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The Development of an Active Roll Control System for Heavy Vehicles

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Abstract

This paper describes the development of an active roll control system for a tractor semi-trailer. The design of the hardware and software are explained. Simulations of the yaw-roll response of the vehicle show that the system will provide significant improvements in rollover stability.

1.0 INTRODUCTION

1.1 Background

Studies have shown that most rollover accidents involve heavy articulated vehicles, and occur on highways (Kusters, 1995). Three major contributing factors to rollover accidents have been identified: (1) sudden course deviation, often in combination with heavy braking, from high initial speed; (2) excessive speed on curves; and, (3) load shift. The accidents are relatively frequent, and the estimated average cost to operators is between USD 120,000 and 160,000 (Harris, 1995). The associated costs of damage to property and traffic delays can be very large.

1.2 Previous Research

The use of active suspension systems on heavy articulated vehicles, particularly to control roll motion, has been researched to a relatively small degree (Kusters, 1995; Besinger, Cebon and Cole, 1995). However, several researchers have indicated potential improvements in rollover safety are possible, even when using relatively low-power, low-bandwidth actuators to control the active suspension system.

Dunwoody (1993) simulated the steady state cornering performance of a tractor semi-trailer fitted with an active roll control system. The system consisted of a hydraulically tiltable fifth wheel coupling and hydraulic actuators that could apply control torques to each of the trailer axles. The control system measured the lateral acceleration of the trailer and the relative roll angle between the tractor and the trailer, and the study found such a system could raise the static rollover threshold by 20-30%.

Lin, Cebon and Cole (1996a) investigated the use of active roll control on a single unit truck using a simple linear model. The performances of systems based on roll angle, lateral acceleration and load transfer feedback were investigated. Control gains were selected by pole placement. The authors recommended using a control system based on lateral acceleration feedback, which demonstrated several key benefits: (1) the ability to tilt vehicle into a corner, providing significant improvements in load transfer; (2) fast transient response; and, (3) relatively simple instrumentation requirements. The study reported that such a system could provide worthwhile reductions in transient and steady state load transfer of up to 30%. Lin et al. also investigated the performance of a roll control system designed using an optimal state feedback technique and a steering input power spectrum based on road alignment data and pseudo-random lane changes. The system performance was marginally superior to that of the lateral acceleration feedback controller.

Lin et al. (1996b) simulated the performance of a tractor semi-trailer with torsionally rigid frames, fitted with a roll control system. The controller was based on lateral acceleration feedback. Control gains were selected by pole placement. The study found that such a system could reduce steady state and transient load transfer for a range of manoeuvres. The study recommended investigating the influence of vehicle frame flexibility on control system performance.

Sampson and Cebon (1998) proposed a vehicle roll control system design methodology based on a linear quadratic regulator. The study found that this design technique allowed the control system designer to make trade-offs between performance and power consumption requirements when designing multiple-actuator roll control systems for tractor semi-trailers and long combination vehicles with flexible frames.

1.3 Experimental Vehicle

The roll control system described in this paper is one sub-system of a computer-controlled experimental vehicle being developed by the Cambridge Vehicle Dynamics Consortium (CVDC). The system consists of five active anti-roll bars (two on the tractor and three on the semi-trailer) driven by nine hydraulic actuators, under computer control. The air suspension systems on the vehicle also incorporate a ride control system consisting of ten high-performance continuously variable semi-active dampers, which were developed by Koni BV (Roebuck, Kitching, Cebon, and de Ruiter, 2000).

Design of the anti-roll hardware is a significant challenge, given the large forces, torques and roll-rates needed for effective control. This paper addresses some key practical constraints, including component strengths and hydraulic limits (power, flow and pressure). Design of the control system software is also governed by challenging performance and safety requirements.

2.0 SUSPENSION HARDWARE

2.1 Conceptual design

The trailer suspension is a modified *Indair* air suspension unit from Meritor HVS, which consists of two independent trailing arms hinged from a transverse beam. The active roll control system consists of a stiff U-shaped anti-roll bar, connected at each end to the trailing arms, and two hydraulic actuators located between the chassis and anti-roll bar (Fig. 1). The actuators apply equal and opposite vertical forces to the bar, thereby twisting it, and applying a roll moment to the vehicle body. The anti-roll bar effectively floats, and its position is determined by the wheel and actuator positions. Use of a single hydraulic actuator would have been considerably simpler, but was not possible because the much larger stroke requirement exceeded the available space under the vehicle.

