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Developments in Semi-Active Heavy Vehicle Suspensions

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ABSTRACT

A prototype semi-active damper is tested in a half-car 'Hardware-in-the-Loop' (HiL) rig with a planar two-axle heavy vehicle model. The benefits of preview control using the prototype semi-active damper are found to be less than theoretically possible, due to the phase lag between the demanded and achieved damping force. It is shown that the performance of the suspension can be improved significantly by compensating for the delay in the prototype damper using a strategy known as 'Phase Lag Compensation'. A new continuously variable truck damper are described, and results of initial performance tests are presented.

1.0 INTRODUCTION

There are various ways in which heavy vehicle suspensions can be improved; so as to reduce body vibration levels; minimise dynamic tyre forces (road damage); and decrease suspension travel (increase payload capacity). The most practical improvement is to use soft (air spring) suspensions, with optimised damping (Cole and Cebon, 1996). From this baseline condition it is possible to improve performance further using computer–controlled active or semi–active components (Cole, Cebon and Besinger, 1994). Of these, the most economically viable solution in the foreseeable future is use of *semi–active dampers*. These are hydraulic shock absorbers with continuously-variable damping properties (Kitching, Cole and Cebon, 2000a).

Previous work on optimising vehicle performance using semi–active suspensions has investigated several control strategies. These include the popular 'Skyhook' damping strategy, linear optimal approaches (Cebon, Besinger and Cole, 1996), the 'Extended Ground-Hook' strategy (Kortüm and Valasek, 1998), and various methods of applying preview (Kitching, Cebon and Cole, 1999). Most previous research has been performed on simulation models or laboratory experiments using 'Hardware-in-the-Loop' (HiL) testing methods. There have been very few field tests on lorries equipped with semi-active suspensions.

This paper reports on continuing research being performed by the Cambridge Vehicle Dynamics Consortium on the use of semi-active dampers and preview control for heavy goods vehicles. The Consortium currently consists of Cambridge, Cranfield and Nottingham Universities along with ten companies from the European heavy vehicle industry.

2.0 FOUR-DOF HIL TEST RIG

2.1 HiL Rig

In order to examine the performance of the prototype semi-active damper shown in figure 1a, using a 'wheel–base preview' control strategy, a two-degree of freedom (DOF) HiL test rig (Besinger, Cebon and Cole, 1995) was extended to incorporate a two axle four-DOF heavy vehicle model (see figure 2). This was representative of the tractor unit of a typical 38 tonne GVW UK lorry. Since only one servo-controlled actuator was available, just one hardware damper could be examined in a HiL configuration. The main benefits of semi-active damping and preview control can be expected to be realised at the drive axle (Kitching, Cole and Cebon, 2000b). Therefore, a theoretical linear passive suspension was used at the steer axle with the prototype hardware damper providing the damping forces at the drive axle. The time step used in the four-DOF HiL simulations was $2ms$ (giving a simulation frequency of $500Hz$).

2.2 Suspension Control Strategies

Two control strategies used to regulate the demanded force for the prototype semi-active damper are examined here:

(i) A Modified Skyhook Damping (MSD) control law (Besinger, Cebon and Cole, 1995) which assumes the inputs to the two-axles of the vehicle are uncorrelated, and demands a force proportional to a weighted combination of the sprung and unsprung mass velocities.

The demanded suspension force F_{dem} for the MSD strategy (in the case of the drive axle of the four-DOF model) can be written as:

$$F_{dem} = C_m\left[\alpha(\dot{Z}_{dw} - \dot{Z}_{db}) + (1-\alpha)\dot{Z}_{db}\right]$$

where C_m is the maximum damping coefficient, Z_{dw} and Z_{db} are defined in Fig. 2, and α is a dimensionless weighting parameter varying between 0 and 1:

$\alpha = 0$ represents *skyhook* damping,
$\alpha = 1$ represents *passive* damping.

(ii) A form of wheel-base preview control (Kitching, Cole and Cebon, 2000b; Sharp and Wilson, 1990), which assumes that the input to the drive axle is a time delayed version of the input experienced by the steer axle and requires all the states of the vehicle to be known.

The demanded control force given by the MSD and Preview controllers assumes that the suspension contains an *ideal* active actuator, capable of both supplying and dissipating power. In practice it is necessary to 'clip' the demand signal when the control law requires the damper to supply power.

3.0 RANDOM ROAD RESULTS

The performance of the four-DOF vehicle model was assessed in response to various random road profiles, and speeds. The results obtained with an optimised non-linear passive damper were used to provide a benchmark from which to gauge the performance of the prototype semi-active damper.

