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FLUID FLOW MODELLING OF A FLUID DAMPER WITH SHIM LOADED RELIEF VALVE

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ABSTRACT

Automobile shock absorber consists of a dissipative element connected in parallel with an elastic element. Fluid damper are most reliable to be used as a dissipative elements in the vehicle shock absorber. Damping action in fluid shock absorbers is obtained by throttling viscous fluid through an orifice. Fluid shock absorbers offer sufficient damping force with compact size, also the damping force is linear in nature. Damping force of the fluid damper depends on orifice properties and on the physical properties of oil used. The paper discusses mathematical modelling of the fluid damper which uses number of shim controlled orifices. FEA is performed to compute stiffness of the shims used with the orifices. Matlab programme calculates pressure difference and damping force across the piston using fluid flow continuity equations. Finally single degree of freedom Matlab simulink of the damper is used to find displacement transmissibility for range of frequencies.

KEYWORDS: Shock Absorber, Automobile Suspension, Fluid Damper

INTRODUCTION

Fluid dampers used in automobile shock absorber consist of a piston with number of orifices and disc shaped shims as shown in Figure 1. Disc shaped shims are used for providing additional damping force at higher suspension velocities [1,2]. Portion of the cylinder above piston is called rebound chamber, whereas the portion below piston is compression chamber. As vehicle undergoes vertical excitations, piston forces the oil from compression chamber to rebound chamber, and vice versa. As the oil flows within these two chambers, pressure loss due to throttling action of oil at orifice valves and leakage path around the piston causes damping force on the piston. The damping force on the piston depends on this pressure difference. For higher pressure difference across the two chambers, damping force is higher and vice versa. For higher rate of oil flow within these two chambers, pressure difference across these is lesser. Damping coefficient of the damper is calculated from the damping force of the damper at given velocity. Fluid dampers in automobile shock absorbers are to be designed for suitable damping coefficient, which is close to 2000 N-s/m for a passenger car. Damping coefficient of the vehicle is important for ensuring least acceleration transmitted to the occupants and also for ensuring least variation in the forces transmitted to the tyre. Accurate mathematical model of the damper is essential for designing the flow passage in the fluid damper.

For lower suspension velocities, oil flow takes place through outer orifices and leakage path around the piston. Whereas for higher suspension velocities, increased pressure difference in compression and rebound chamber causes disc shaped shim valves to lift from its position. This results in additional flow passage for the oil through inner orifices, which provides additional damping force.

For lower velocities of suspension, acceleration transmitted to the occupants is of primary concern; this can be ensured by providing sufficient oil flow between the two chambers of shock absorber. For higher suspension velocities, if oil flow rate in kept high, excessive movement of the tyres results in hitting over travel of the shock absorber. This necessities designing the flow area in the shock absorber, such that complete stroke of the shock absorber is utilized. Also variation of the tyre force transmitted to the tyre along the stroke of the damper should be minimum. Higher variation in this force will reduce tyre life and also damage the road. Also net force transmitted from the tyre to the road should be always in down ward direction, to ensure safety of the vehicle [3].

