ISSN 2395-1621

Design, Development and Comparative Analysis of Passive & Active Fluid Damper for Wood Router to Reduce Hand-Arm Vibration

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ABSTRACT

Wood router machine is a high speed (hand held) wood working machine used to remove material from wood work-piece. Subsequent vibrations make it difficult to operate the machine for longer time and also tool consumption per unit cut has been found to be very high. While using the Router, we are facing the problem of Hand arm vibration. Hand-arm vibration (HAV) is vibration transmitted from a work processes into workers' hands and arms. It can be caused due to operating hand-held power tools, hand-guided equipment and by holding materials being processed by machines Multiple studies have shown that regular and frequent exposure to HAVs can lead to permanent adverse health effects, which are most likely to occur when contact with a vibrating tool or work process is a regular and significant part of a person's job. So the final conclusion will be drawn on the basis of theoretical and experimental results. Our main aim is to design router machine parts & damper parts, validation for strength calculations of critical components using suitable analysis software. Testing of the developed router will be with and without the passive-active fluid damper at same cutting speeds. The values will be determined using a Fast Fourier transform (FFT) analyzer on different component, thus we will do the comparative analysis of the performance of the router machine. So the final conclusion will be drawn on the basis of theoretical and experimental results.

Keywords- Hand-arm vibration(HAV), Fast Fourier transform(FFT), passive-active f luid damper.

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Before power routers existed, the hand tool form was frequently used, especially by pattern and staircase makers. The first hand-held power routers were invented in 1915 and Jet Motor Hand Routers put the name Onsruters. The name derived from a combination of the inventor's last name "Onsrud" and the term "router". The Onsruter married a router plane with an end-mill to create the first handheld power router. The idea for the Onsruter came when a rail road company decided they wanted to power the front light on a Steam Locomotive of the waste steam from the engine. Oscar Onsrud and his son Rudy came up with, and submitted, a design for a Jet Motor (air turbine) to generate the power for the light, however, they were failed to win the project. A few months later Rudy was talking with a friend about his frustrations making the grove in the bottom of a cane chair using a router plane. A spark came in Rudy's head; he could repurpose the Jet Motor, which he had spent so much time for developing, to run on compressed air and spin a modified end-mill and make the routed grove easily. The modified end-mill would have to spin at 30,000 RPM, instead of the 3,000 RPM of a milling machine, in order to cut wood and not to burn it. These bits also needed a steeper rake and clearance angle to evacuate the chips than needed on a traditional end-mill. These new bits became known as router bits.

Further refinement produced the plunge router, invented by ELU (now part of DeWalt) in Germany in the 1940s. This is even better adapted for many types of work. In the

1960s, the power tool form of router became the more common form. Developed routers are often used in place of traditional moulding planes or spindle molder machines for edge decoration (molding) of timber.

Routing and milling appears similar, but are applied to different materials and so these require tools that are significantly different in detail. The mechanism of chip formation for both tools is different, according to the material.

Routing is applied to relatively soft and brittle materials, typically wood. As these materials are soft in small sections, router may be run at extremely high speeds and so even a small router may cut rapidly. When milling metals, the material is relatively ductile, although remaining strong even at a small scale. The cutters are thus runs more slowly, even when used in multi-horsepower milling machine.

There are two types of routers-plunge and fixed. When using a plunge base router, the sole of the base is placed on the face of the work with the cutting bit is raised above the workpiece, then the motor is turned on and the cutter is lowered into the workpiece. With a fixed-base router, the cutting depth is set before the tool is turned on. The sole-plate is then either rested flat on the workpiece overhanging the edge so that the cutting bit is not contacting the work (and then entering the work from the side then the motor is turned on), or the sole-plate is placed at required angle with the bit above the work and the bit is "rocked" over into the work once the motor is turned on. In each case, the bit cuts its way in, but the plunge router does it in a more refined way, although the bit used must be shaped so that it bores into the wood when lowered.

The baseplate (sole plate) is generally circular and may be used in conjunction with a fence attached to the base, which then ties the router against the edge of the work, or via a straight-edge clamped across the work to obtain a straight cut. Other means of guide the machine include the template guide bushing secured in the base around the router cutter, or router cutters with built in guide bearing. Router and cutter run against straight edges. Without this, the varying reaction of the wood against the torque of the tool makes it impossible to control with the precision normally required.