The performance of the four-DOF HiL model with a conventional passive suspension at the drive axle of the vehicle (using an Iveco-Ford damper) is illustrated in figure 3 (*solid feint line*). The graph presents the RMS body acceleration and the RMS dynamic tyre force. It illustrates the performance of the four-DOF model with the prototype semi-active damper. The control strategies are: (i) MSD control (*dashed line*), and (ii) Preview control (*dotted* *line*). The points denoted by $P4_{opt30}$ for the passively suspended vehicle, and $S4_{msd30}$ and $S4_{pre30}$ for the MSD and Preview controlled semi-active suspensions respectively, represent a good compromise between minimising the RMS tyre force and the RMS body acceleration.

The main benefits of the semi-active suspension are in reducing the RMS body acceleration, with minor benefits in terms of the RMS tyre force. Compared to $P4_{opt30}$ the RMS body accelerations at the drive axle were reduced by 9.1% and 15.4% for $S4_{msd30}$ and $S4_{pre30}$ respectively.

Also shown in Figure 3 as a dotted line is the operating boundary from the theoretical simulations using an ideal semi-active suspension. The ideal performance with Preview control $S4_{idl30}$ is clearly considerably better than the measured performance $S4_{pre30}$. This is because the prototype semi-active damper cannot produce the ideal damper performance due to the phase lag between the demanded and achieved damping forces (Kitching, Cole and Cebon 2000a).

4.0 PHASE LAG COMPENSATION

4.1 Introduction

Preview control provides the possibility of compensating for the delay in developing the damping forces, which is an important limitation of real semi-active dampers (Nagiri, Doi, Shoh and Hiraiwa, 1992; Morita, Tanaka and Kishimoto, 1992). This technique will be referred to here as Phase Lag Compensation (PLC). In essence this entails using the feed-forward information captured through the Preview control to predict the demanded damping force at a compensation time t_c ahead of the time the damper force is required. The following sections describe an experimental investigation of PLC.

4.2 The Prediction Model

The objective of the prediction model is to anticipate the future states at the drive axle of the HiL vehicle. One of the simplest prediction models is a two-DOF quarter truck model, but this fails to account for the pitching of the vehicle body or the coupling between the steer and drive axles. A four-DOF prediction model was therefore used. The prediction model has a shorter wheel-base than the HiL model, with the steer axles of both models subjected to the same road input (see Kitching, Cebon and Cole, 1999 for details).

4.3 Deterministic Road Profile Simulations with PLC

The benefits of PLC were investigated with the HiL rig, using a filtered-step input at a vehicle speed at 14*m/s*. The results are summarised in figure 4 (a) and (b). Both figures show that increasing the compensation time t_c improves the response until around

t_c=20*ms*, where it starts to increase. The minimum at approximately t_c=20*ms* is evident in all the results analysed and is to be expected, because the time delay in the prototype damper is known to be approximately 20*ms* (Kitching, Cole and Cebon, 1999).

4.4 Summary of PLC Results

A summary of the improvements obtained with the prototype suspension relative to the optimised non-linear passive suspension for the four-DOF HiL simulations is presented in figure 5. The top half of figure 5 shows the results for a surface profile typical of motorway conditions, while the bottom figure presents the results for a filtered step 'ramp' input. The benefits available through PLC for reducing the RMS body acceleration of the HiL vehicle are very encouraging for both random and transient road profiles. The reductions achievable in RMS tyre forces are less significant.

5.0 SEMI-ACTIVE DAMPER DEVELOPMENT

From the work with the prototype semi-active damper, Fig. 1a, a specification was developed, for a 'production' damper to achieve sufficient performance benefits (see Table 1). Koni Dampers in The Netherlands developed a damper to this specification.

	C_{max} (kNsm^{-1})	C_{min} (kNsm^{-1})	$\frac{C_{max}}{C_{min}}$	100% Force transition time at V=0.15m/s (ms)		66.7% Force transition time at V=0.15m/s (ms)	
				ΔF>0	ΔF<0	ΔF>0	ΔF<0
Desired Specs.	130	5	26	40	40	-	-
Acceptance Level	130	20	6.5	-	-	20	20
Measured Specs.	100	5.5	18	66	33	32	17

Table 1. Specifications for the 'production' semi-active damper.

5.1 Technical Details of Koni Semi-Active Damper

The Koni Continuously Variable (or semi-active) Damper (CVD) is a unidirectional flow design, providing both bump and rebound damping forces from the same servo controlled valve. A voice-coil controls fluid flow in the servo valve, and consequently the pressure in the 'amplifier'. A piston attached to the amplifier converts this pressure into preload on the main valve. As soon as the damper moves, and fluid flow has commenced, the servo valve builds up a pressure determined by the current in the voice coil. The resulting preload on the main valve governs the ease of flow of the majority of the displaced damper fluid.

Figure 6 (de Kock, 1993) shows a cross sectional view of the base of the damper. Movement in either direction results in fluid passing through the central tube (1) to the main valve (2) which is controlled by the amplifier piston (3) and coil spring (4).