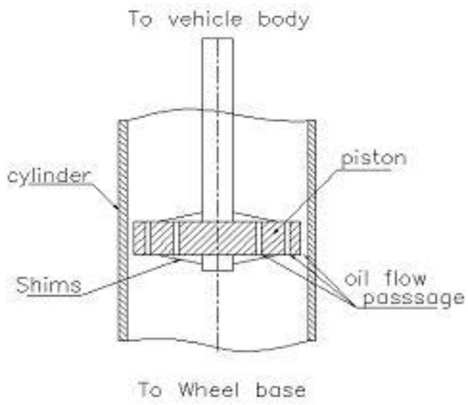


Figure 1: Construction of Fluid Damper Used Automobile Shock Absorber

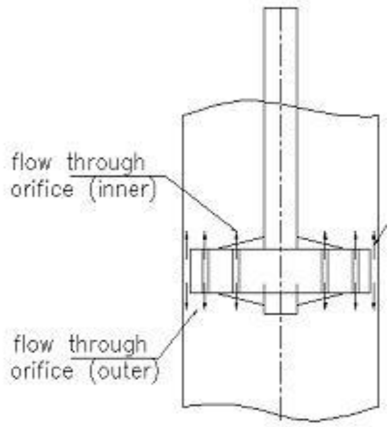


Figure 2: Oil Flow in the Fluid Damper

Limin et al (2010) have presented passive suspension with hydraulic shock absorber mini-car with improved design [4]. Stepper motor added to adjust settable orifice in valve of the shock absorber. Authors have presented simulation model in Matlab simulink for the shock absorber and bench test have been performed for validating the results. Semi-Active Suspension presented by the authors is simple in structure with low cost as compared to costly systems presently available. Zhou et al (2008) have modelled throttle slice deformation of the shock absorber. The authors have studied differential equation for deformation of the slice valve, and obtained deformation coefficient of the slice at various velocities. However the authors have not studied damping coefficient of the damper. Zhu et al (2008) have analysed taxing performance of the landing gear shock absorber, using forces on the piston of shock absorber [5]. Simulation in Adams indicates that side orifice, variable orifice and high pressure cavity in the damper can reduce over travel of the shock absorber. Also increasing volume of high pressure can reduce excessive stresses in lo pressure cavity. Zhou et al (2008) have performed simulation on telescopic shock absorber using fluid flow equations [6]. Authors have modelled oil flow through orifice, side path of piston and orifices controlled by shim valves.

Authors have analysed fluid damping force across stroke of the shock absorber. Eyres et al (2004) have outlined several possible methods of modelling a passive hydraulic damper, with a bypass tube [7]. This tube is opened by a pre-compressed relief valve. Initially a simple algebraic model is derived which is developed into a more computationally complex model incorporating the dynamics of the internal spring valve and fluid compressibility. Numerical simulations presented by the authors indicate realistic dynamical phenomena and suggest key design parameters. A valve between the two sides of piston is controlled by precompressed spring. When the pressure exceeds certain value this conical valve lifts from its seat providing extra oil flow passage.

Oil flow area between the rebound and compression stroke should be designed for required damping coefficient. The presented paper describes mathematical model of the damper based on flow equation for fluid flow through orifices and side leakage path. Finite element analysis (FEA) is performed on the shims to compute cracking pressure at which the shims open up providing extra damping force at higher suspension velocities. Matlab simulink model computed pressure difference and fluid damping force on the piston. Finally Matlab simulink model calculated motion transmissibility of the damper.

MATHEMATICAL MODEL

Figure 3 shows oil flow in the compression and rebound chamber of the fluid damper. Qlp is the flow of oil through piston clearance of the damper. Qv is the oil flow through outer orifice, whereas Qb is the oil flow through inner orifice. Total oil flow are in both compression and rebound stroke is obtained as summation of oil flow through side clearance and piston orifice.

$$Q = Q_v + Q_b + Q_{lp} \tag{1}$$

Ap is the cross sectional area of piston, whereas Ar is the cross sectional area of the piston rod. Considering piston velocity, flow rate in compression stroke is given as [7],

$$Q = C_d(A_p)\dot{x} \tag{2}$$

Similarly flow rate in rebound stroke is given as [7],

$$Q = C_d(A_p - A_r)\dot{x} \tag{3}$$

Where

A_p - Piston area

A_r - Rebound area

C_d - Coefficient of discharge

For calculating pressure difference across the damper we need individual flow rate across leakage path and orifice valves.