2.2 Actuator specifications

An analysis of the kinematics and dynamics of the roll control system was performed to determine the required stroke and size of the hydraulic actuators. The stroke required to move the sprung mass to the maximum roll angle (as limited by the suspension travel) of ±6.1° was 85 mm. The maximum required actuator force was determined for both steady state and transient manoeuvres.

The maximum steady state actuator force, required to hold the sprung mass at zero roll angle during a 0.5g (vehicle rollover threshold) steady state turn, was calculated to be 110kN.

The worst case transient force is that required to drive the sprung mass sinusoidally at a specified frequency with the maximum amplitude of roll angle (Sampson, McKevitt and Cebon, 1999). The system has a roll resonance at 1.8 Hz. This places an upper limit on the achievable system bandwidth of approximately 1.2 Hz. The maximum actuator force requirement below this frequency is the DC value of 128 kN. The actuators selected to meet this specification have 125 mm bore, and produce 137 kN for a rod diameter of 56 mm and a pressure of 210 bar. The large actuator force necessitates a large piston area, and thus large fluid flows through the actuator. For a 1 Hz oscillation, a cylinder of volume 1.1l requires a flow rate of 2.2 ls^{-1}. The limited flow rate through the servo-valves further constrains the achievable response bandwidth of the roll control system. Accumulators are used to store hydraulic fluid to allow the system to operate for a limited number of extend-retract cycles. Harmonic motion is not possible indefinitely.

3.0 CONTROLLER DESIGN AND IMPLEMENTATION

3.1 Vehicle Modelling

The vehicle model used in the design of the roll control system is an extension of the simple three degree of freedom single unit yaw-roll model developed by Segel (1956). The model has been extended to model several vehicle units coupled together, with each having yaw, sideslip, front roll and rear roll freedoms (Fig. 2). The front and rear sections of each vehicle unit are coupled with a torsionally-flexible frame A simple tyre model represents the change in tyre cornering stiffness with vertical load. Control torques u_f and u_r, representing the torques applied by the active roll control system, act on each section of the sprung mass. Limitations in the available torque and the response time of the hydraulic actuators are also included. Equations of motion and further details may be found in Sampson, McKevitt and Cebon, 1999.

3.2 Controller Architecture

A distributed controller architecture has been adopted, consisting of a "global" control unit and multiple "local" control units, communicating over a digital Controller Area Network (CAN) (Fig. 3). This architecture simplifies the physical installation, maximises real-time performance, and enables modular code development and rapid prototyping. The Local Controllers process signals from the transducers, and perform closed-loop control of the hydraulic actuators in response to demand signals from the global controller. The Global Controller reads signals from the CAN bus, performs vehicle control calculations, and sends demand signals to the local controllers at each axle.

3.3 Local Controller Design

The main component of a Local Controller is a PID controller with lag pre-filter which controls roll moment, but because a floating anti-roll bar arrangement is used, the local controller also includes an outer loop that ensures that the centre of the anti-roll bar is held at zero displacement (McKevitt, 1999). This is necessary to ensure sufficient ground clearance and to enable maximum roll stroke to be achieved. The dynamics of the vehicle system and the feedback sensors complete the feedback loop. The actuator model captures several limitations in the control system hardware, notably the limits on maximum actuator force, maximum flow rate through the servo-valve, and bandwidth of the servo-valve. The flexibilities of the mechanical and hydraulic components in the active anti-roll bar assembly are included in the vehicle and actuator models.

Gains for the Local Controllers were selected using pole placement, with the aims of ensuring robust stability and good steady state tracking of the demand torques, and well as a fast rise time, fast settling time and smooth step response. The local control system has a rise time of approximately 0.3s in response to a step roll torque demand signal of 60 kNm (McKevitt, 1999).This is sufficiently fast for this application, especially given that the steering input spectrum that forces the vehicle roll motion is concentrated below 1 Hz (Lin, 1994).

