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A Review on Design Development & Analysis Of Passive And Active Damper In Hand Held Wood Working Machine

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ABSTRACT

Hand-arm vibration (HAV) is vibration transmitted from a work processes into workers hands and arms. It cannot be caused by operating hand-held power tools and hand-guided equipment or by holding materials being processed by machines. Multiple studies have shown that regular and frequent exposure to HAV 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. The injuries that one could suffer include damage to blood circulatory system (vibration white finger [VWF], sensory nerves, muscles.[1]

Keywords- **Hand-arm vibration (HAV),high speed wood working machine,viscous fluid damper.**

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I. INTRODUCTION

The wood working jig saw machine is a high speed wood working machine used to cut wood work-piece in furniture making, Casting pattern making, wooden seat design, wood prototyping etc. Subsequent vibrations makes it difficult to operate the machine for longer time and so also power consumption per unit cut has been found to be very high , and vibrations lead to inaccuracy in cutting and error in profile shape.Thus methodology used in vibration Isolate the vibrations in the tool from the grip surfaces by introducing a viscous fluid damper in between the cutter body and the grip handle. The damper selected for the purpose is a semi-active one where in the damping coefficient of device is adjustable.[3]

Project work comprises of design and development of wood jig saw machine with 350 watt power where in the handle mounting will be designed to operate with and without fluid damper. Mathematical modeling of semi-active fluid damper to isolate vibrations produced during cutting process fluid damper where in damping coefficient is varied by use of modified damper orifice design.Fabrications and Testing of the developed wood router cutter without damper and with the semi-active damper to determine the Overall damping coefficient& RMS values at two cutting speeds.Comparative analysis of the performance of the jig saw machine by varying materials (Teak wood & packaging wood) so as to cutting speed (m/min), Material removal rate & dimensional accuracy.3-D modeling of set-up using Unigraphix Nx-8.0 CAE of critical component and meshing using Ansys i.e. the Preprocessing part.Mechanical design validation using ANSYS critical components of the system will be designed and validated Validation of strength calculations of critical for both modal and strength analysis.[4]

I. LITERATURE SURVEY

F. Weber,**[1]** A semi-active vibration absorber with real-time controlled magneto rheological damper (MR-SVA) for the mitigation of harmonic structural vibrations is presented. The MR damper force targets to realize the frequency and damping adaptations to the actual structural frequency according to the principle of the undamped vibration absorber. The relative motion constraint of the MR-SVA is taken into account by an adaptive nonlinear control of the internal damping of the MR-SVA.

M.F. Hassan, M. Mailah,[2] This paper highlights a simulation study involving the application of an active force control (AFC) strategy to suppress vibration on the rear handle of a handheld tool. The research was carried out to investigate the performance in terms of vibration reduction capability of a feedback controller employing AFC-based schemes on a selected powered portable machine (Hedge Trimmer Maruyama model Ht230D). Four types of control schemes were closely examined and compared involving the classic proportional-integral-derivative (PID) control scheme, the AFC with crude approximation (AFCCA) method, the AFC with iterative learning method (AFCILM)and AFC with fuzzy logic (AFCFL) method. Inherent vibration was measured from the real operation of the handheld tool through operational deflection shape (ODS) experiment. This data was later used in the simulation work together with other forms of modelled disturbances to test the robustness of the control scheme. Results show that the AFC scheme is able to suppress the vibration at the rear handle much better compared to the conventional PID control scheme.

Alan R. Klembczyk,[3] Oftentimes a dynamic system or structure requires that it be analyzed for performance or structural suitability within the context of a shock and vibration environment. This paper provides an outline of various applications and methods of implementing isolation, shock absorbing and damping within a wide array of dynamic systems and structures. Successful integration of these useful tools is essential when solving problems within the world of shock, vibration, and structural control. The theory, analytical techniques, and options for hardware selection are often complex tools. Therefore, it is essential that the analyst and the systems engineer be equipped with a basic understanding of control schemes and isolation system attributes that have been proven effective and reliable in a wide variety of shock and vibration isolation applications in the past. Specifically, key definitions are presented that are widely used within the shock and vibration community. Additionally, useful formulae are presented that will provide the user with an initial path forward with respect to solving typical problems. Finally, a comparison of different types of shock isolators, shock absorbers and dampers will outline their specific advantages and disadvantages when using them for typical applications within the commercial, military, and aerospace sectors.