Flow rate across outer orifice is give as [7],

$$Q_v = A_b\sqrt{\frac{2(P_c - P_r)}{\rho}} \tag{4}$$

Where, ρ − kinematic viscosity of the oil

Ab – flow area

Pc – Pressure in compression chamber

Pr – pressure in rebound chamber

Flow are through leakage path across the piston and cylinder is given as [7],

$$\text{Qlp} = \left(\frac{(Pc - Pr)\,b^3}{12\,\mu l} + \dot{x}\,\frac{b}{2}\right)\pi \text{Dp CD} \tag{6}$$

Where, Δp - Pressure difference in compression and rebound chamber

μ - kinematic viscosity of the oil

b - width of piston

Dp - diameter of piston

l - Piston circumference

CD - Coefficient of discharge

Flow equation through inner orifice which is controlled by the shims is complex one. For solving this flow equation, the shim is modelled in ProE and deflection of the shim for pressure on the same is found using Ansys analysis. Figure 3 shows pressure applied on the shim because of the oil flow, whereas Figure 4, shows deflection of the shim for applied pressure. Stiffness values were calculated for four standard shims thickness values: 0.01”, 0.012”, 0.015” and 0.02”.

Shim stiffness values were calculated for five different shim hole configurations as, with zero holes and with 2, 3, 4 and 5 holes. Comparison to the shim with no holes demonstrates the effects of holes in the shims on the shim stiffness.

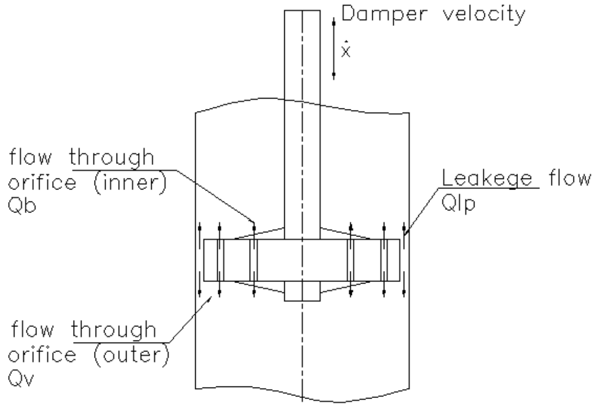


Figure 3: Oil Flow in the Fluid Damper

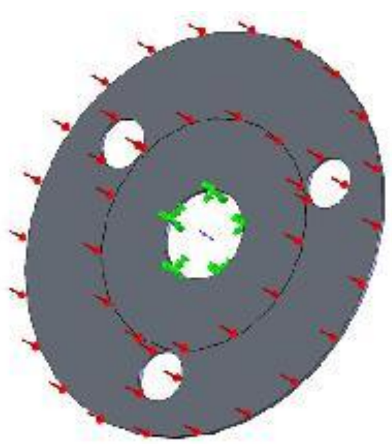


Figure 4: Pressure Applied on the Shim

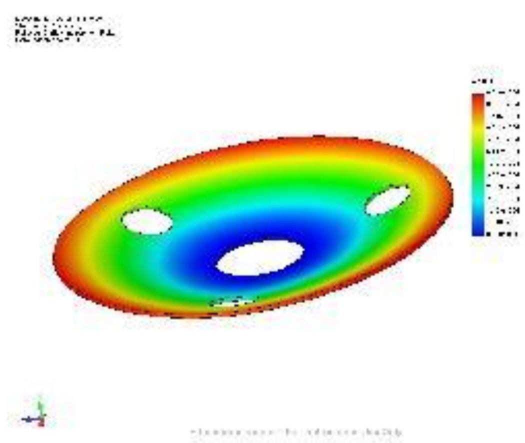


Figure 5: Deflection of three Hole Shim - Ansys Analysis

Figure 6 shows plot of stiffness and number of holes calculated using the FEA for various shim thicknesses. An increase in thickness gives an increase in the shim stiffness. The difference in number of holes is shown to be minimal. The percent difference in each thickness case is less than seven percent error for any number of holes tested. This leads to the conclusion that a stiffness of the solid shim can be used for simulations.