A router is a tool used to rout out (hollow out) an area in the face of a relatively hard workpiece typically of wood or plastic. The main applications of wood routers are woodworking, especially cabinetry. The hand tool form of router is the original form. Wood router is a specialized type of hand plane with a broad base and a narrow blade projecting well beyond its base plate power tool form of router, with an motor driven spindle, is the more common form, and the hand tool is now often called a router plane.

Fig.1.1 Wood router [13]

Wood router bits have many common and uncommon uses. They can be used to round off the edges of a normal office desk as shown in fig.1.1 or to create intricate and beautifully decorated wood panels. A wood router bit is used to hollow out an area on the face of a plank of wood. The tools consist of a broad-based wooden hand plane, which is used to flatten or reduce the thickness of a smooth surface of lumber, with a narrow blade that protrudes beyond the base plate. Today's routers are used in place of traditional molding planes when edge decoration is needed.

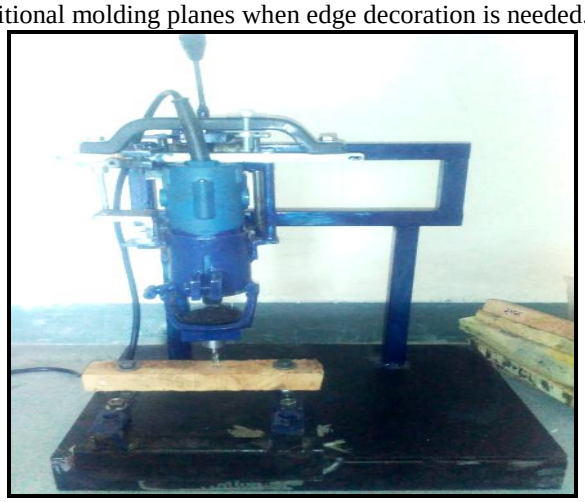


Fig.1.1 Wood router Machine Model

Wood router bits have hundreds of different varieties to create decorative effects to the edges of wood. These tools can be found with edge bits or non-edge bits. Edge bits have small wheel bearings which act as a barrier against the work in edge making. Non-edge bits require the use of a barrier, which can be found either on a router table or attached to the piece of wood or router.

Wood router bits can differ in diameter and can range in ½ inch diameters to 3/8 inch diameters. The speed of the bit's rotation varies based on the size of each tool and can range from 8,000 to 30,000 rpm.

Popular shapes that are made by wood router bits as shown in fig.1.2 include chamfer, v-groove, cove, round nose, rabbeting, dado, round-over, dovetail, Roman ogee and beading.

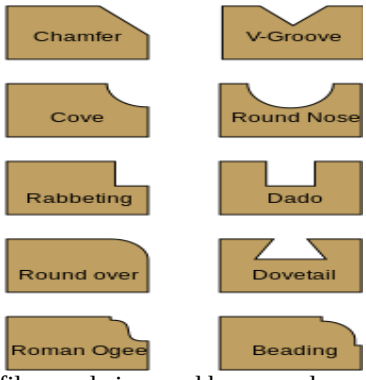


Fig.1.2 Profiles made in wood by several common router bits [13]

1. Problem Statement

To work under the presence of HAV is difficult, Vibration induced by Wood router machine are high.

II. LITERATURE REVIEW

Health & Safety Guidance on Hand-Arm Vibration (HAV)[1] has provided guideline for using handheld equipment. 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. Vibration disorders related to the use of hand-held power tools has been known and reported since long. It is therefore essential that low vibrating tools are developed and used. The new vibration regulations in Europe, based on the Physical Agents (Vibration) Directive, have put increased focus on the vibration control in industry.

NERC health & safety procedure [2] 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.

