THE DEVELOPMENT OF A SEMI-ACTIVE TRUCK DAMPER

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ABSTRACT

This paper describes the development of a continuously-variable, semi-active damper for heavy commercial vehicles. A high speed proportional valve allows the damping coefficient to be varied continuously. The influence of oil flow forces on the valve spool position control are investigated. A mathematical model of the damper is described and simulated results are compared with measured responses. The step response of the damper is measured and the factors affecting speed of response are determined. The force tracking performance of the semi-active damper is examined under realistic operating conditions using a Hardware-in-the-Loop testing method.

1   INTRODUCTION

The concept of semi-active damping was originally proposed by Crosby and Karnopp [5] in 1973, and since then the use of semi-active dampers in automobiles has been studied extensively, both theoretically and experimentally. Experimental studies [6-8, 12] have shown that semi-active dampers offer benefits in ride comfort and road holding, compared to passive dampers. However, time delays, valve dynamics, fluid compliance and friction have been found to significantly affect damper performance.

The application of semi-active dampers to heavy vehicles has received relatively less attention. Experimental tests on a Military tank equipped with semi-active dampers were reported by Miller and Nobles [10]. Their twin tube damper incorporated voice-coil actuated on-off valves at either end of the piston chamber. Over the duration of testing, a large number of mechanical failures occurred but some improvements in ride comfort were reported. An experimental investigation of a two-state damper for off-road heavy vehicles was described by Nell and Steyn [11]. The semi-active damper used an externally mounted on-off control valve consisting of a pressure dependent two-way cartridge valve combined with a spool valve. Tests revealed that the valve could switch states before any significant change in damper force was measured. Besinger et al used a

'Hardware-in-the-Loop' (HiL) test rig to assess experimentally the performance of two different semi-active dampers. A prototype semi-active heavy vehicle damper which used an external stepper motor to rotate an internal valve gave unsatisfactory performance due to oil flow forces acting on the valve [2]. A continuously variable automobile semi-active damper was used to represent a truck damper by multiplying the force measured in the HiL rig [4]. The damper was found to give significant reductions in calculated road damage and body acceleration, compared to optimum passive damping.

None of the dampers considered in these studies was able to provide continuously variable damping suitable for heavy commercial vehicles. The objective of the work described in this paper was to develop a continuously variable semi-active damper specifically for heavy vehicles. The prototype damper is being used in a programme of research to assess experimentally the benefits of semi-active damping in terms of road damage, ride comfort, working space, road holding and roll-over stability.

2   PROTOTYPE DAMPER DESIGN

The hydraulic circuit for the damper is shown schematically in **Figure 1**. A high speed proportional valve controls the pressure drop through the circuit, thereby varying the damping coefficient. The arrangement of check valves provides unidirectional oil flow around the circuit, simplifying the requirements for the proportional valve. A reservoir accommodates the oil displaced by the volume of the piston rod and is pressurised to prevent cavitation occurring.

Figure 2 shows a cross-section of the damper body and the proportional valve. The filter, reservoir, a check valve and connecting pipe-work are not shown in this figure. In a production version, these components would be incorporated into the damper body. The solenoid operated proportional valve consisted of three main components: (i) a spool moving in the valve body to provide a variable orifice area. (ii) a *solenoid* to move the spool against its centralising springs. (iii) a *displacement transducer* to measure the spool position.

The structure of the semi-active damper controller is shown in **Figure 3**. The inner loop has a PID controller which acts to control the spool position and thereby control the valve orifice area. Control of the force generated by the damper is achieved by the PID controller in the outer loop. Initial tests showed that oil flow forces acting on the spool significantly affected the spool position. Improvement in spool position control was achieved by the addition of a solenoid current proportional to the oil flow rate.