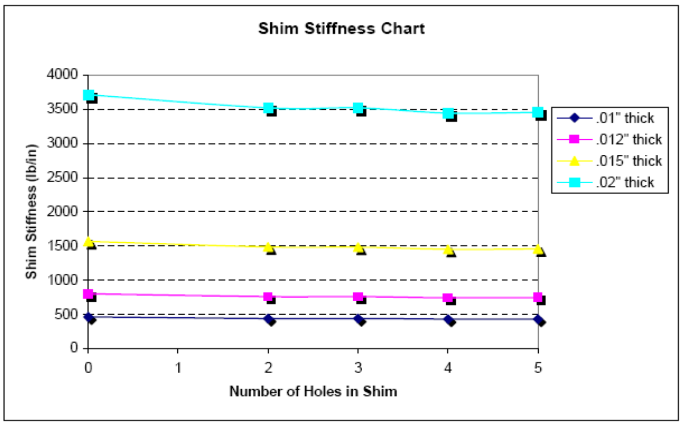


Figure 6: Shim Stiffness Chart for Varied Shim Thickness

While calculating Qb, two pressure drops are associated with this flow rate as [7],

ΔP_{valve} = Difference in pressure below shim and that of rebound chamber (7)

ΔP_{po} = Difference in compression chamber pressure and that of pressure below shims (8)

Where

Pc - pressure in compression chamber

Pr - pressure in rebound chamber

Pv - pressure below the shims

k- stiffness of the shims as calculated form FEA

Figure 7 shows model of oil flow through inner orifice.

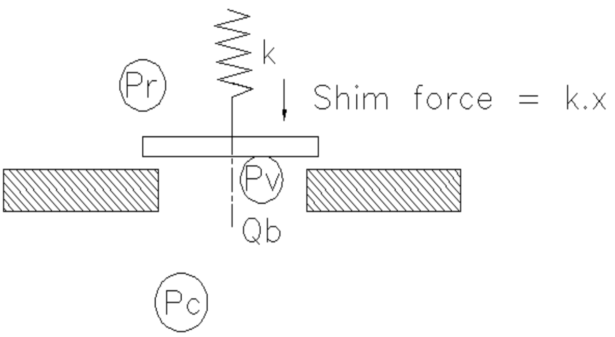


Figure 7: Oil Flow through Inner Orifice

As fluid exits the compression chamber and flows through the piston orifice, the first pressure drop occurs. This is denoted ΔPpo.The second pressure drop, ΔPvalve occurs across the valves after the flow has exited the piston orifice. Obviously, ΔPpo + ΔPvalve is equal to (Pc-Pr)

The flow rate through the piston orifice for these pressure drops is given as [7],

$$Q_v = A_0\sqrt{\frac{2\Delta p_{po}}{\rho}} \tag{9}$$

The flow through the piston orifice is equal to the flow through valves due to conservation of mass. Valve flow is driven by the pressure drop shown in equation [7],

$$Q_v = A_{v,flow}\sqrt{\frac{2\Delta p_{valve}}{\rho}} \tag{10}$$

The complexity arises when modelling the Av, flow term. The flow leaving the piston orifice has contacted the shim stack and essentially turned 90 degrees. For this flow, the flow area is the cylinder wall area defined by the circumference of the shims and height of the shim stack deflection [7].

$$A_{v,flow} = \alpha\, \pi D_v y\, C_D \tag{11}$$

where Dv – valve diameter

y – shim deflection

In equation 11, πDv is the circumference of the largest shim in the damper. The term α is area flow correction factor. We used a value of 0.5 in his model for a damper with three compression holes and three rebound holes. The damper has a variable number of piston flow orifices. Equation 10 and 11 gives [7],

$$Q_v = (\alpha\, \pi\, D_v y)\sqrt{\frac{2\Delta P_{valve}}{\rho}} \tag{12}$$

Finally force on the piston is written as,

$$F = m_p\ddot{x} + (compression\ pressure\,)A_c - P_r(rebound\ pressure)A_r \tag{13}$$

Equations as discussed above 1 to 13 are solved using Matlab simulink model. Figure 8 and 9 shows simulation results for compression chamber pressure and damping force at 1 Hz frequency with 25 mm amplitude. Plot of compression pressure and piston velocity is circular loop, whereas the plot of piston damping force is straight line, indicating linear variation of the damping force with respect to piston velocity.