The fluid flow divides into a main flow (which flows through the main valve); and a control flow (which passes through a hole in the amplifier piston) into the amplifier chamber. The fluid exhausts through the amplifier chamber through the pilot valve (5) which is attached to the otherwise freely moveable voice coil (7). The pilot valve is held closed by a second small coil spring (6). The pilot valve, in co-operation with the voice coil, controls the pressure in the amplifier chamber and by that the preload on the main valve. A fail-safe function has been built–in so that in case of an electrical failure, the damper functions as a passive damper.

Compared with conventional servo valves, the system is less expensive to build, but has it's own specific problems. The fluid flow through the piston is not constant but depends on the damper pressure. Furthermore there is no mechanical or electrical feedback from the main valve to the piston valve, but only a pressure feedback that enables the voice coil to control the pressure in the amplifier chamber and consequently the damper forces. Inherent to these kind of systems are instability problems which can be solved by adding damping to the pilot and amplifier valve. This damping tends to increase the response times of the valve, so a tuning process is necessary. All problems were solved but they do demand a thorough knowledge of the system to tune it correctly.

5.2 Testing of Koni Semi-Active Damper

The majority of damper testing was aimed at the two critical areas of performance: damping level and force step transition.

5.2.1 Damping Level Testing

The damper was connected to a hydraulic actuator, with a load cell used to measure the force produced at the mountings. A signal generator was used to move the actuator in a sinusoidal motion, while the force and displacement were logged. Plots of force against velocity for various damping settings were then drawn.

From Table 1 and Figure 7a, it can be seen that the lowest required damping coefficient C_{min} was met, as was the ratio of maximum to minimum damping coefficient C_{max}/C_{min}. These are the most critical parts of the specification. The required maximum damping coefficient of 130*kNs/m* would be achievable by modification of the main valve, which would maintain the max/min damping level ratio constant, but would involve a higher value of C_{min}, (probably 10 or 20 *kNs/m*). It was decided to leave such alterations until the first set of field trials have been completed.

5.2.2 Force-Step Transition Testing

In the force transition testing, the ram was made to extend / contract the damper with a sawtooth motion, so as to give the damper periods of constant velocity of 0.15*m/s*. During these constant velocity stokes, the damping setting was subject to a step change, and the force was monitored.

The specified performance for the damper is shown in Table 1, with a plot of the test results in figure 7b. It can be seen that an increase of force takes longer than a decreasing transition. Comparison with the specifications reveals that all but the most severe force transitions completed within the specified time, and further testing showed that at almost all other speeds, the entire force transition specification was met. Response times become proportionally better with increasing damper speed. It is also necessary to account for the fact that the force build–up depends on the rise–time of the driving current, which in turn depends on the rise-time of the power amplifier being used to control the damper.

5.3 Final Semi-Active Damper Design

The final version of the damper is shown in Figure 1b. The pressurised external oil reservoir enables the damper to operate at any angle (even upside-down) by ensuring that no air is ever drawn into the hydraulics. This also enables the control wires to exit upwards to the body of the vehicle, rather than onto the axle.

6.0 CURRENT WORK

The current program of development should reach a new stage within the next quarter. A tri-axle tanker-trailer unit supplied by General Trailers will be equipped with a set of custom modified independent suspension units. These will incorporate six of the finished Koni dampers. Field trials on the vehicle will enable PLC strategies to be investigated. It will also provide a platform on which new control strategies can be tested. The objective is to develop a practical package of hardware and software which can be fitted to heavy goods vehicles

Also fitted to the suspensions are a set six hydraulic rams to allow the vehicle to be tilted during manoeuvres; thus helping to reduce the risk of vehicle roll-over. The details of this research are given in the accompanying paper 'The Development of an Active Anti-Roll System for Heavy Vehicles' (Sampson, Jeppesen, and Cebon, 2000).

7.0 CONCLUSIONS

(i) A four-DOF HiL test rig has been developed for heavy goods vehicles.

(ii) The theoretical benefits of preview control for semi-active suspensions were partially achieved experimentally. Relative to an optimised non-linear passive suspension, preview control improved the RMS body acceleration at the drive axle by 15.4% for the HiL simulations over a random motorway roads profile.

(iii) For the motorway simulations, Phase Lag Compensation (PLC) reduced the RMS body acceleration at the drive axle by a further 7%, relative to the optimal measurements without PLC.

(iv) The PLC for a filtered-step input resulted in a 20% reduction in the RMS body acceleration at the drive axle, compared to the measurements without PLC.

(v) It was found that there was little to be gained in terms of reducing the RMS dynamic tyre force and suspension working stroke through PLC. Increasing the accuracy of the prediction model is expected to improve this aspect of PLC.

(vi) Development work has lead to the production of a semi-active damper for heavy vehicles, which will be used in future research.