3   DAMPER PERFORMANCE

3.1   Mathematical Modelling

Nonlinear mathematical models of the damper and its proportional valve were developed to predict the performance of the semi-active damper. The valve model relates solenoid current (I_{sol}) to spool position (X_{spool}). The solenoid, which converts current to force, is a nonlinear device [13]. The nonlinear solenoid model was linearised to aid design of the valve spool position feedback controller as shown in Figure 3. The spool and solenoid core were modelled as a point mass moving against a linear spring and damper. The mass and stiffness values were measured. The damping coefficient was chosen to give good agreement between measured and simulated spool position responses.

The valve model was validated against measured step responses and transfer functions (solenoid

current to spool position). The valve spool PID controller gains were selected to provide fast response without significant overshoot. **Figure 4** shows the measured and simulated closed loop step responses for five different demanded spool positions. Agreement between measurement and simulation is very good. All the responses have an initial 3ms delay, which arises from the power amplifier in the controller.

The damper model relates damper force (F_{damper}) to the spool position (X_{spool}) and damper input velocity ($\dot{X}_{damper}$). The three components of the model are depicted in **Figure 5** and are:

1   The combined flexibility of the nylon mounting bushes and the mounting frame were modelled as a cubic spring.

2   The compressibility of the oil was modelled using the relationship between bulk modulus and air-oil volume presented by Audenino and Belingardi [1]. Following their experiences of hydraulic dampers, an air to oil percentage of 2.5% by volume was assumed.

3   The relationship between valve spool position and orifice area was modelled as a circular sharp edged orifice [9]. Measurements of pressure drop, flow rate and spool position were used to validate the model.

3.2   Measured Performance

The damper was tested by connecting the piston rod to a high bandwidth servo-controlled hydraulic actuator. The damper body was mounted to a rigid framework and a load cell measured the damper force. The PID gains of the damper force controller were initially set with the aid of the mathematical model and then adjusted to give optimum measured step response. **Figure 6** illustrates the measured and simulated force responses for two consecutive demand steps of 10kN, with the damper input velocity at a constant 150mm/s in extension. The measured and simulated responses are in good agreement, but both show that there is approximately 25ms delays before the force rises to the demanded level. The mathematical model was used to show that the delay is caused mainly by the compression of the air-oil emulsion.

In order to test the damper under more realistic operating conditions, a 'Hardware-in-the-Loop'(HiL) test rig was used as described by Besinger *et al* [4]. The HiL tests involved a two degree-of-freedom 'quarter truck' model with parameters representative of a laden air-sprung heavy vehicle, as detailed in Table 1. The model was excited with artificially generated random road profiles as described in [3].

Quantity	Parameter	Value	Unit
Sprung Mass	M_S	4450	[kg]
Unsprung Mass	M_u	420	[kg]
Suspension Stiffness	k_S	0.5	[MN/m]
Tyre Stiffness	k_t	1.95	[MN/m]

Table 1 HiL quarter car model parameters.

A clipped Modified Skyhook Damping (MSD) control strategy was used as described in [4]. In this strategy the demand damper force (F_{demand}) is given by:

$$F_{demand} = C_m \left[\alpha V_{rel} + (1 - \alpha) V_{body} \right] \qquad \text{if} \qquad F_{demand} V_{rel} > 0 \qquad (1)$$

$$F_{demand} = 0 \qquad \text{if} \qquad F_{demand} V_{rel} \leq 0 \qquad (2)$$

Figure 7 illustrates the force tracking performance of the damper, for a Minor Road input at a speed of 14m/s. The dominant frequency is 15Hz, corresponding to the wheel hop mode of vibration. It is apparent from this figure that the measured force is often greater than the demanded force when the damper is *extending,* whereas the measured force is often less than the demand when the damper is *contracting*. This is because the volume of oil in compression when the damper is *contracting is* approximately three times that when the damper is *extending*. The larger volume of oil results in increased time for the demand force to be achieved. The damper force PID controller gains were chosen to achieve a satisfactory compromise between the two directions. It is thought that the force tracking performance shown in Figure 7 is sufficient to realise the theoretical benefits of semi-active damping. This is the subject of on-going research.

4   CONCLUSIONS

1   A continuously variable heavy vehicle semi-active damper has been successfully developed and tested.

2   Linear and nonlinear mathematical models of the damper were developed and validated. The models confirmed that compressibility of the hydraulic oil significantly influenced damper response.