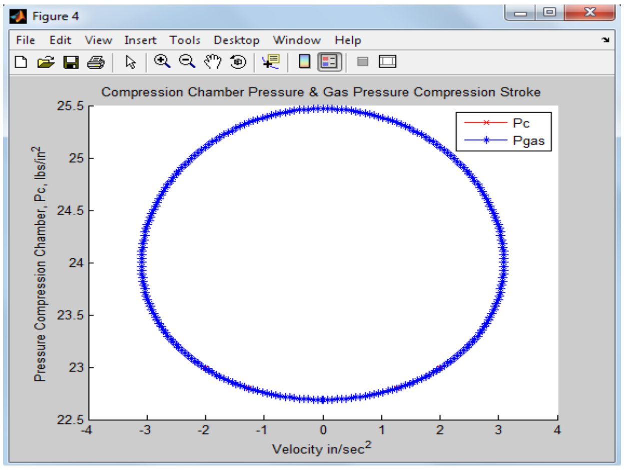


Figure 8: Plot of Compression Pressure and Piston Velocity

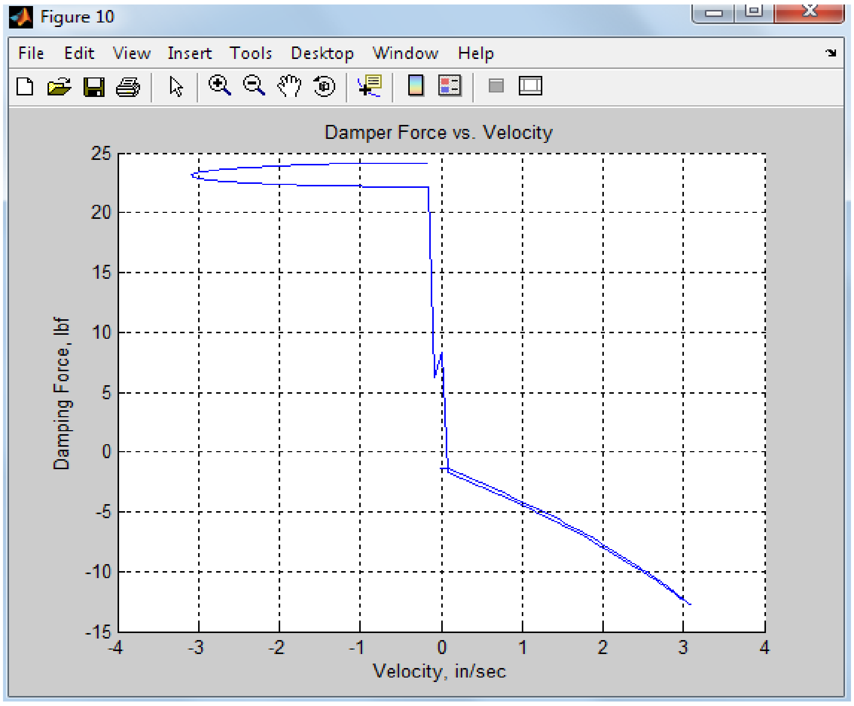


Figure 9: Plot of Damping Force and Piston Velocity

Finally Matlab simulink model with single degree of freedom is used to calculate transmissibity of the damper. Figure 10 shows Matlab model of the damper whereas Figure 11 shows simulation result for displacement at the piston for input displacement of 25mm. Load supported by the damper is 4kg, 8 kg, 12 kg and 17 kg resp.

Following parameters are used during the simulation,

Diameter of piston: 25 mm, Side clearance between piston and cylinder: 0.15 mm, width of the piston: 15 mm, diameter of throttle holes: 1.5 mm, number of throttle holes: 12 nos.

Figure 11 shoes Matlab simulink model of the fluid damper.