Xueyan S. Xu et.al [3] have conducted trial for finding out vibration power absorption (VPA) of different hand-arm substructures was estimated using biomechanical models of the hand-arm system in the bent-arm and extended arm postures, derived from impedance and transmissibility responses. The distributed VPA due to broadband random excitation and vibration spectra of different hand-held power tools were estimated. The results showed that the extended arm posture should be avoided since higher power (1.63 Watts) was absorbed in the hand-arm system in the extended arm posture than in the bent-arm posture (0.67 Watts) for identical hand forces and excitation level. The VPAs in the arms are greater in the low frequency region (below 25 Hz) than those of the hand. The VPA distributions of the hand are however greater than those of the arms above 100 Hz, the VPA values are however smaller than those below 25Hz. Although the trends of the distributed VPA of the hand-arm substructures due to broadband random excitation and power tool vibrations are similar, the VPA due to power tool spectra showed peaks at the operating frequencies of the tools and their harmonics and the percentage of power absorbed in different hand-arm substructures was dependent on the operating speed of the power tool. The higher the operating speed the higher the power absorbed in the hand. Peaks in the VPA of the fingers and palm, the substructures mostly affected by vibration-induced white fingers, occurred around 160 and 60 Hz, respectively, while those of the arms occurred in the 5e16 Hz frequency range (the vicinity of the maximum weight in ISO 5349-1 (2001) weighting, 12.5 Hz). This study suggests that different frequency weightings derived from distributed VPA of different substructures of the hand-arm system may resolve the reported discrepancy between the results of hand injury assessment using the current ISO 5349-1 (2001) guidelines and epidemiological studies.

Charlotte Astrom et.al, [4] compared the prevalence of symptoms of Hand-arm vibration syndrome (HAVS) and musculoskeletal symptoms in the neck and the upper limbs, between professional drivers of terrain vehicles and a referent group. Driving terrain vehicles are related to experiencing some symptoms related to HAVS, such as numbness, sensation of cold and white fingers, suggesting that there is a possible association between exposure to HAV generated from steering devices in terrain vehicles and symptoms of HAVS. Driving terrain vehicles is also related to experiencing musculoskeletal symptoms in the neck, shoulders, and wrist. These symptoms seem to be related to the exposure time.

Mirta Widia et.al, [5], [14], [15] 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.

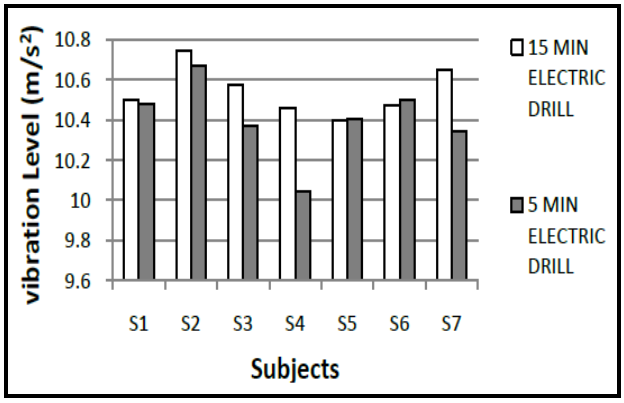


Fig.2.1 Vibration Level Drilling Wood Material [5]

Table no.2.1 Vibration level effect on Grip strength [5]

Subjects	5 min electric drill			15 min electric drill		
	GS (Before) (N)	GS (After) (N)	Decrease Ratio	GS (Before) (N)	GS (After) (N)	Decrease 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.2.1 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.

Lars Skogsberg[6] stated that Forces acting on the tool cause vibration. Tools for industrial use must be of very robust design to withstand the very hard use 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 tools can be regarded 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. These frequencies are with few exceptions below 200 Hz. Handles however cannot always be regarded as rigidly connected to the 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 acts on the tool and the result is vibration.

In all cases forces are the source of vibration. This leads to the three basic principles to control vibration:

i. Control the magnitude of the vibrating forces. Examples are the balancing unit on a grinder or the differential piston in a chipping hammer.
ii. Make the tool less sensitive to the vibrating forces. Examples can be when the mass of the guard on a grinder is rigidly connected to the tool to increase the inertia of the tool.
iii. Isolate the vibrations in the tool from the grip surfaces. Examples are vibration dampening `handles on grinders or pavement breakers, the air-spring behind the blow mechanism or the mass spring system in a chipping hammer.

An industrial power-tool can in most cases be regarded as a rigid body. The handles are not always part of this rigid body.

Forces acting on this rigid body are the source of vibration. The forces are either forces from the process or process independent e.g. unbalances in rotating parts.

