Design and Evaluation of Hydro-Pneumatic Friction Damper Suspension System

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ABSTRACT

Perceived comfort level and ride stability are the two most important factors in the evaluation of suspension system in a mobile vehicle. It is extremely difficult to simultaneously maintain a high standard of vehicle ride, handling and body control in the vehicle by using conventional passive suspension system. However, the use of active suspensions would result in better comforts than the passive ones. This paper presents the design and analysis of a pneumatic friction damper and hydro-pneumatic friction damper. A non-linear quarter car model is developed, which includes pneumatic actuation by pressure regulation. The performance of the proposed model was assessed in terms of level of vibration reduction. Simulations on a prototype model show that the proposed system has good performance and robustness.

Keywords: *Active Suspension, Friction Damper, Hydraulic Damper, Pressure Regulation, Vibration Isolation*

1. INTRODUCTION

The main functions of a vehicle suspension system are to isolate the body from road unevenness disturbances and to maintain the contact between the road and the wheel. Therefore, the suspension system is responsible for the ride quality and driving stability. The design of a passive suspension system is a compromise between this conflict demands. However, the improvement in vertical vehicle dynamics is possible by developing active suspension system. In recent years, the development of pneumatics controlled suspension dampers and actuator has increased the research on vehicle safety versus ride comfort trade off. In order to maintain the level of comfort for passengers and drivers, and still maintain the high safety standards of automobiles, suspension designers have been forced to look beyond the conventional suspension systems.

Crosby and Karnopp (1974) originally proposed the basic concept of semi-active damping and then the use of semi-active dampers in automobiles has been studied extensively. The authors have contributed an excellent review of

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many of the past efforts in the area of semi-active suspension design and provided a background of the information that is required to understand semi-active suspension systems. A review of the state-of-the-art of controlled suspensions has been carried out by Hedrick and Wormely (1975) and by Goodall and Kortum (1983). In these investigations, semi-active control and linear optimal control with full state feedback along with simple on-off control strategies were employed to reduce both the tyre force and body acceleration of a heavy truck. However, it is worth mentioning here that the air springs for passenger cars are commercially available and there are not enough researches devoted on their performances. The work presented by Quaglia and Sorly (1996) discussed the vehicular air suspensions from design aspects, but not from control viewpoint. Considering the market requirement for passenger vehicles, it was found that integration of leveling, variable damper control technology and stiffness control are very beneficial. The new active friction damper provides a possible solution for the problem.

There are two basic types of suspension namely, the primary suspension used between the vehicle chassis and axle assembly and secondary suspension system mounted between the vehicle body and seat. Much work has-been reported (Reynolds, 1993; Rayliegh, 1945) regarding the primary and secondary suspension systems. The study carried out by Williams (1997) classified the active suspensions as the high-bandwidth (fast active) and low bandwidth (slow active). High bandwidth active hydraulic actuators control the body motion and wheel motion. On the other hand, low bandwidth suspensions employ pneumatic actuators to control the body motion while the wheel motion is controlled by conventional passive spring and damper. Over the past three decades, many control methods for active suspension have been studied. Review studies by Clarr and Vogel (1989) and Sharp and Corolla (1987) summarized the various commonly used control techniques. More recently, controls based on different approaches have been employed in active suspension system. These include approaches based on linear and non-linear control (Gao et al., 2006; Hong et al., 2002; Elimadany & Abdlizabbar, 1999), optimal control (Elbheiry & Karnoop, 1996) and modern robust control such as H-infinity (Palmeri et al., 1995; Stribrsky et al., 2002; Wang et al., 2001). In the past, Fuzzy logic based active and semi-active suspension system (Kashani & Strelow, 1999) was also being employed for the control purpose. Among them, skyhook control (Hong et al., 2002) is the most important concept considered for active suspensions.

The main aim of this paper is to discuss the investigations carried on the pneumatic friction and hydro-pneumatic friction dampers. In this approach, laboratory prototype models of the aforementioned friction dampers have been developed. In order to study the complete system behavior, the developed prototype models have been interfaced to Lab VIEW software module with laser displacement pickup. The details of the key design features and test results were presented. The pressure control regulator controls the friction load coming to the damper is considered in the present model. The non-linearity behavior of the friction pads and pressure developed at the axial hole on the piston for different loads are investigated.