Haruhiko KURINO,[4]This paper presents an ingenious passive hydraulic damper for structural control with high performance equivalent to that of a semi-active damper. This damper maximizes or minimizes the damping coefficient by regulating the opening of a flow control valve housed in the device, and absorbs much of the structure's vibration energy than a conventional passive viscous damper. The remarkable feature of this device is that all the valve control is carried out autonomously utilizing the pressure balance between two hydraulic chambers without any outer power resources. First, we explain the self-regulating hydraulic mechanism. Second, we present the results of dynamic loading tests on a full-scale prototype device (maximum force: 2MN) under both sinusoidal waves and non-stationary seismic response waves. It is thus confirmed that the developed device has the expected excellent energy dissipation capacity, and that the damper's dynamic characteristics can be well simulated by a simple analytical model. Finally, we demonstrate the results of seismic response analyses using an MDOF building model, and discuss the control performance of the device.

G.Z. Yao,[5]In this paper, a semi-active control of vehicle suspension system with magnetorheological(MR) damper is presented. At first a MR damper working in flow mode is designed. Performance testing is done for this damper with INSTRON machine. Then a mathematical model, Bouc–Wen model, is adopted to characterize the performance of the MR damper. With optimization method in MATLABand experimental results of MR damper, the coefficients of the model are determined. Finally, a scaled quarter car model is set up including the modelof the MR damper and a semi-active control strategy is adopted to control the vibration of suspension system. Simulation results show that with the semi-active control the vibration ofsuspension system is well controlled.

Martin Orecny,[6] The article deals with two alternatives of semi active suspensions of a seat of working machine. A magneto-rheological damper is used in the first case and in the second case the suspension of the seat is a combination of magneto-rheological damper and dynamic absorber. The dynamic absorber is composed from passive elements. In both cases, the dampers are controlled by thewell-known Sky hook algorithm. The dynamic analysis is focused on the influence of the applied passive dynamic absorber on the reduction of the seat displacements. The passive parameters of the seat suspension and the dynamic absorber were evaluated based on the optimization process using genetic algorithms according the defined minimization function.

Cristiano Spelta,[7]The aim of this work is the analysis and design of a control system for vibration and noise reduction in a washing machine. The control system is implemented via a semi-active magneto rheological (MR) damper located on the suspension that links the drum to the cabinet. The entire design procedure is outlined: first, the semi-active actuator is described and an experimental protocol is proposed and tested; two adaptive control strategies are proposed, designed and tested. Some experiments are done in an anechoic chamber to assess the noise reduction. The reported results show the effectiveness of the proposed control system.

II. PROBLEM STATEMENT

Undesired vibration can disturb our comfort, damage to structures, reduction of tools performance and machinery noise level but somehow it can be channeled for producing or extracting energy rather than suppress it. However, the command practice that being implemented is to control and suppress the undesirable vibration because it is creating unwanted sound (noise), the secondary major problem. One of the effects of undesired vibration is the Hand-arm Vibration Syndrome (HAVS). This syndrome involves circulatory disorders (e.g. vibration white finger), sensory and motor disorders, and musculoskeletal disorders, which may occur in workers who use vibrating powered portable tools.