There are three basic principles for vibration control. Control the magnitude of the vibrating forces. Make the tool less sensitive to the forces. Isolate the vibration in the tool body from the grip surfaces.

All three principals are used in vibration control on power tools either one by one or combined on the same tool.

Andrew K. Costainet.al, [7] 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. 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 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 sliding against each other. Structural or material damping, the category most commonly applied for industrial vibration control, is caused by internal friction within the 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.

M. Shinozuka et.al [8] has conducted research on passive and active fluid dampers in structural applications. Passive fluid damper are substantially more reliable, demand no power & cost significantly less compared to active damping system

i. Passive damper will have a constant spring stiffness and constant damping coefficient
ii. Active damper will have constant spring stiffness but variable damping coefficient by use of modified damper orifice design.

Rao V. Dukkipati & J. Srinivas [9] 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.

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.

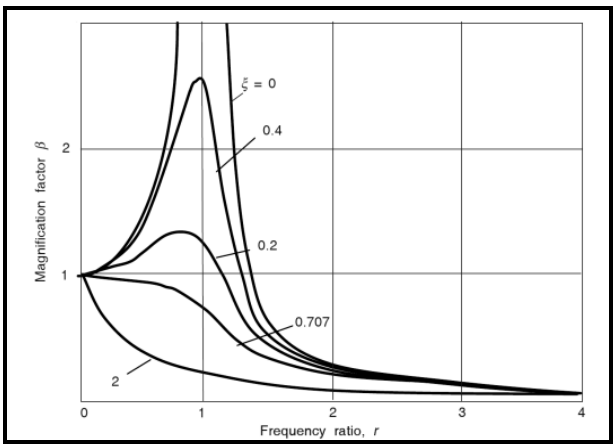


Fig.2.2 Magnification factor [9]

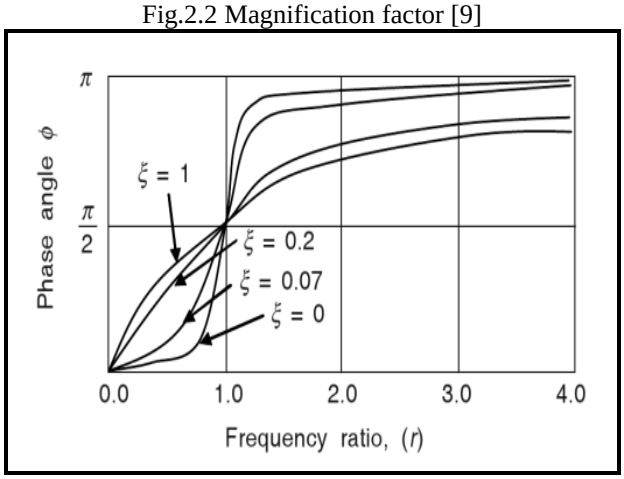


Fig.2.3 Phase angle [8]

Philippe DUFLOT et.al [10] &D. LEE et.al [11], [12] 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 for providing 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 stated 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 to 2.

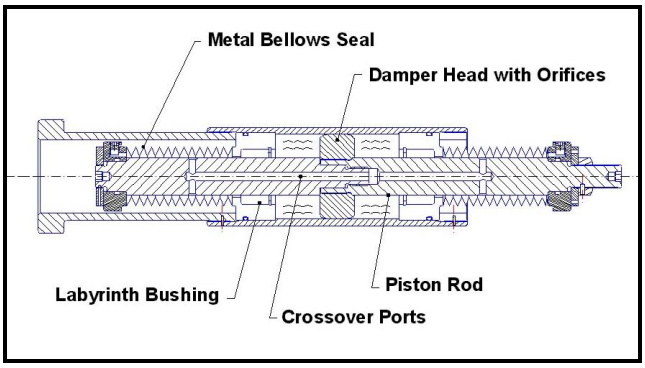


Fig.2.4 Viscous Damper [10]

Fluid viscous damper operate on principle of fluid flow through an 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 period of time. The pressure difference between the two chambers side cause silicone oil to flow through an orifice in the piston head and input energy is transformed to heat, which dissipates into the atmosphere.