3   Satisfactory force tracking performance of the semi-active damper was achieved under realistic input conditions using a HiL test rig.

Work is underway to assess the performance of the semi-active damper in terms of vehicle responses such as road damage and ride comfort.

5 ACKNOWLEDGEMENTS

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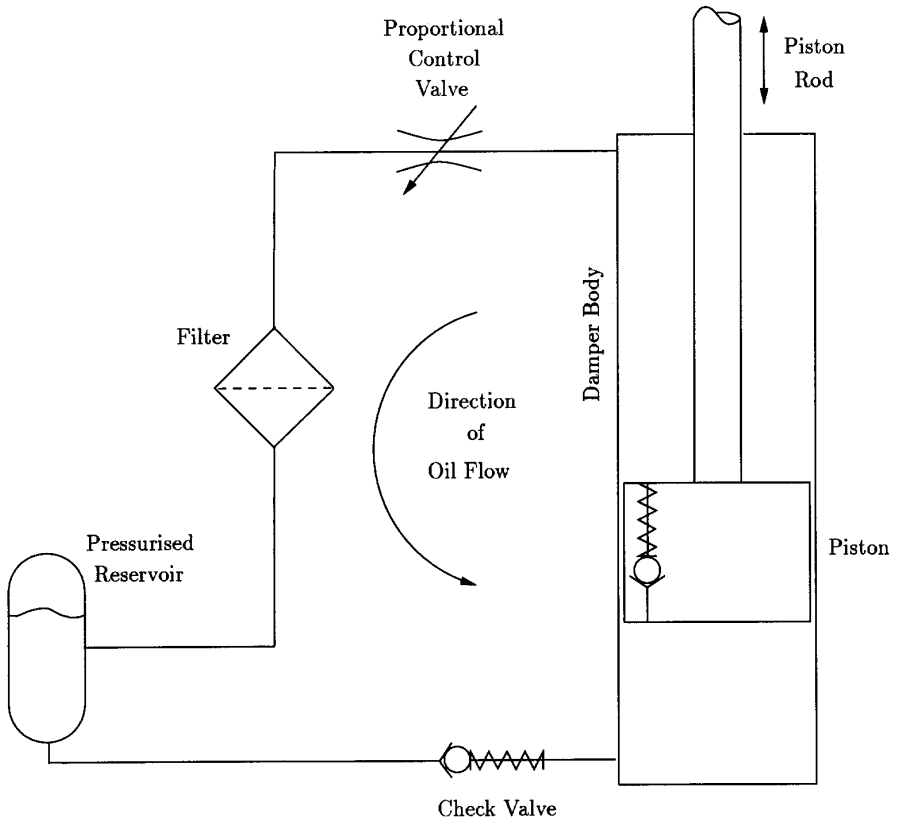
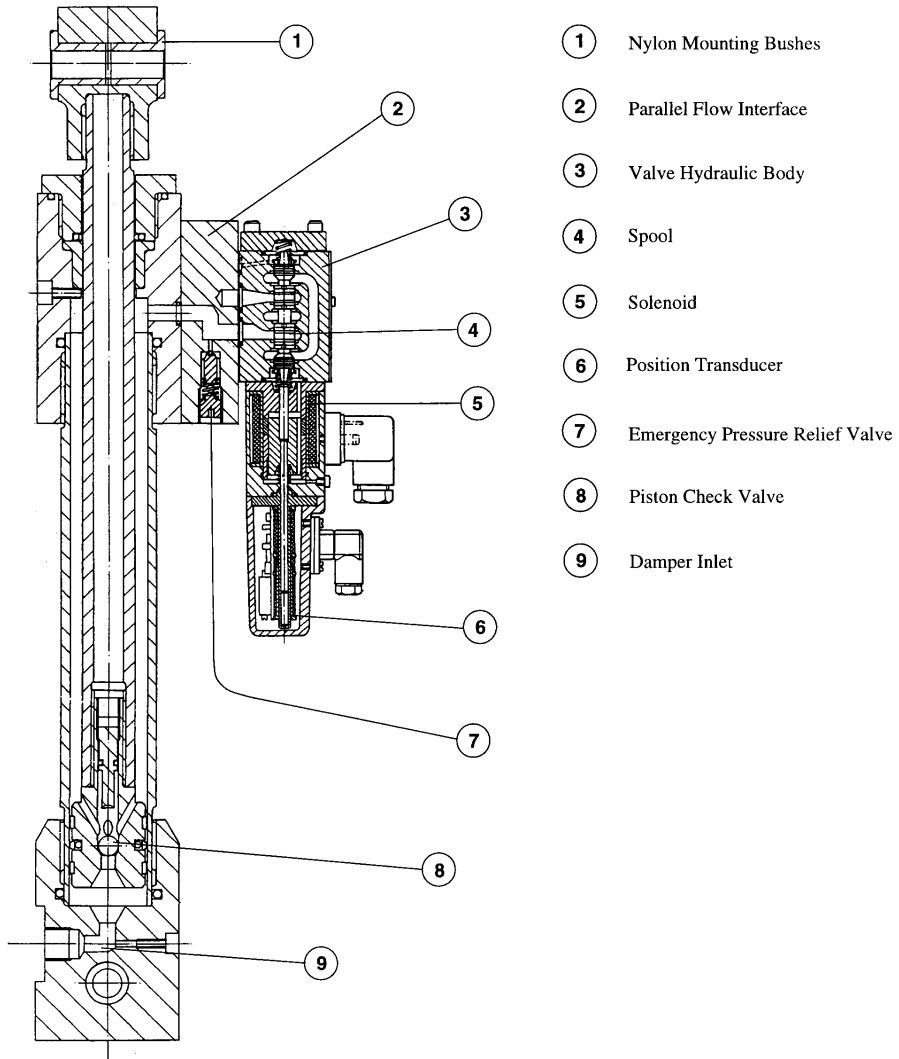**Fig. 1 Hydraulic Circuit of the Semi-Active Damper.**