III. PROJECT OBJECTIVES

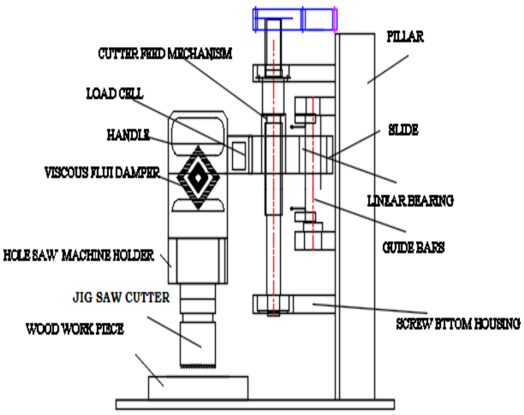
1. Design and development of hole saw machine 350 watt power with reduction gearbox to, increase cutting efficiency for hole sizes diameter 15 mm to 35 mm using high speed steel jig saw cutter.
2. Design and development of the passive and active type fluid damper to isolate and reduce the vibrations generated during sawing operations.

3. Testing of the developed jig saw cutter with and without the passive and active damper to determine the Overall damping coefficient & RMS values at two cutting speeds.
4. Comparative analysis of the performance of the hole saw machine with and without viscous fluid damper as to cutting speed (m/min) & dimensional accuracy. Validation of strength calculations of critical components using ANSYS. i.e. the post processing part for following parts.
5. Creation of Prototype: The selected mechanism and machine along with the damper will be designed using following machines :
   - Centre lathe
   - Milling machine
   - DRO- Jig Boring machine
   - Electrical Arc Welding

V. METHODOLOGY

- Need For Design
- Problem Statement
- Design Ideas & Alternatives
- DesignFeasibility
- Design Development
- Final design
- Detailed Design
- Prototype & Testing
- Production

A . LAYOUT & DESIGN OF THE SET-UP.



Active damping: Active damping refers to energy dissipation from the system by external means, such as controlled actuator, etc.

Passive damping: Passive damping refers to energy dissipation within the structure by add-on damping devices such as isolator, by structural joints and supports, or by structural member's internal damping.

VI. DESIGN

Machine design is an art and technique of planning the construction of a new or improvised machine. The machine may be entirely new in concept or modification of the existing machine for better utilization & economy. Development Design needs considerable scientific & innovative design ability in order to modify existing design into a new idea by adopting new materials or methods of manufacture. In this case although the designer starts from existing design the final product may differ quite markedly from the original product. Hence a careful design approach has to be adopted. The total design work, has been split up into two parts:[5]

- System design
- Mechanical Design

System design:

System design mainly concerns the various physical constraints and ergonomics, space requirements, arrangement of various components on main frame at system, man + machine interactions, No. of controls, position of controls, working environment of machine, chances of failure, safety measures to be provided, servicing aids, ease of maintenance, scope of improvement, weight of machine from ground level, total weight of machine and a lot more.

In mechanical design the components are listed down and stored on the basis of their procurement, design in two categories namely,

- ➢ Designed Parts
- ➢ Parts to be purchased

For designed parts detached design is done & distinctions thus obtained are compared to next highest dimensions which are readily available in market. This amplifies the assembly as well as postproduction servicing work. The various tolerances on the works are specified. The process charts are prepared and passed on to the manufacturing stage.

The parts which are to be purchased directly are selected from various catalogues & specified so that anybody can purchase the same from the retail shop with given specifications.

Mechanical design:

Mechanical design phase is very important from the view of designer .as whole success of the project depends on the correct deign analysis of the problem. Many preliminary alternatives are eliminated during this phase. Designer should have adequate knowledge above physical properties of material, loads stresses, deformation, failure. Theories and wear analysis , He should identify the external and internal forces acting on the machine parts
These forces may be classified as ;

a. Dead weight forces
b. Friction forces
c. Inertia forces
d. Centrifugal forces
e. Forces generated during powertransmission etc.

