Development of Semi-Active Suspension System- Variable Orifice based Damper Prototype

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Abstract – *The semi-active suspension in automotive employs the damper unit with varying performance based on condition of roads, bumps, potholes when vehicle passes over them. The variable orifice dampers used for tuning of dampers, are once tuned gives particular kind of dynamic response. This aspect of existing variable orifice damper is examined for developing the semi active suspension system in this research work. The mathematical modelling for damping resistance is developed and is compared for various orifice sizes, this data is then used for developing the proposed CAD model dimensions of damper. The final objective is to calibrate the developed prototype to make it work best suitable as mechanically adjustable semi-active suspension system, which is presented in another research paper. The simulation work is done with the software MATLAB, and CAD modelling is done with the help of software Creo 4.0*

***Key Words*: Mechanically adjustable orifice, Mathematical modelling, MATLAB, Semi-active suspension, MR dampers vs mechanically adjustable dampers**

1. INTRODUCTION

Handling and ride comfort are very important characteristics that influence the quality of the vehicle. These characteristics depend on the suspension system of the vehicle. A semi-active damping system is basically a dissipative element in which the dissipation law can be actively modulated. This system due to its cost effectiveness, light weight and low energy consumption, is expected to be used in passenger vehicles in the near future. With sensible control law, the semi-active systems provide an intermediate performance between fully active systems and passive systems. An active suspension requires significant external power to function and that there is also in considerable penalty in complexity, reliability, cost and weight. With a view to reducing complexity and cost while improving ride, handling, and performance, the concept of a semi-active suspension has emerged. In this kind of system, the conventional suspension spring is usually retained, while the damping force in the shock absorber can be modulated in accordance with operating conditions. With passive damping these requirements are conflicting, since a hard damper results in better stability but reduced ride comfort, whereas a soft damper results in the converse effect. Semi-active devices can be broken into two distinct categories: those that use smart materials, and those that do not. Both are computer controlled intelligent systems, yet their principles of operation differ. Smart material devices include electro-rheological and magneto-rheological (MR) fluids, Shape Memory Alloys, Piezoceramics, and any other material whose physical properties change in response to magnetic, electrical, or thermal stimulus. Semi-active devices that do not use smart materials instead use more conventional actuators to modify their output force. Common actuators include voice coils and solenoid valves, pneumatic or hydraulic cylinders, as well as stepper motors and servos.

The resistance or dampening characteristics of passive suspension damper depends mainly on the resistance offered by the fluid flowing into compression and extension chamber through orifices or valves.

1.1 Adjustable valve damper

The geometrical feature of these flow valves (or orifice) in the damper, orifice area is responsible for the resistance offered by the damper. The effect of this orifice area change on the force output i.e. the resistance of the damper, is studied in this project.

The variation in the orifice area can be achieved with opening and closing the valves as in usual hydraulic circuits. This technique to vary the valve area is discussed in the thesis in later chapter.

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1.2 MR materials

A magnetorheological (MR) material is one for which the rheological properties, such as yield stress and viscosity, depend upon the magnetic field. MR liquids are sometimes described as 'low voltage' in contrast to ER liquids, but this is misleading. They are not subject to any voltage, only to a magnetic field, but the field is generated by a current of order one amp at low voltage in a field coil external to the liquid.

The MR liquid is formed by suspending numerous small solid particles, typically, a few micrometers in diameter, in a low-viscosity mineral or silicone carrier oil. The average diameter is about 8 mm with a normal range of 3–10 mm. The solid particles are ferromagnetic, basically just soft iron. Fibrous carbon may be added, and also a surfactant to minimize settling out. The result is a very dense 'dirty' grey to black oil. The shear strength achievable with magnetic field on is typically 50–100 kPa at fields of 150–250 kA/m.

The magnetic activation is not sensitive to electrical conductivity, so temperature has less effect than for ER devices. [3]

Gordaninejad and Breese examined the effects of temperature on the force characteristics of three separate MR dampers [9]. Additionally, heat generation over time was evaluated experimentally. A theoretical model was developed to represent the temperature increase in a MR damper due to a constant sinusoidal input and current to the electromagnet.