3.4 Global Controller Design

The objective of the Global Controller is to minimise lateral load transfer in response to steering inputs, since it is excessive lateral load transfer that causes vehicle rollover. Lateral load transfers due to centripetal acceleration and lateral coupling forces are set by the vehicle dimensions and trajectory. However, other load transfer terms, due to vehicle body roll and torques applied by adjacent vehicle units through couplings, are strongly influenced by the performance of the suspension and the active roll control system. The roll control system is designed to work using simple instrumentation. The lateral acceleration at the centres of mass of the tractor and trailer is measured, as are the roll rates of both vehicle units.

The gains for the global controller were selected using pole placement. The proportional gains applied to the lateral acceleration signals were designed to give equal rollover thresholds at each axle during steady state cornering. This maximises the rollover threshold of the vehicle as a whole and minimises changes in vehicle handling performance. High gains on lateral acceleration, which are necessary to tilt the vehicle units into turns, cause instabilities in the roll dynamics. However, these instabilities can be stabilised by adding roll rate feedback, which increases the damping of the roll modes. The root locus plot in Fig. 4 shows the effects of lateral acceleration and roll rate feedback on the stability and speed of trailer axle feedback loop. Attempts to stabilise the system using derivative feedback on lateral acceleration proved ineffective.

The performance of the active roll control system is compared with that of a passive suspension system for a severe lane change manoeuvre in Fig. 5. The active system is able to tilt the vehicle into turns, reducing load transfer. The normalised load transfer of both the tractor and trailer units are reduced by around 25% over the passive system. The roll angle into the turn of the tractor is greater than that of the trailer. This difference in roll angles produces a torque between the tractor and trailer which reduces the load transfer at the trailer axles. The benefits that can be obtained using this roll moment co-operation effect are limited if the fifth wheel or the trailer chassis is very flexible in roll.

Simulations of the response of the active system during steady cornering indicate that reductions in load transfer of around 20% are achievable.

3.5 Development of Controller Hardware and Software

The Controller Area Network is an industry standard architecture governed by ISO11898. Due to limits on the bus data rate, two separate buses are being used, one for roll control and one for ride control. Fig. 6 shows the arrangement of sensors for the Roll Control CAN. There is one Local Controller on the tractor and one Local Controller on the trailer for each CAN.

The Global Controller is based on an Intel Pentium II 400 MHz PC. The global control system software is developed in a graphical environment, translated into C code, compiled and downloaded onto the Global Controller. The Watcom C compiler and the Mathworks products Simulink, Real Time Workshop and xPC are used to achieve this.

The Local Controllers are being developed in collaboration with Mektronika Ltd. They are modular controller boxes containing a single backplane which allows various printed circuit boards to be added. The main board is designed around a Siemens C165 microprocessor, while additional boards allow analogue and digital I/O, signal conditioning, connection to the CAN bus etc. The local control system software is developed in C and downloaded onto the microprocessors.

3.6 Safety Considerations

The control system must be designed to avoid vehicle instability due to inappropriate application of roll torques. Guidelines for commercial products (HSE, 1987) recommend that a Programmable Electronic System should be at least as safe as a passive system it replaces. Any commercially available Active Roll Control system must therefore be tested rigorously for safety after component failure.

Financial and time constraints restrict a research project from undertaking such a thorough level of testing, but it is crucial to be able to trace faults in order to improve the system. An initial assessment of the effects of component failure (Jeppesen, 1999) showed the need for some form of redundancy (at least two independent measures of a parameter) in all safety-critical feedback data, and the need to be able to shut down the system independently of the Global Controller and CAN bus. A separate hardwired "stopline" linking the power supplies of all controllers, and which any controller can activate, is being implemented to facilitate this.

4.0 CONCLUSIONS

An active roll control system, based on a modified passive suspension system, has been developed for a tractor semi-trailer. The system uses active anti-roll bars, controlled by hydraulic actuators, to control roll motion at each axle. The system uses simple instrumentation to measure the lateral acceleration and roll rate of the tractor and trailer. Simulations of the yaw-roll performance of a tractor semi-trailer fitted with the active roll control system indicate that the system will provide steady state and transient improvements of up to 25% of the rollover stability of the vehicle. Development of the control strategies is continuing.