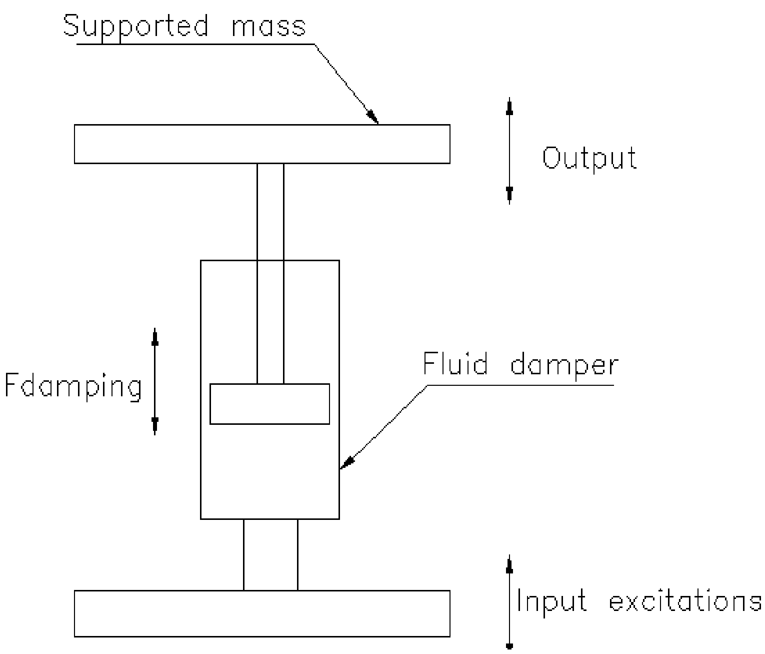


Figure 10: Single Degree of Freedom Model in Matlab

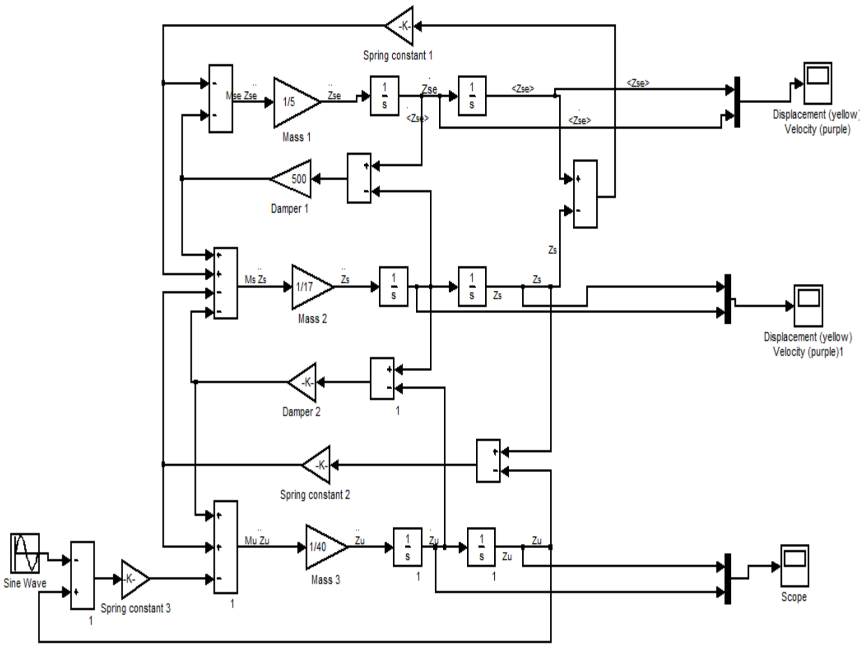
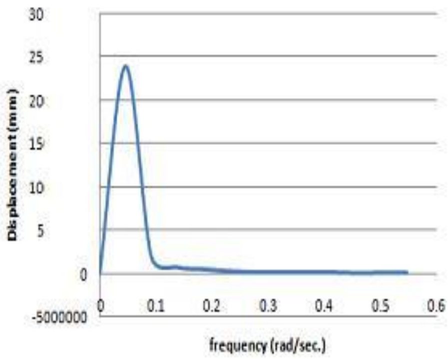
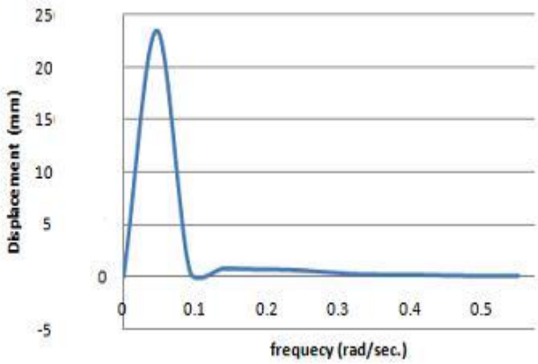