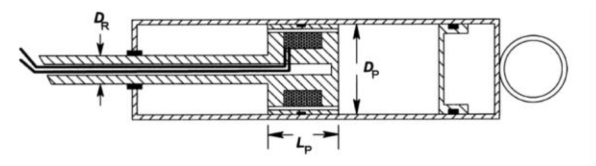

Fig -1: A typical MR damper design[3]

The semi-active suspension system developed so far are using MR (magneto-rheological) dampers. The working of the MR dampers depends on the ferromagnetic fine powder suspended or mixed in base damper oil. **The Dixon J []** discusses the MR damper and working of the MR damper.

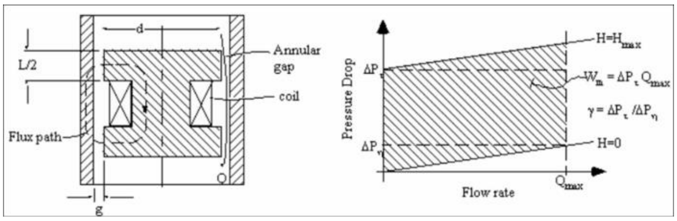

Fig -2: Section view: Typical mono tube MR damper

A significant amount of research has been conducted on electro-mechanically adjustable dampers, although much of it has been proprietary. Kitching, et al., developed a solenoid actuated spool valve hydraulic damper [12]. The non-linearity of the valve was modeled and evaluated experimentally through hardware in the loop testing. The semi-active damper showed the most influence in the low frequency bounce (heave) mode, reducing RMS body accelerations by up to 12.3 %, depending on road conditions. The semi-active damper was also shown to reduce peak absolute body acceleration due to transient bump inputs, without compromising settling time. The damper developed in [12] is similar to many other semi-active hydraulic dampers previously studied. ZF Sachs GmbH utilize a similar solenoid actuated proportioning valve in their Continuous Damping Control® semi-active hydraulic damping system.

Usman and Parker developed a low cost electro-hydraulic proportioning valve suitable for semi-active suspensions. Their goal was to reduce the unit costs associated with high precision machining, while providing a valve that is insensitive to fluid contamination and periods of inactivity [21]. A floating double disc configuration was developed and modeled mathematically. Magnetic and fluid forces versus displacement were calculated and used to relate differential coil current to disc position. Sun and Parker expanded on the previous work by integrating the double disc valve configuration into a semi-active damper [20].

MR dampers utilize magnetorheological fluids, instead of hydraulic oils, as the working fluid. The apparent viscosity of the MR fluid is varied by controlling the current through an electromagnet. This electromagnet creates magnetic flux path between the piston and damper body which locally energizes the fluid flowing through an annular gap between the outside of the piston and the damper body. A schematic of the flux path, piston, and damper body layout is shown in Figure 2, as well as a plot of the pressure drop and flow rate relationship for different magnetic field strengths.[6]

2. MATHEMATICAL MODELLING OF VISCOUS DAMPING

Conventional hydraulic dampers work by passing fluid through an orifice, or series of orifices, in response to the presence of a displacement input. When the working fluid, typically petroleum based oil, is passed through the damper valves, a pressure differential is created on opposing sides of the piston. This pressure differential acts upon the area of the piston to create a force which opposes damper motion. To better understand the nature of the various forces at work in a conventional monotube damper, a free body diagram (FBD) is shown in Figure 2.

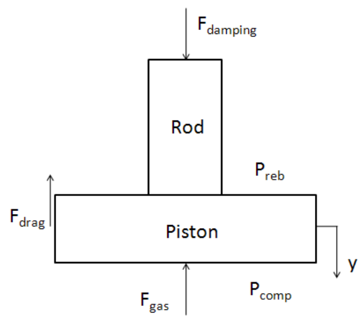

Fig -3: Free body diagram of damper internals

In addition to the damping, drag, and gas forces shown in the FBD, the pressures on the rebound and compression sides (P_{reb} and P_{comp}, respectively) give rise to the dominant force which reacts $F_{damping}$. A force balance in the y-direction yields,

$$\Sigma F_y = 0$$

$$F_{damping} + P_r\left(A_p - A_r\right) - P_c\left(A_p\right) - F_{drag} - F_{gas} = m * a_y$$