2. DRY FRICTION DAMPING

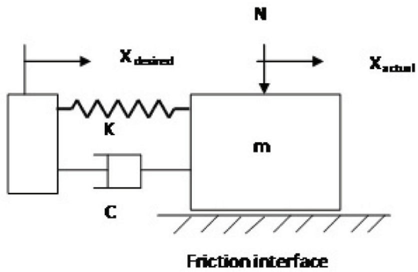
Coulomb or dry friction damping results when sliding contact exists between two dry surfaces. The damping force is equal to the product of the normal force and the coefficient of dry friction. The friction force is always opposite to the direction of motion. The friction force is given by:

$$F = \mu N \tag{1}$$

where μ is coefficient of friction, F is frictional force, and N is normal reaction. Frictional force divided by the velocity of the piston gives frictional damping.

The friction model is shown in the Figure 1. Friction has often been modeled by an algebraic

Figure 1. Friction model



equation relating velocity and normal force to friction. It is well known however that friction possesses dynamics associated with varying velocity. A rather complete description of friction modeling and its impact on control is well documented in the literature (Armstrong et al., 1994). However, it was recognized that dynamics associated with varying normal force play an important role in the system response. Friction dynamics associated with variations in normal force are fast in comparison to those due to velocity fluctuations (Dupont, 1993; Guglielmino & Edge, 1980). It is observed that pure dry friction characteristics' are of no practical use because of their non-linearity, but a controlled friction damper can be made to behave in a variety of ways emulating spring-like pseudo, viscous characteristics. The external force (F_{ext}) required by an active friction damper is generally defined as,

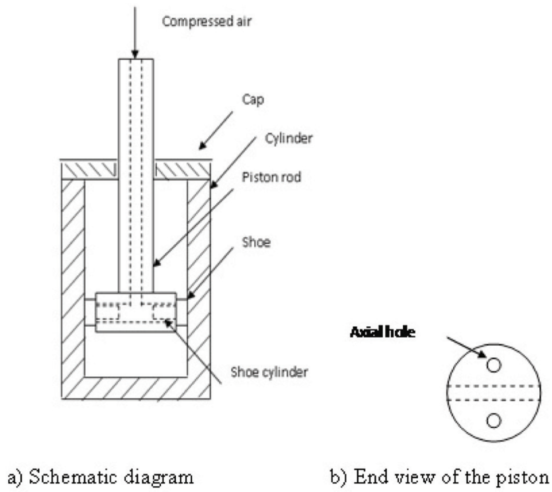
$$F_{ext} = M\ddot{x} + C\dot{x} + Kx + \sum_{i=1}^{m} \mu N_i \qquad (2)$$

here M is the mass in Kg, C is the damping coefficient, K is the spring stiffness, N_i is the normal force of i^{th} friction damper and μ is the friction coefficient. When C = 0, the system work as pure friction damper

In this work, the controlled damping element is modeled as a frictional damper. This device, which conceptually is composed of a plate fixed to a moving mass and a pad against it. An external normal force is applied to a mass by the pad and consequently, in the presence of relative motion between the pad and the plate, a frictional damping force is produced.

The pneumatic friction model is depicted in Figure 2. The model mainly consists of a conventional spring and piston-cylinder assembly system. A friction damper was constructed in the piston such a manner it enable to work as pure friction damper and friction damper with a conventional viscous damper used in a vehicle. In the piston two shoes are provided which activated by the compressed air. When the pressure acts on the shoe cylinder, shoe displace towards the cylinder surface with force is equal to shoe cylinder area multiplied by supply pressure. The friction force acting on the surface of the cylinder is μ times the applied force on the shoe. Compressed air is supplied through the centre of the piston rod and pressure is controlled by the pressure regulator. On the axial direction on the piston two 2 mm holes are provided for Hydraulic damping purpose. The damper can be used as pure friction damper without filling the oil and Hydro-friction damper with filling the oil inside the cylinder. External energy supplied by the air compressor regulates the pressure; which actuates the friction pad, according to variable static and dynamic load of the system. The friction pads undergo dry friction damping. In the present investigation, coulomb damping is considered. When the vehicle is in static state i.e., the vehicle with dead weight, a radial pressure is imparted on the friction pads, which sustain the sudden shock when

Figure 2. Pneumatic/hyrdo-pneumatic damper



the rider sits on it. This causes friction pads to expand and recoil slowly. On the other hand, when the vehicle is in motion, the suspension system is active type, i.e., both linear and radial pressures are acting on it.