MOTOR SELECTION

Thus selecting a motor of the following specifications

Single phase AC motor

Commutator motor

TEFC construction

Power = 350 watt

Speed= 0-9000 rpm (variable)

Motor is an Single phase AC motor, Power 350 watt, Speed is continuously variable from 0 to 9000 rpm. The speed of motor is variated by means of an electronic speed variator. Motor is an commutator motor i.e., the current to motor is supplied to motor by means of carbon brushes. The power input to motor is varied by changing the current supply to these brushes by the electronic speed variator, thereby the speed is also is changes. Motor is face mounted and is bolted to the motor base plate. The reduction gear box standard spiral bevel gear box for 1:3 reduction ration hence the speed at the input to the flexible shaft is 3000 rpm. The gear box selected is suitable to transmit 550 watt power. (REF :Dewalt 4" DC wheel grinder)

DESIGN INPUT DATA

Analysis For Power Requirement Of Machine.

I). CATALOGUE METHOD

Reff:- PRODUCT CATALOGUE NORTON GRINDWELL LTD.

CUTTER used for W00D cutting applications are;B88 x –44 A24 T

These wheels have following performance characteristic:

1. MATERIAL REMOVAL RATE (MRR) = 0.05 cm^2 / Sec.
2. AVAILABLE POWER = 0.04 TO 0.125 KW /inch of wheel.
3. MAX OPERATING SPEED = 6000 RPM.

TARGET:- Application is wood/pvc cutting ;

Max^m diameter =ϕ 20 mm ≈ 0.8"

To CUT Material =wood / PVC

⇒Power Requirement at m/c spindle = 0.8 x 0.06

= 0.048 Kw.

= 0.064343 Hp

Let us round off the Input power to; 1/15HP

⇒ INPUT POWER= 1/15 HP

II) .THEORETICAL METHOD

INPUT DATA

For bar cutting applications;

Ideal metal removal rate;

MRR= 0.05 cm^2/Sec;

Thickness of BLADE = 1.2mm

⇒Volume of material

Removed / sec = MRR x t

= 0.05 x0. 12

= 0.006cm^3/sec

⇒ Q = 0.36 cm^3/min.

In plunge grinding an optimum in feed rate for bar cutting application= 0.05 mm/rev

Average unit power requirement for CUTTING:

⇒ Average unit power requirement for cutting at anin feed rate of 0.05 mm/sec;

U=0. 18 Kw/cm^3 /min.

Now ; Power (N)= U x Q

Where ;

(N) = Power (Kw)

U = unit 1 cm^3 /min

Q= Volume of material removed /min

⇒ N= 0.36 x 0.12

N= 0.0432 Kw ≈ 0.057 HP

⇒ INPUT POWER = 1/15 HP

DESIGN OF INPUT SHAFT:

T_{Design} = 2.3 Nm.

= 2.3 x 10^3 N.mm

Selection of input shaft material

Ref :- PSG Design Data.

PgNo :- 1.10 & 1.12,1.17

Designation	Ultimate Tensile Strength N/mm^2	Yield strength N/mm^2
EN 24 (40 N; 2 cr 1 Mo 28)	720	600

Using ASME code of design;

Allowable shear stress;

$Fs_{all}$is given stress;

Fs_{all} = 0.30 syt = 0.30 x 600

=180 N/mm^2

Fs_{all} = 0.18 x Sult = 0.18 x 720

= 130 N/mm^2

Considering minimum of the above values;

fs_{all} = 130 N/mm^2

As we are providing dimples for locking on shaft ;

Reducing above value by 25%.

⇒fs_{all} = 0.75 x 130

= 97.5 N/mm^2

a) Considering pure torsional load;

$$T_{design} = \frac{\Pi \times fs_{all}\ d^3}{16}$$

$$\Rightarrow d^3 = \frac{16 \times 2.3 \times 10^3}{\Pi \times 97.5}$$

d = 4.7 mm

Selecting minimum diameter of spindle= 6 mm from ease of construction as standard 6x 6mm square is available for ease of construction.

VII. EXPECTED OUTCOMES

This paper gives an overview of the design, development & Analysis Of Passive And Active Damper In Hand Held Wood Working Machine. Design and development of hole saw machine 350 watt power with reduction gearbox to, increase cutting efficiency for holesizes diameter 15 mm to 35 mm using high speed steel jig saw cutter.

From the experiment investigation of active and passive damper the hand vibrations are reduced and damping coefficient increases.

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