1	Nylon Mounting Bushes
2	Parallel Flow Interface
3	Valve Hydraulic Body
4	Spool
5	Solenoid
6	Position Transducer
7	Emergency Pressure Relief Valve
8	Piston Check Valve
9	Damper Inlet

**Fig. 2   Cross-Section of the Semi-Active Damper
(reservoir, filter, check valve not shown).**

Key :-
F_{demand} = Demand Damper Force
X_{demand} = Demand Spool Position
I_{sol} = Solenoid Current
X_{spool} = Achieved Spool Position
F_{damper} = Achieved Damper Force
$\dot{X}_{damper}$ = Damper Velocity
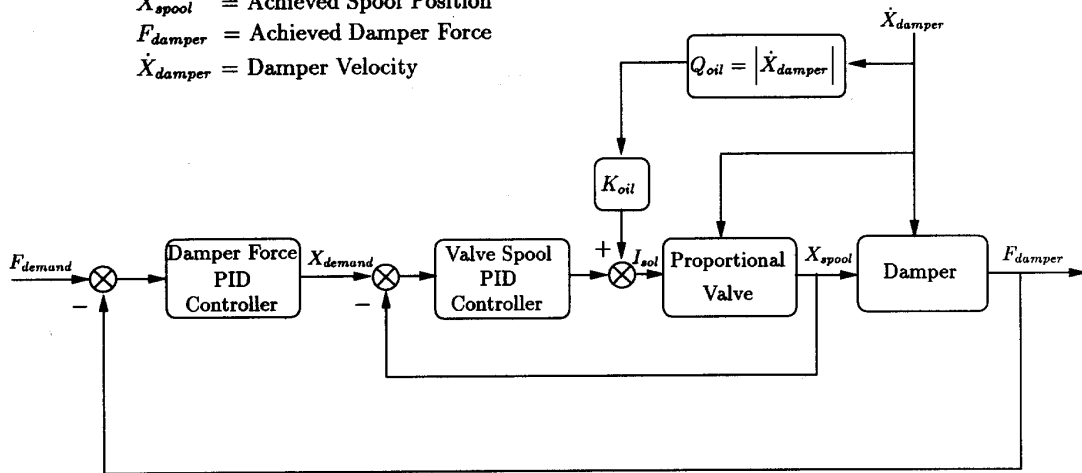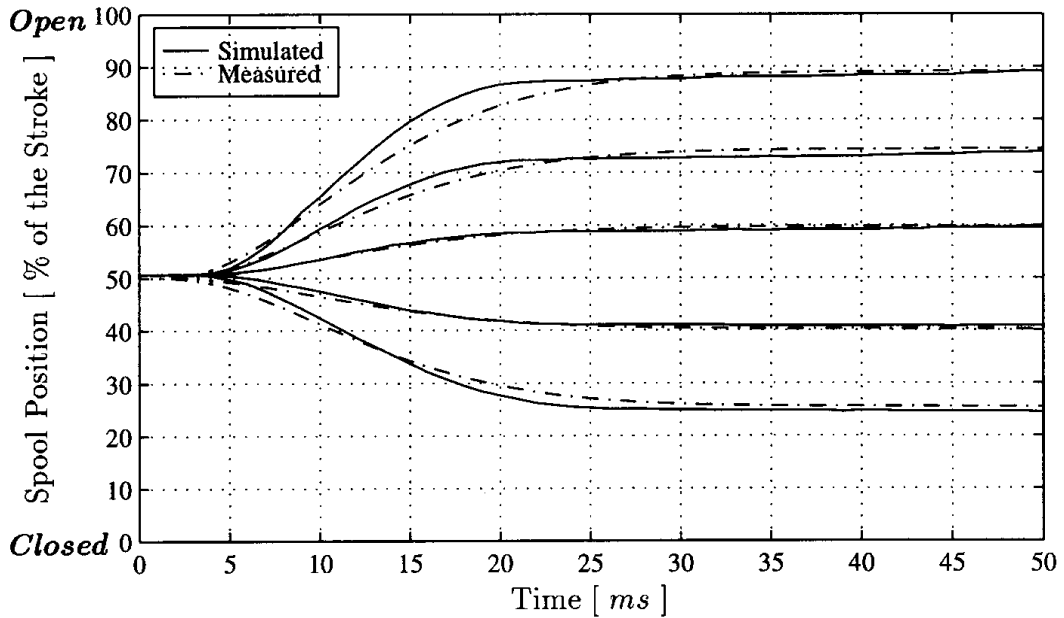

Fig. 3   Feedback Control of the Semi-Active Damper.