Figure 3 shows the photograph of the fabricated damper with spring assembly. A 5 kg weight is added on the top of the platform and it is displaced to 5 mm step input and the response curves were obtained. Vibration levels are measured using laser pickup and Lab VIEW software and analyzed through the origin software. Laser pick up is mounted on the top of the damper to measure the displacement. Figure 4 shows the 2D drawing of the friction damper model and Figure 5 shows the experimental setup of the friction damper.

3. RESULTS AND DISCUSSIONS

Friction in the actuator was represented by coulomb friction and velocity-dependent terms obtained from previous study (Gao et al., 2006). Experiments were performed with different pressures supplied to the friction pad for step input and results were obtained using the Lab VIEW software. The damper is actuated without lubricant oil inside the cylinder. The dynamic response in terms of the variation of displacement with respect to time obtained is illustrated in the Figure 6. X axis the time is in mille seconds and Y axis is the displacement in mm.

It is observed from Figure 6 that the behavior of system is highly non-linear. Moreover, when the applied pressure is low (2.22KN/m^2), due to harshness of the friction pad, the system can exert the damping effect. Hence, it is observed that the displacement sharply decreases with time (Figure 6a). On the other hand, for increased applied pressure, the friction pad initially exerts large friction and hence the system exhibits an increased displacement with time. However, as time progressed, the damping effect improves which turn reduces the displacement as depicted in Figure 6(b-d).

The hydro pneumatic friction model analysed in the study is similar to pneumatic friction model. However, in the hydro pneumatic friction model, oil is employed in the cylinder and two axial holes are provided in the piston. Thus, there are two dampers provided in the system, one is the pneumatic friction damper provided by friction pad and the other is the hydraulic damper due to oil flow in the cylinder through the hole provided in the piston. When the piston is in compression, part of the

Figure 3. Photograph of the fabricated hydraulic –pneumatic damper

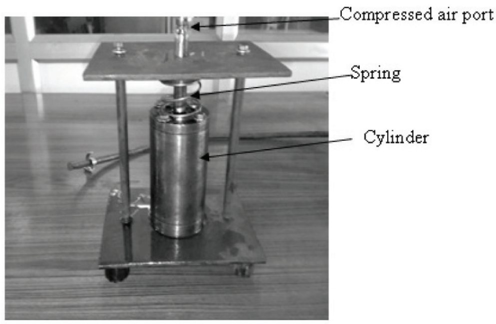
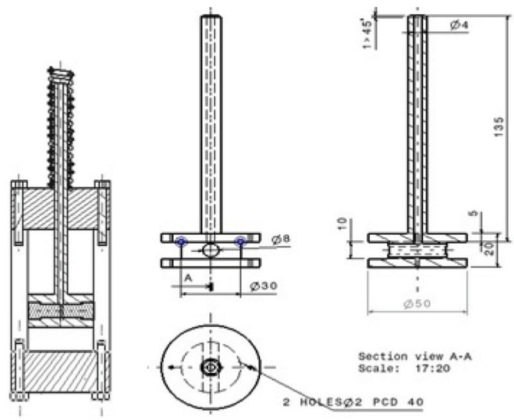


Figure 4. 2D drawing of the friction damper model



oil flow from cylinder bottom chamber to the upper chamber through the orifice.

For analytical modeling of the hydraulic damper, consider piston area A_p and area of opening of the axial hole A_0, when piston displaced by the action of dynamic force, the quantity of fluid displaced is:

$$Q_p = A_P \frac{dy}{dt} = A_p V_p \tag{3}$$

where y is the displacement of the piston and V_p is the velocity of the piston moment.

The fluid flow rate, Q_0 through the axial hole can be obtained using continuity equation and momentum equation as:

$$Q_0 = A_0 \sqrt{2g \frac{(p_1 - p_2)}{\rho}} = A_P V_P \tag{4}$$

Where p_1 and p_2 are upstream and downstream pressure ρ is the specific weight of the fluid and g is the gravitational constant.