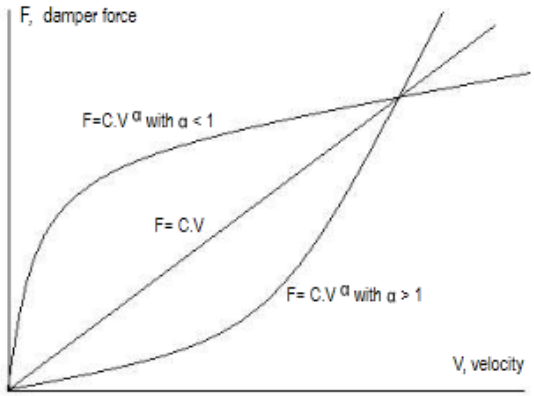


Fig.2.5 Force against velocity for different exponent values [10]

From this paper we conclude that there is no spring force in this equation. Damper force changes 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 few. However, the detailing of these elements varies highly 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.

III. RESEARCH METHODOLOGY

As working on router machine, the transverse vibrations are transferred to hand. Our aim is to minimize those vibrations, so the damper is added in between machine body & handle in vertical direction.

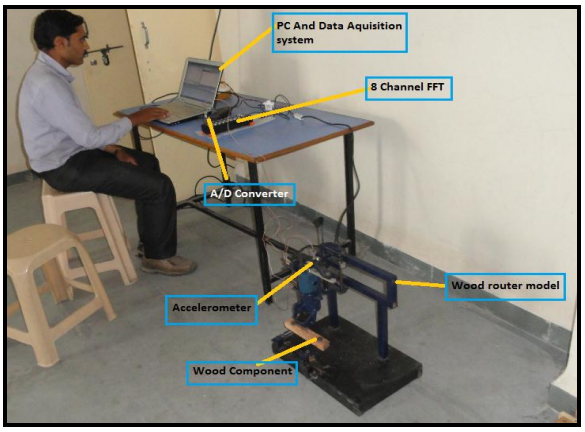


Fig.3.1 Machine testing setup

1. Design of hydraulic damper body

Material selection:

Designation	Ultimate Tensile strength N/mm^2	Yield strength N/mm^2
Aluminium	110	40

----------Table-1-28, Machine design data book, pg. no.-1.50

$fs_{c\ all} = 12 N/mm^2$

Hoope's stress due to fluid pressure:

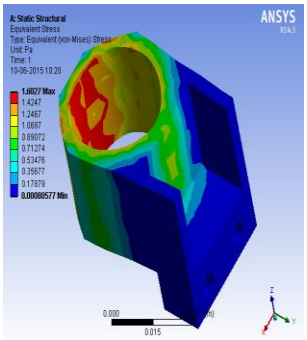
P= 3 bar,

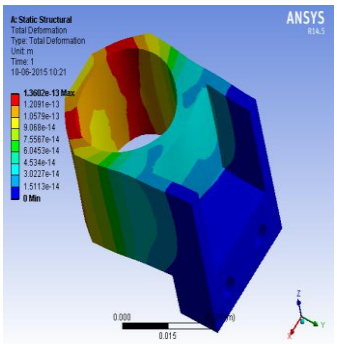
fc_h= 1.26N/mm^2

As fc_h<fc_{all} ; damper body is safe.

2. Analysis of damper body:

i. Vonmises stress & Deformation





3. Design of piston rod

Material selection:

Designation	Ultimate Strength N/mm^2	Yield Strength N/mm^2
EN24	800	680

----------PSG Design Data. Pg.no.1.10, 1.12 & 1.17

D irect Tensile or Compressive stress due to an axial load.

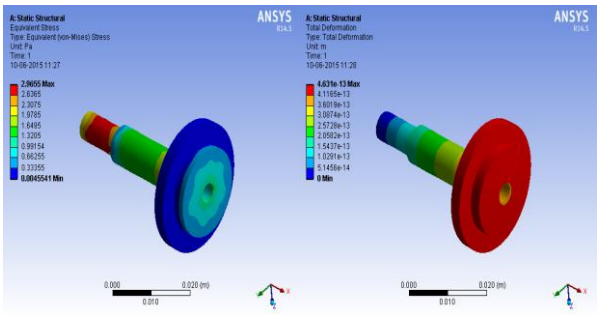
fs_{all}=108 N/mm^2

fc_{act} = 0.92 N/mm^2

As fc_{act}< fc_{all}; Piston rod is safe in compression.