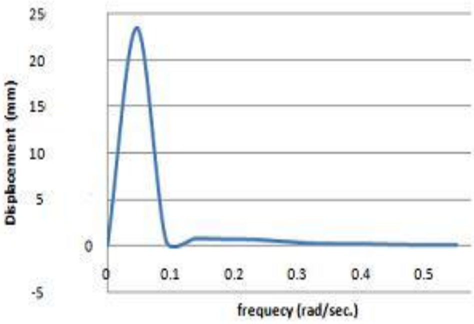
Figure 11: Matlab Simulink Model of the Fluid Damper



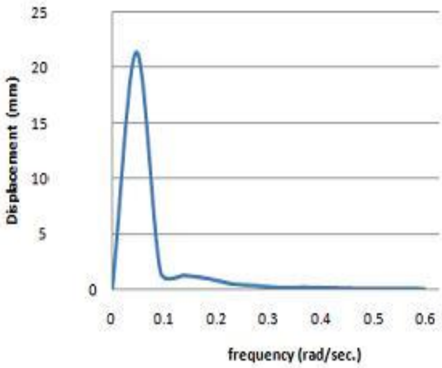
(a) For Load Supported of 4 kg



(b) For Load Supported of 8 kg



(c) For Load Supported of 12 Kg



(d) For Load Supported of 17 Kg

Figure 12: Simulation Results for Displacement at Supported Mass at 1 Hz Frequency

Figure 12 shows displacement transmitted to the supported mass at various frequencies for different values of sprung mass. Figure 13 shows simulation result for frequency response when the supported mass is 17 kg. As seen from the plot, displacement transmissibility is maximum near 1 Hz frequency.

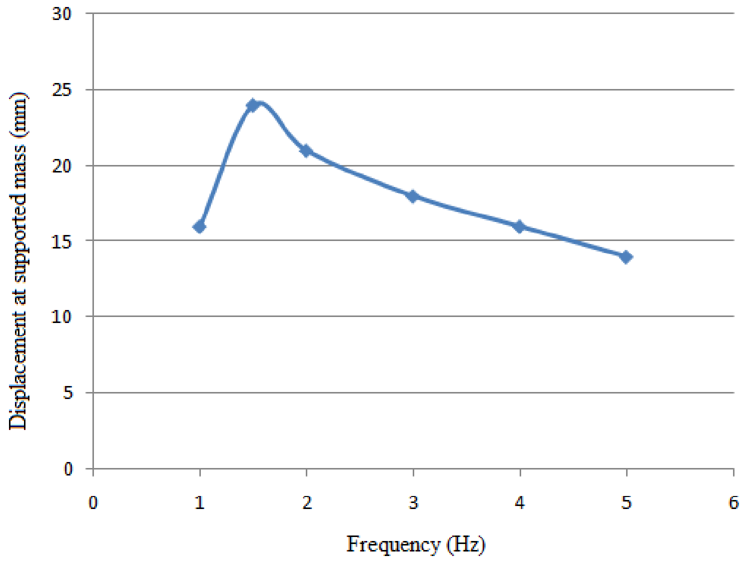


Figure 13: Frequency Response Curve

CONCLUSIONS

Mathematical model of a fluid damper consisting of orifices and shim controlled valves is presented by the paper. It has been observed that for larger bleed area, transmissibility reduced due to increase in oil flow rate. Increase in frictional losses also reduces transmissibility of the damper. It is better to have lower transmissibility at lower frequencies, whereas it is recommended to have higher transmissibility and lower oil flow rate at higher frequencies.

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