The system uses a distributed controller architecture, consisting of a global controller and several local controllers, all communicating over a CAN bus. The distributed architecture simplifies installation, optimises performance and allows rapid prototyping. The system is being designed to remain safe in the event of hardware or software failure.

A prototype vehicle fitted with the active roll control system will be tested shortly.

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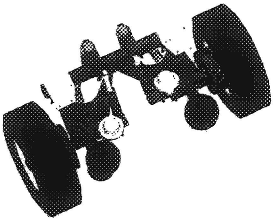
FIGURES



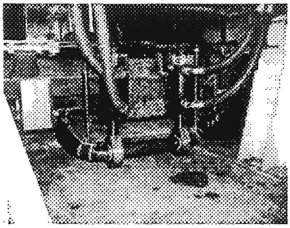


Fig. 1. Trailer suspension, showing the location of the actuators and the anti-roll bar: (a) plan view; (b) front elevation.

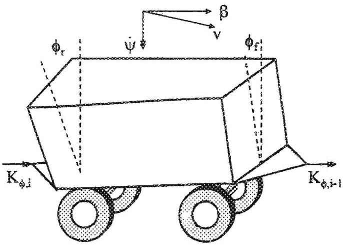


Fig. 2. Schematic diagram of a generic vehicle unit.

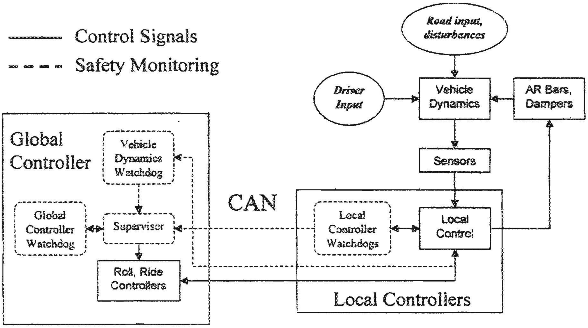


Fig. 3. Schematic of Vehicle and Controller interactions.

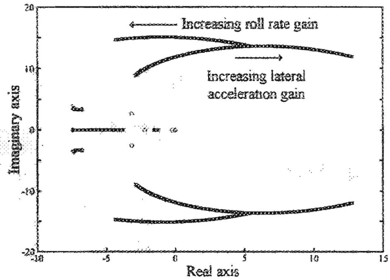


Fig. 4. Root locus plot for trailer axle roll control feedback loop, showing the effects of lateral acceleration gain and roll rate gain on system stability.

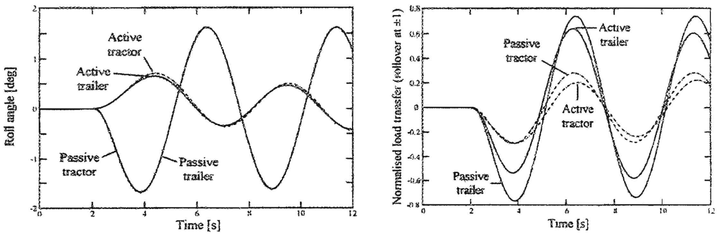


Fig. 5. Response of active and passive vehicles during a lane change manoeuvre: (a) roll angle; (b) load transfer.

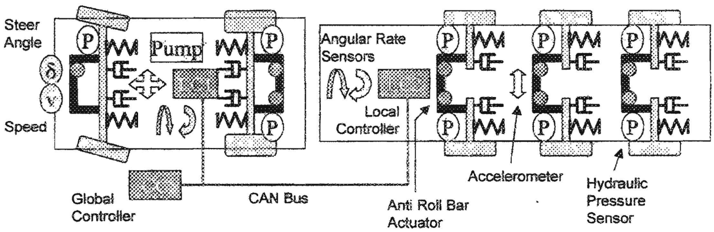


Fig. 6. Sensors, Actuators & Controllers.