4. Analysis of piston

i. Vonmises stress & Deformation



IV.RESULTS & DISCUSSION

Induced vibrations in Wood router machine model are measured by using 8 channel FFT analyzer [DEWE 43A]. Vibrations are induced in the form of acceleration (g) hence; Plots are drawn frequency v/s acceleration which is shown in fig.4.1, 4.2, and 4.3. The testing is done by using different wood component. In this testing we have used samples of Devgad, Jackfruit, Mango, Neem & Sal etc. Testing was done for three conditions without Damper, With Passive-Damper and With Active-Damper.

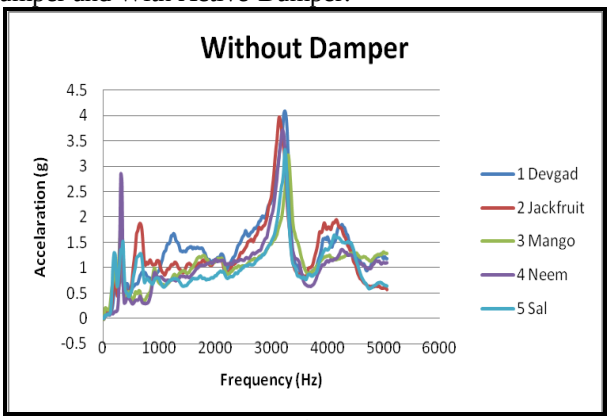


Fig.4.1 FFT graph for without Damper

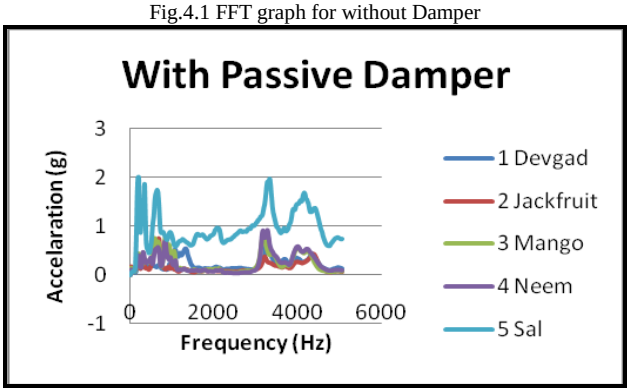


Fig.4.2 FFT graph for with Passive-Damper

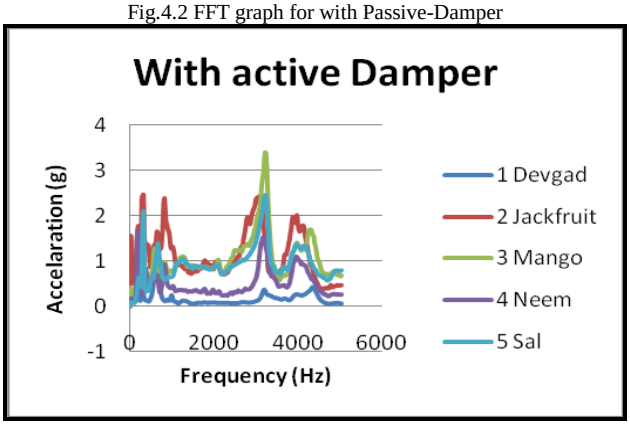


Fig.4.3 FFT graph for with Active-Damper

Table No.4.1 Analysis result for damper

Part Name	Maximum theoretical stress N/mm^2	Von-mises stress N/mm^2	Maximum deformation Mm	Result
Damper body	1.26	1.6	1.36 x 10^{-13}	safe
Piston	0.92	2.9	4.63 x 10^{-13}	safe

V.CONCLUSION

From FFT graph it is seen that peak acceleration is induced when tested the machine without damper. In passive damping case we have maintained its orifice diameter constant and minimum so its adverse effect was acceleration reduced by 50% & In Active Damper we could vary the orifice diameter hence the acceleration values reduced by requirement.

ACKNOWLEDGMENT

I wish to express my appreciation and gratefulness for the m ost cooperative attitude of Dr. C. R. Sonawane for their able guidance thoughtful suggestions and the constant moral sup port in completing this paper

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