8.0 ACKNOWLEDGEMENTS

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FIGURES

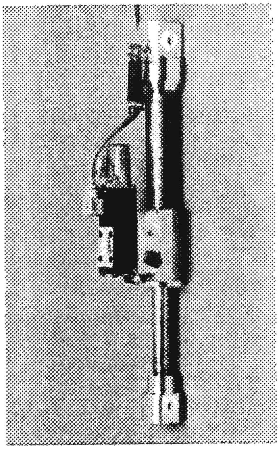

Figure 1a The CUED prototype CVD semi-active damper. (Kitching et al 2000a)

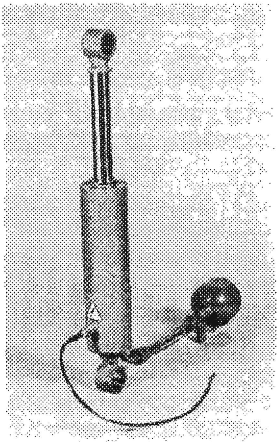

Figure 1b Koni production version semi-active damper

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Figure 2: The four-DOF HiL heavy vehicle tractor unit model

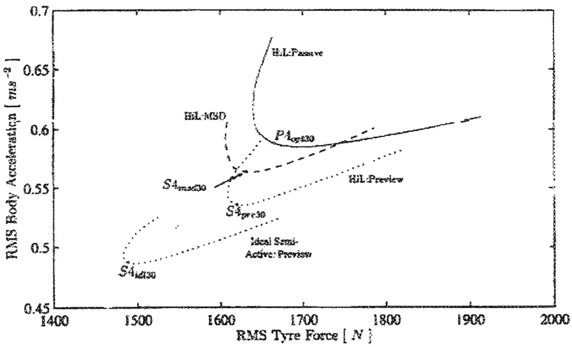

Figure 3: Conflict Diagram of RMS tyre force for the four DOF HiL simulations for motorway input conditions.

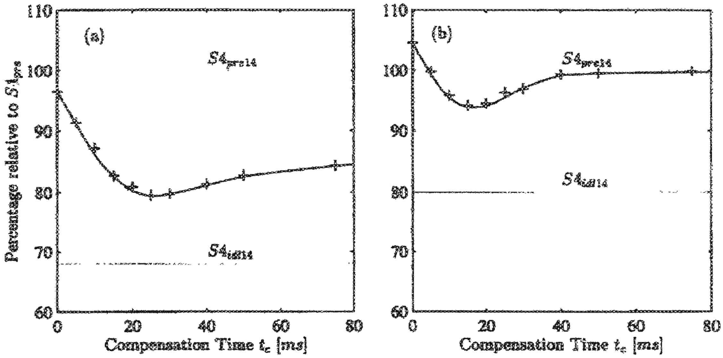

Figure 4: Summary of benefits from PLC for the filtered-step input : (a) RMS body acceleration at the drive axle, (b) RMS dynamic tyre force at the drive axle

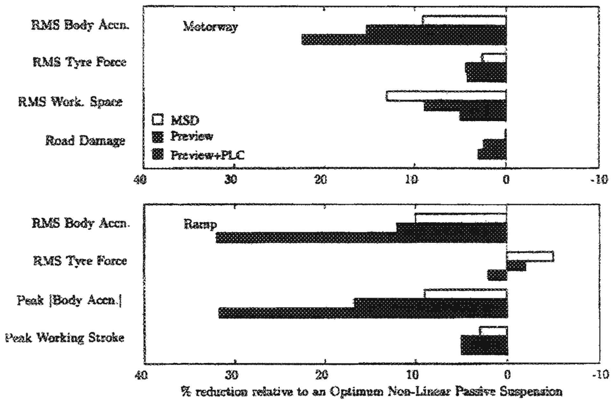

Figure 5: Summary of the measurements with the prototype semi-active damper using a four DOF HiL test rig. All criteria with the exception of road damage relate to the measurements at the drive axle only. (a) Motorway; (b) Filtered step (ramp).

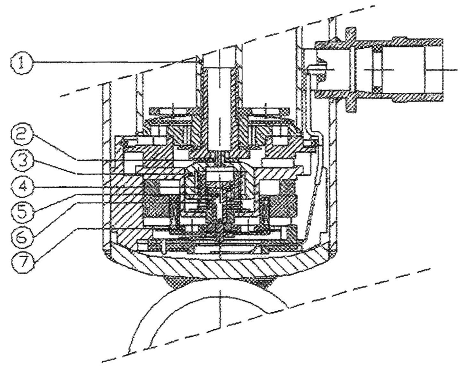

Figure 6: Diagram showing sectioned view of base of damper.

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Figure 7a and 7b : Results of the damping level tests and results of force-step transition tests on Koni prototype CVD semi-active damper