Fig. 4   Closed loop step responses for the valve with a constant damper velocity of 150mm/s. (Demand signals removed for clarity.)
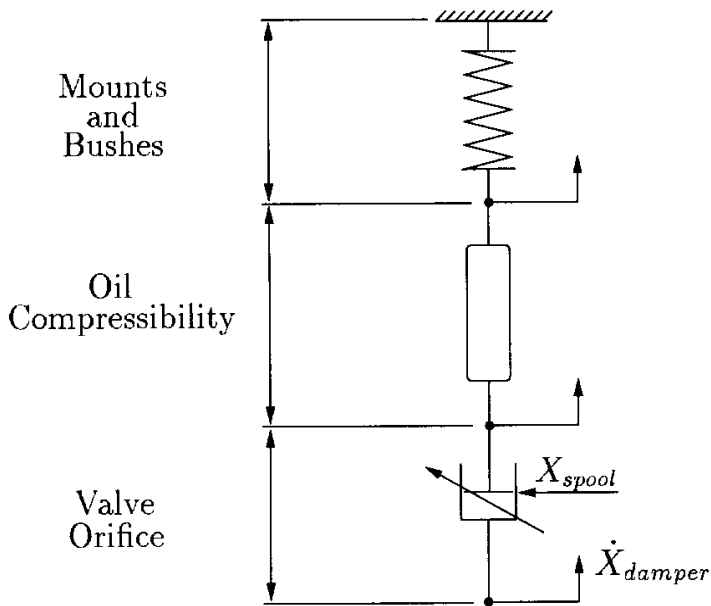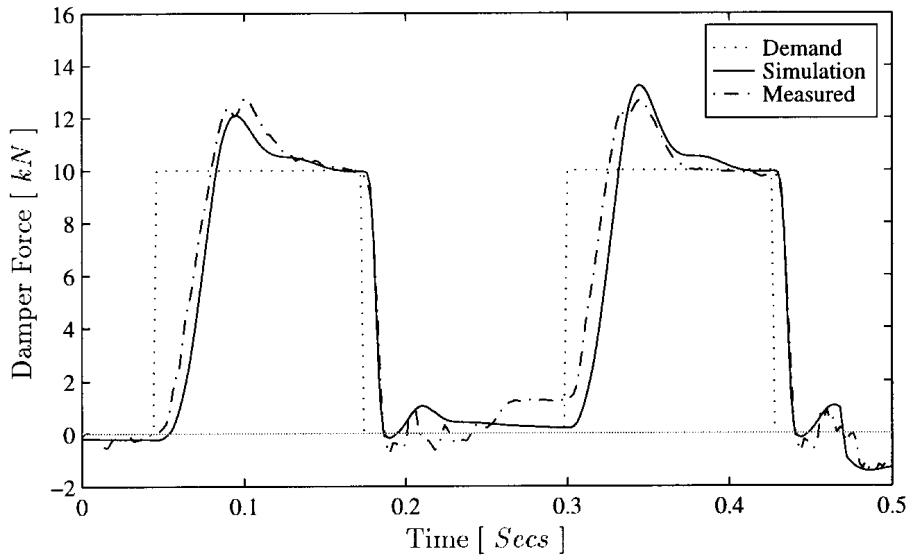
Fig. 5 Schematic of the simple damper model.

Fig. 6   10kN step force tracking with a constant damper velocity of 150mm/s.
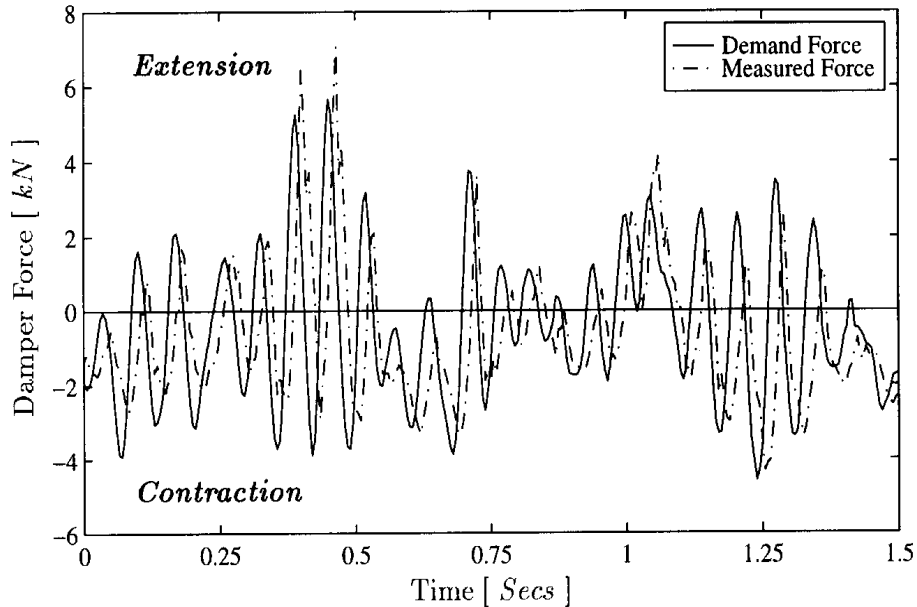
Return
to paper

Content

Fig. 7 HiL Measurement for a Minor Road at 14m/s using MSD Control with C_m = 24kNs/m, a = 0.8.