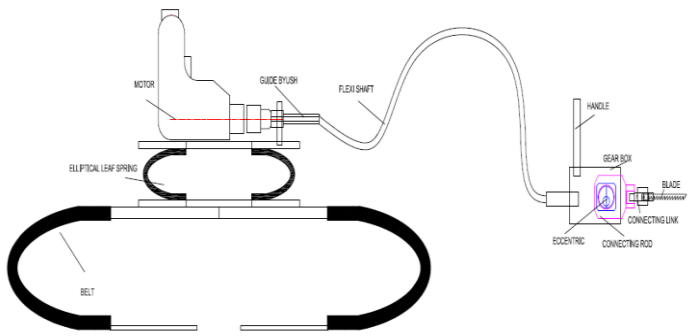


Fig. 8 Actual Model drawing

ANALYSIS OF ELLIPTICAL LEAF SPRING

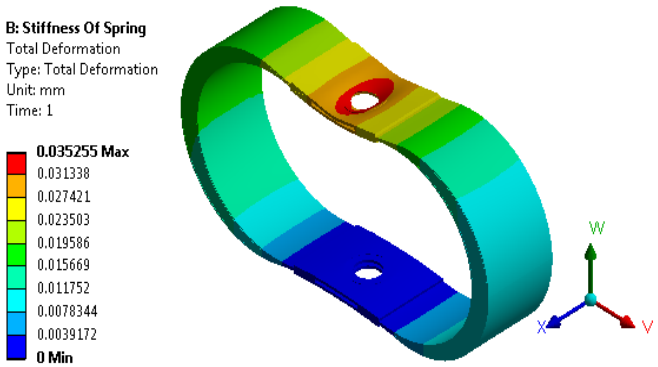
1. Stiffness of spring

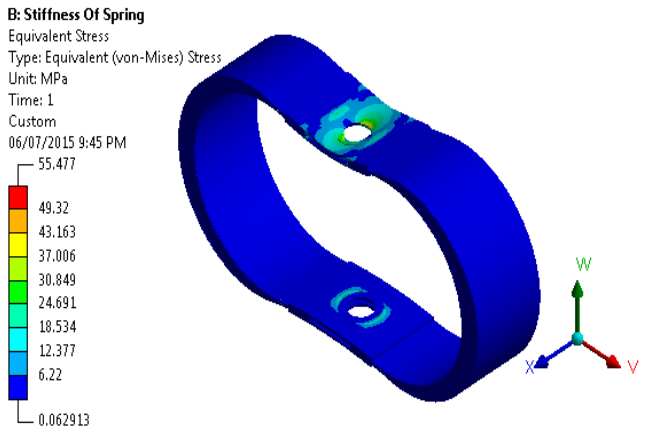
Geometry

Meshing

Boundary conditions

Post processing result

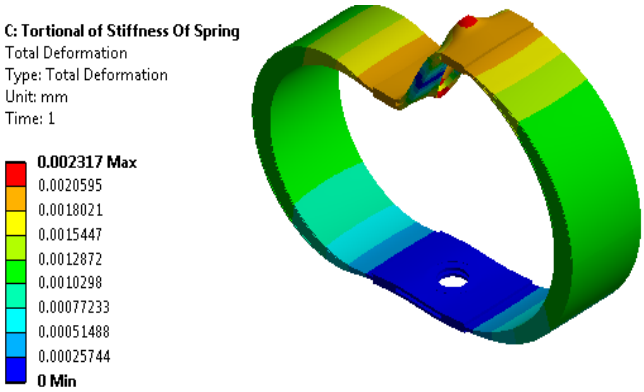




1. Torsional stiffness of spring

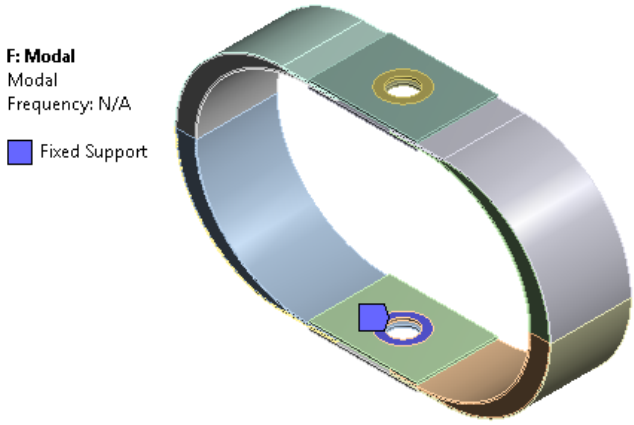
Boundary conditions

Post processing result

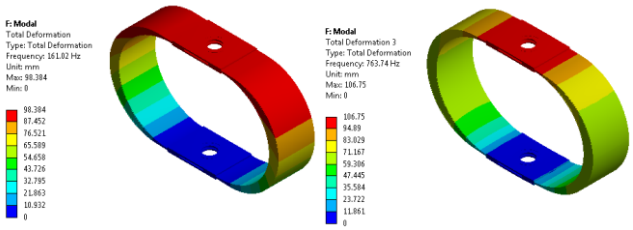


3. Modal Analysis

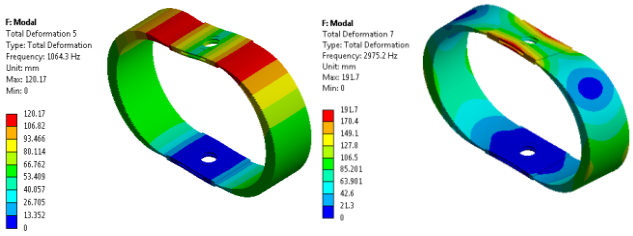
Boundary condition



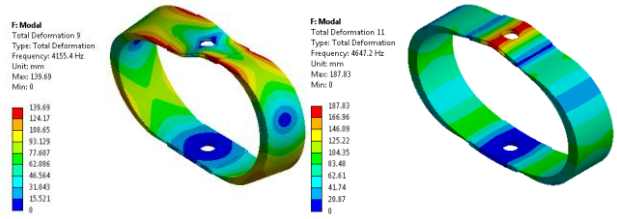
Ist and 2nd mode shape



3rd and 4th mode shape

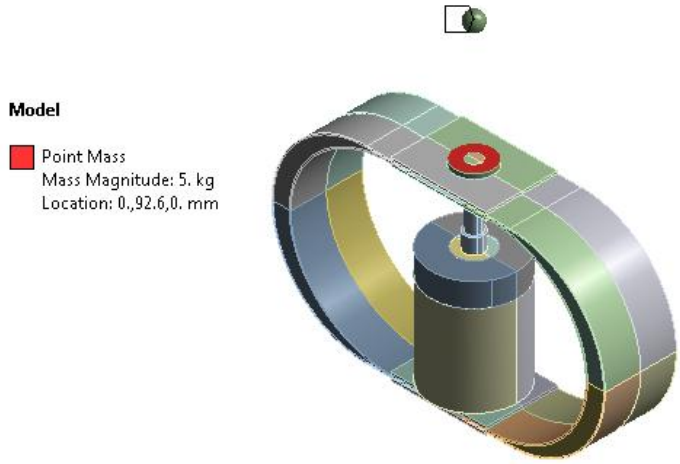


5th and 6th mode shape

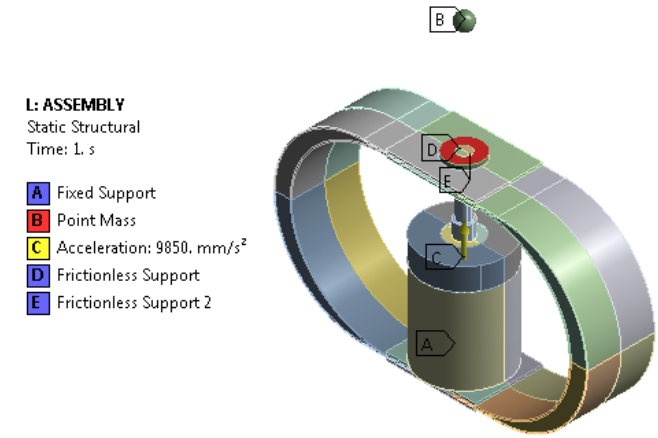


4. Structural analysis of Elliptical spring & damper

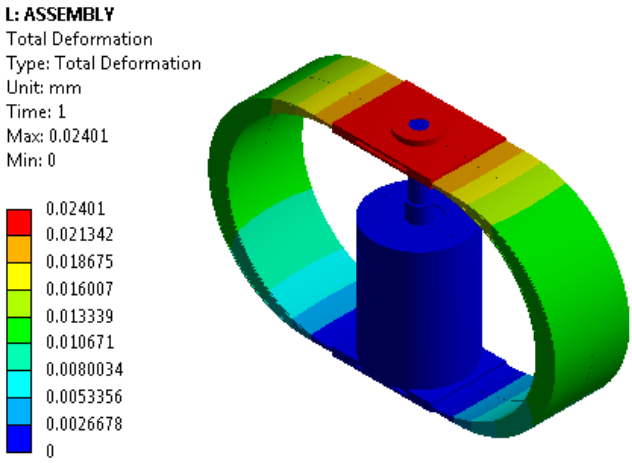
Geometry