where $F_{damping}$ is the input force, P_{reb} and P_{comp} are the static pressures on the rebound and compression sides of the piston, A_p is the area of the piston, A_r is the cross-sectional area of the rod, F_{drag} is the coulomb friction force between piston guide and damper body, and F_{gas} is the force due to the gas reservoir. F_{drag} is a function of the materials used for the wear band, damper body, damper rod, and main bearing, their surface finishes, and the fitment between them. For this analysis F_{drag} is neglected because it is generally very small when compared to fluid forces. F_{gas}, which depends on the pressure in the gas reservoir and the rod area upon which this pressure acts, is also neglected because it gives rise to a spring force that is independent of velocity. Thus, equation reduces to-

$$F_{damping} = P_{comp}\left(A_p\right) - P_{reb}\left(A_r - A_r\right)$$

The major resistance given by the damper is due to pressure head loss in oil when oil flowing about orifice. The formulae for calculating head loss are as follows. Thus by substituting the selected dimensions of damper assembly the pressure head losses were calculated with following formulae

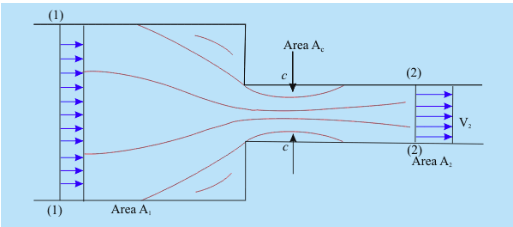

Fig 4: Fluid flow sudden contraction

$$h_L = \frac{V_2^2}{2g}\left[\left(\frac{A_2}{A_c}\right)-1\right]^2 = \frac{V_2^2}{2g}\left[\left(\frac{1}{C_c}\right)-1\right]^2$$

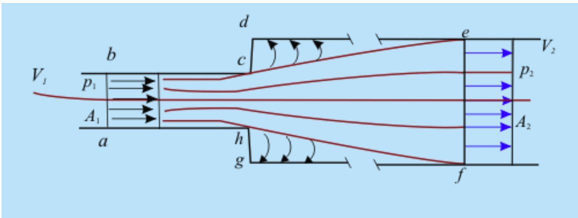

Fig 5: Fluid flow sudden enlargement

$$h_L = \frac{(V_1 - V_2)^2}{2g} = \frac{V_1^2}{2g}\left[(1\text{-}\left(\frac{A_1}{A_2}\right)\right]^2$$

2.1 Piston- Disc Valve Geometry

To allow the required area variation, the assembly of the piston in hydraulic damper can be modified as shown in Fig 5. It shows the section view of the damper; a disc valve is incorporated within a two-piece piston. The orifices on the disc can be rotated out of phase with the orifices on the main piston, allowing the effective flow area of the valve to be controlled via external actuation.

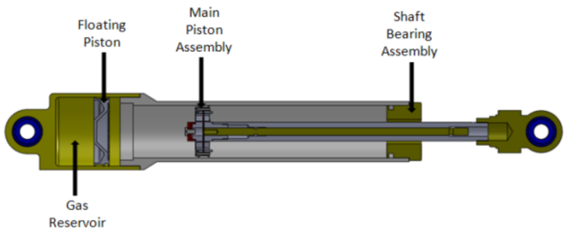

Fig 6: The damper section view

The valve design is shown in further detail in Figures 3.3 and 3.4, each depicting section views of the piston assembly, the first of which includes the control rod that rotates the internal disc. The rebound and compression decoupling is important because desired rebound and compression damping often differ considerably. This decoupling maybe achieved by using the thin shims.