The force carried by the piston is equal to area of piston multiplied by difference in pressure and fixed hydraulic damping coefficient is equal to force acting on the piston divided by velocity of the piston. By manipulating equations (3) and (4) the damping coefficient, $C = \frac{F_P}{V_P} = \frac{A_P(p_1 - p_2)}{V_P}$ can be obtained as:

Figure 5. Experimental model of friction damper

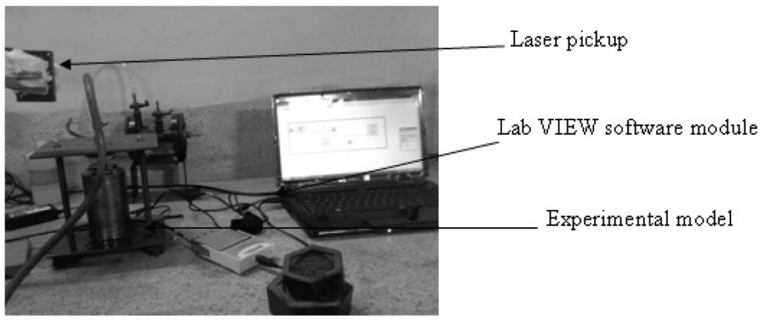
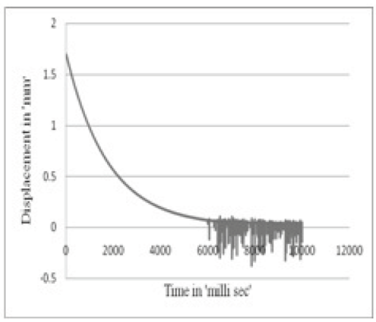
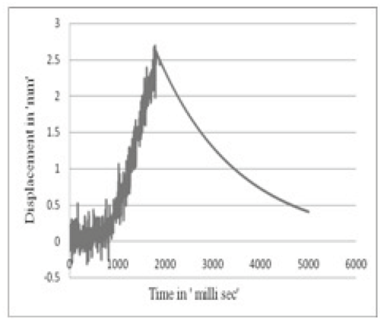


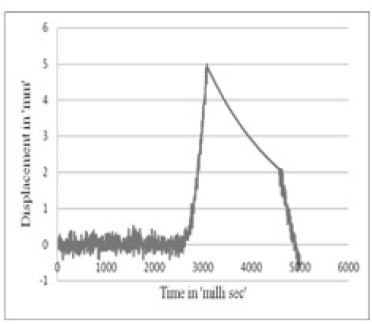
Figure 6. Dynamic response of the pneumatic friction damper



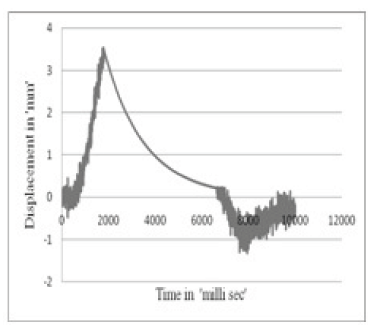
a) Applied pressure 2.22KN/m^2



b) Applied pressure 4.45KN/m^2



c) Applied pressure 6.68KN/m^2

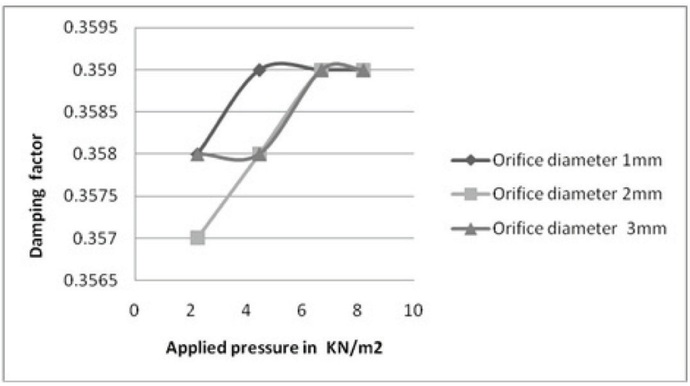


d) Applied pressure 8.92KN/m^2

Table 1. Damping factor for different orifice diameter

Applied pressure P_1 K N/m^2	Orifice diameter, d_o=1mm, calculated Damping factor	Orifice diameter, d_o=2mm, calculated Damping factor	Orifice diameter, d_o=3mm, calculated Damping factor
2.229	0.358	0.357	0.358
4.459	0.359	0.358	0.358
6.688	0.359	0.359	0.359
8.198	0.359	0.359	0.359