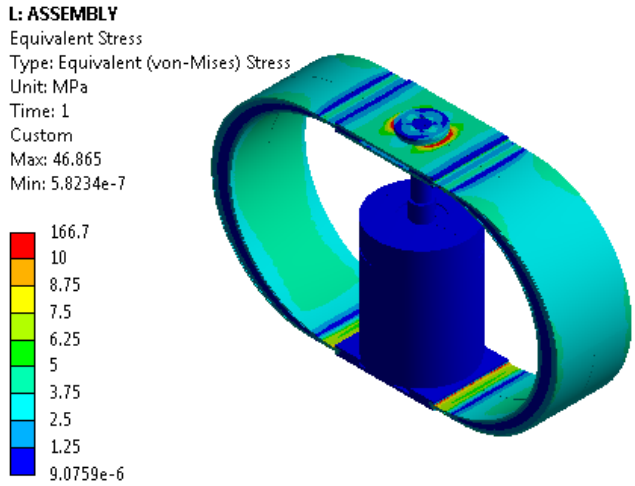


Boundary conditions



Post processing result





IV.CONCLUSION & RESULTS

aDue to adverse effect of hand arm vibrations we need to control those vibrations & we conclude that grip strength decreases as the as interval time of operating tool increases. The compressive stiffness of the elliptical leaf spring by theoretical calculations is 1410.41 N/mm and by FEA analysis it is 1460.24 N/mm. The natural frequencies at 1st to 6th mode shape respectively are 161.02 Hz, 763.74 Hz, 1064.3 Hz, 2975.2 Hz, 4155.4 Hz & 4647.2 Hz. As the maximum stresses in the component are much more less than yield strength of the material, hence components design is safe. Also we conclude that damping force doesn't depend on stiffness of spring. It is depends upon damping coefficient & velocity of piston

ACKNOWLEDGMENT

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