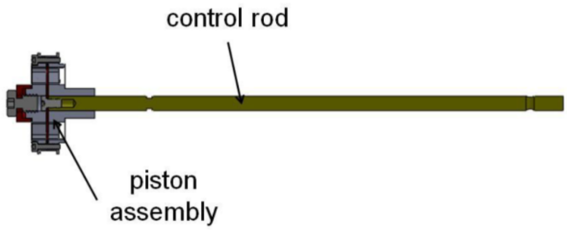

Fig 7: Piston Assembly

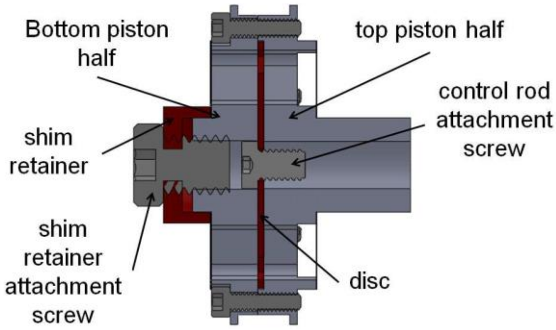

Fig 8: Piston Assembly without control rod

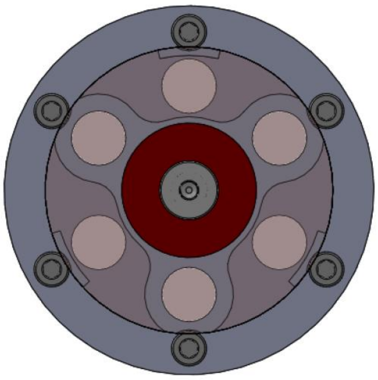

Fig 9: Piston Assembly front view

The formula derived for above calculation is as foll0ws-

$$Area\ Open(compression\ or\ extension) = 3 * \left[2 * r^2 * cos^{-1}\left(\frac{d}{2 * r}\right) - \frac{d * \sqrt{4r^2 - d^2}}{2}\right]$$

$$d = 2 * R * \sin\left(\frac{\theta}{2}\right)$$

Where, d= Distance between the hole on piston and respective hole on disk,
R=Radius at which the center of hole is situated on piston,
θ= The angle of disk turned with respect to damper piston
r= Radius of hole on piston (or orifice)

3. SIMULATION

The input data for the simulation is as follows
Piston diameter= 40 mm,
Hole diameter on piston= 8 mm and 10 mm
Position of hole diameter from piston center=28 mm,
Angle= 0^{o} to 24^{o}

Table -1: Results

Relative Position (Angle)	Hole Radius, r= 3 mm	Hole Radius, r= 4 mm	Hole Radius, r= 8 mm
0	84.8230016469244	150.796447372310	235.619449019234
1	76.0363889860067	139.075281816393	220.964706726709
2	67.3090830285452	127.398890302516	206.346149999856
3	58.7015461609999	115.812512305351	191.800191321587
4	50.2766764436387	104.362345792499	177.363705643527
5	42.1013687206500	93.0960993851025	163.074283955891
6	34.2485950996590	82.0636437788007	148.970515879334
7	26.8004206616365	71.3178185359942	135.092315040855
8	19.8527714905555	60.9154785510896	121.481305386694
9	13.5237784318741	50.9189151162581	108.181293471286
10	7.97050624444608	41.3978814665410	95.2388626600194
11	3.43054101494638	32.4326425714503	82.7041428371896
12	0.386690224332264	24.1188829605926	70.6318387786262
13	0	16.5763207789880	59.0826520479661
14	0	9.96580819250863	48.1253266481196
15	0	4.53067030422575	37.8397367753296
16	0	0.746744015768744	28.3218390834275
17	0	0	19.6922830276123
18	0	0	12.1132029996445
19	0	0	5.82747150048806
20	0	0	1.28943888319652
21	0	0	0
22	0	0	0
23	0	0	0
24	0	0	0