Figure 7. Plot of pressure Vs damping ration for different diameters



$$C = \frac{{A_P}^3 V_P \rho}{{A_0}^2 2g} \qquad (5)$$

where FP is the viscous force which can be obtained from the force acting on the piston and $V_P = \frac{F_P}{A_P}$. Using the equation (1) and (5) theoretical damping factor due to friction and hydraulic damping can be determined. The damping factor of the orifice is determined from

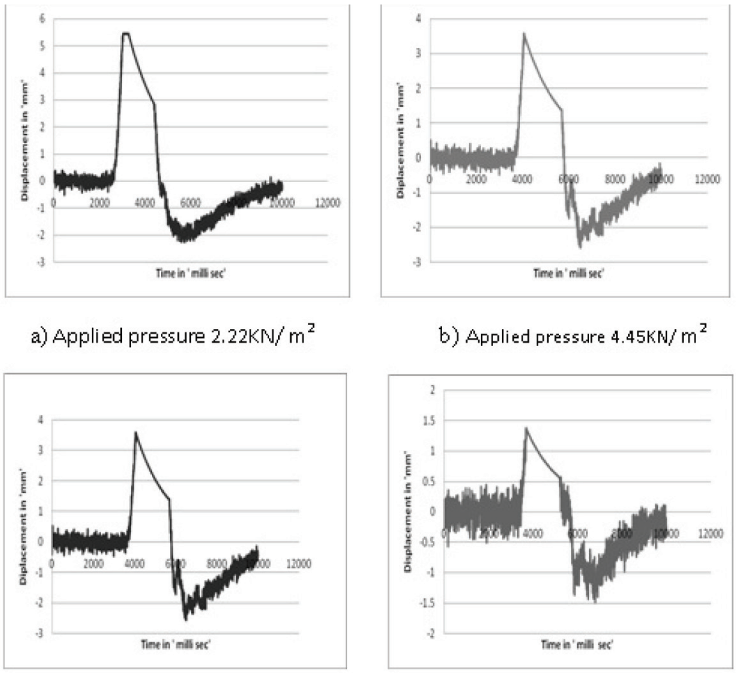
$$\zeta = \frac{C}{C_c} \qquad (6)$$

where, C is the actual damping factor and C_c is the critical damping factor. Table 1 gives the damping factor for different axial hole diameter on the piston for different pressures.

The theoretical damping factor values were determined as a function of applied pressure for different orifice diameters is depicted in Figure 7. It is observed from Figure 7 that the system exhibits linear responses for different applied pressure for 2mm orifice diameter. Hence, 2 mm orifice diameter was considered in the prototype design.

With the above design features in the prototype model, dynamic response of the system is obtained and is shown in the Figure 6 and Figure 8. Two important observations can be drawn from the dynamic response shown in Figure 6 and Figure 8. It is observed that the system exhibits similar responses for different applied pressure. Also, the hydro pneumatic friction model takes more time with reduced displacement as compared to pneumatic friction model. Therefore hydro pneumatic friction model provide larger damping effect with re-

Figure 8. Dynamic response of the hydro-pneumatic system for different loads



a) Applied pressure 2.22KN/ m^2

b) Applied pressure 4.45KN/ m^2

c) Applied pressure 6.68KN/ m^2

d) Applied pressure 8.92KN/ m^2

duced vibrations and better riding comfort as compared with pneumatic friction model.

From the above discussions, it is seen that the experimentation on the pneumatic friction damper system showed an unexpected behaviour at low supply pressure, dynamic performance was extremely poor. The first hypothesis made was that at low pressure an air pocket could have been trapped inside the valve: at low pressures, a small quantity of air enormously reduces bulk modulus, which adversely affects the dynamical response of the system. Hence it can be that the pure pneumatic friction damper is not suitable for better rider comfort.

CONCLUSION

The details of the investigations carried out on pneumatically activated friction damper and hydro pneumatic friction damper models are presented in this paper. Laboratory prototype models were developed and dynamic testing performance was carried out by means of LabVIEW software module with laser pick up. The details of the key design were provided. The dynamic test results illustrated that the hydro-pneumatic friction damper model exhibits better damping performance as compared to pneumatically activated friction damper.

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