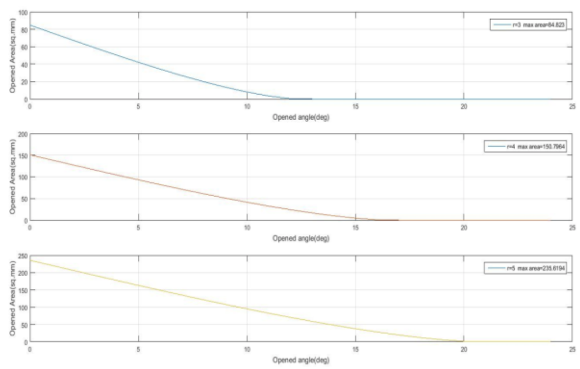

Fig 10: Orifice area variation wrt disc valve position

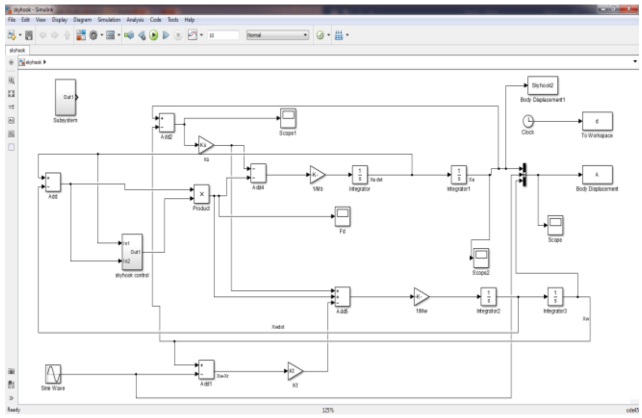

Fig 11: MATLAB Simulink model for finding damping force wrt displacement and velocity of piston

The results for damping force versus velocity of piston are given below-

The input data for the simulation is-

Density of damper oil = 860 kg/m^3,

Height of bump = 100 mm

Length of bump = 800 mm

Static Pressure = 1 atm=1.0132 bar

Disc position= 9^{o},

Input Frequency = 5 Hertz

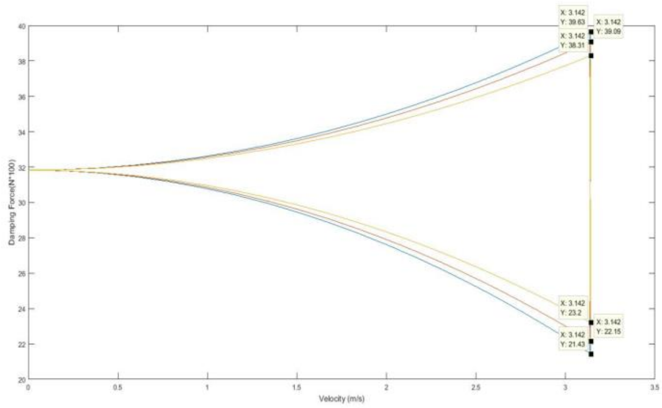

Fig 12: Damping force Vs. velocity of piston

The results for damping force versus displacement of piston are given below-

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Disc position= 9^{o},

Input Frequency = 5 Hertz

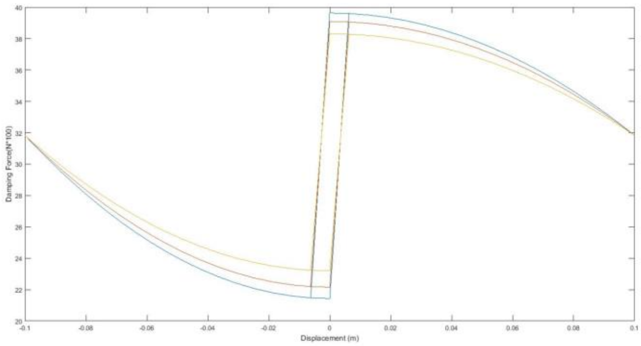

Fig 13: Damping force Vs. displacement of piston

4. CONCLUSIONS

4.1 Area opening variation with disc position for various orifice diameter

The change in disc position by rotating disc (changed relative position) changes the orifice valve area non-linearly. The change in the area thus is second degree curve, or the area is function of the square of the position angle

$$Valve\ Area, A = f(\theta)$$

4.2 Damping force Vs. velocity of the piston

As an example, for one of the case for disc position θ= 10°
In Compression cycle- for maximum velocity of piston
Hole radius r= 3 mm, Maximum damping force= 3963 N
Hole radius r= 4 mm, Maximum damping force= 3909 N
Hole radius r= 5 mm, Maximum damping force= 3831 N
In Extension cycle- for maximum velocity of piston
Hole radius r= 3 mm, Maximum damping force= 2143 N
Hole radius r= 4 mm, Maximum damping force= 2215 N
Hole radius r= 5 mm, Maximum damping force= 2320 N

4.3 Damping force Vs. velocity of the piston

The results obtained for various disc positions with circular hole were simulated in this study simulation.

The Orifice area geometry along with disk hole geometrical features can be modified to get more linear valve area (orifice) variation.

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