This paper describes the design and implementation of an active roll control system for a heavy vehicle. A simple yaw-roll model of a tractor semi-trailer is developed, validated and used for control design. Two simple control strategies are investigated and implemented on an experimental vehicle. The implementation of the control strategies on the digital controllers is detailed.

Keywords: heavy vehicle, roll-over, active roll control, active suspension, anti-roll bars, hydraulic actuators.

1. INTRODUCTION

1.1. Background
Roll-over of heavy vehicles is a serious worldwide safety problem. Over 15,000 rollover accidents per year involving commercial heavy vehicles occur in United States, including 9,400 involving tractor semi-trailer combinations [1].

The average cost of a rollover accident in the UK is estimated to be between £75,000 and £100,000 for the operator, including recovery and repair of the vehicle, road surface repair and product replacement [2]. Extra indirect costs, such as hospitals and emergency services, are evaluated to be £4 million per year [3]. Overall, the cost of rollover accidents in the UK can be estimated to be £40-60 million annually, excluding environmental costs and costs arising from traffic delays. These high costs are the primary motivation for research into active roll control of heavy vehicles.

Roll-over accidents mostly involve articulated vehicles and occur on highways [4]. Three major causes of rollover have been identified: sudden course deviation (often combined with heavy braking and high initial speed); excessive speed on curves; and shifting load. It is usually difficult for the driver to sense the roll-over behaviour of a tractor semi-trailer combination as his perception is mainly based on the response of the tractor, and the trailer is generally the first unit to rollover. Palkovics et al. classified rollover accidents into four categories: preventable, potentially preventable, non-preventable and preventable unknown [5]. It was shown that only a minority of rollover accidents could have been avoided with a warning device, potentially more with a skilled driver, but half of the rollover accidents they investigated were not preventable by driver action alone. This highlights the need for an active safety system for heavy vehicles.

1.2. Previous research
Extensive research has been done on the use of active suspension for automobiles, but comparatively few researchers have investigated heavy-duty vehicles, especially the control of roll motion. Most of the studies that have been published are largely theoretical, and relatively little data is available concerning the practical or experimental implementation of active systems.

Dunwoody and Froese studied active roll control of a semi-trailer, where the rollover threshold was increased by tilting the trailer into the corner [6]. Their anti-roll control system consisted of a modified tiltable fifth wheel and hydraulic actuators, and was entirely contained within the trailer. Calculations showed an increase in rollover threshold of up to 30%.

Lin et al. performed static and dynamic tests on an articulated vehicle to validate a non-linear vehicle simulation [7]. A simple linear model was then developed to investigate three different control strategies: roll angle, lateral load transfer and lateral acceleration feedback. The latter was found to be the most attractive solution as it was able to tilt the vehicle into the corner, thereby reducing significantly the lateral load transfer, and required relatively simple instrumentation.

Sampson and Cebon developed a generic yaw/roll model for multiple unit articulated vehicles [8]. They designed a state-feedback roll control system (LQR) for a flexible tractor semi-trailer combination. It was found to reduce steady state and peak transient load transfer significantly when compared to a passive vehicle. The influence of frame flexibility on controller design was also investigated.
1.3. The CVDC active truck project

The Cambridge Vehicle Dynamic Consortium (CVDC) is currently developing an experimental 38 tonne tanker semi-trailer, fitted with active suspensions and continuously variable dampers. The active components consist of hydraulic actuators mounted in the middle of a floating anti-roll bar (ARB) on each axle, each actuator being controlled by a servo-valve (Fig. 1). The control architecture consists of 4 Local Controllers or LCs (one for roll and one for ride on both the tractor and the trailer) and a PC-based Global Controller or GC. The controllers communicate with each other via two CAN buses (one for roll and one for ride). Fig. 2 shows the experimental vehicle.

Fig. 1: CAD model of the CVDC active suspension.

Fig. 2: CVDC computer-controlled experimental vehicle

2. IMPLEMENTATION ON EXPERIMENTAL VEHICLE

2.1. Hardware and instrumentation

The trailer roll instrumentation consists of:

- 6 suspension travel transducers mounted between the suspension trailing arms and the chassis of the trailer, to measure suspension roll angles;
- 3 bar-centre displacement transducers to measure the average position of the ARBs;
- 6 pancake load cells mounted on top of the actuators, to measured the load provided by the actuators;
- 6 pressure transducers to measure the pressure in each hydraulic accumulator;
- 1 roll rate and 1 yaw rate sensor;
- 1 accelerometer to measure lateral acceleration;
- 2 strain gauges on the trailing axle to measure lateral load transfer.

The tractor instrumentation currently consists of:

- 1 roll rate and 1 yaw rate sensor;
- 2 accelerometers to measure longitudinal and lateral acceleration;
- 2 string potentiometers (one to measure steering angle and one to measure articulation angle between tractor and trailer);
- 1 Datron optical speed sensor to measure forward speed and sideslip angle (not available for all tests, in which case the speed was measured from the tachometer).

More instrumentation is to be added when the tractor is equipped with active suspensions. There is also additional instrumentation for the ride control system.

On the trailer, all the cables from the sensors are bundled up together in a wiring loom (one per axle) and fed into a ‘junction box’, which provides power, amplification and buffering for some signals such as displacement transducers, and switches the relevant solenoid valves on and off through logic circuitry in order to have the correct hydraulic mode (actuators locked, ‘floated’ i.e. passive, active or in ‘damped float’ mode). The logic circuitry is operated via two logic lines coming from the trailer roll LC. The voltage demands for the servo-valves also get relayed and filtered through the junction boxes. There are two cable trays (one for power and one for signals) that run alongside the trailer and connect the roll and ride junction boxes to the trailer LCs.

The trailer roll LC is connected to the other LCs and to the GC via 2 CAN cables (one for roll and one for ride) which run from the tractor cab to the rear engine raft on the trailer. The engine raft houses the hydraulic pump, the hydraulic tank, a diesel engine and alternator to provide power and the cabinets for the trailer LCs. For the trailer, a substantial amount of information is being sent over the CAN: not only sensor readings, but also control gains, controller demand signals and control mode, etc… The GC also acts as safety monitoring device that shuts the system down in case a fault is detected [9]. This is summarised in Fig. 3:

Fig. 3: Hardware and instrumentation on the CVDC experimental vehicle.
2.2. Trailer roll local controller (ICON 2000)

The LCs are custom-built units by Mektronika Systems called “ICON 2000” (Intelligent CONtrol). Several cards can be slotted into the controller according to the user requirements:

- Power supply card;
- Main CPU card with a C165 microprocessor, which performs all the sensor reading routines, CAN message packing and unpacking, timing tasks, communication with other cards, etc…
- Controller card with a C165 microprocessor, which performs all the tasks and calculations related to actuator control, with digital outputs (logic signals for solenoid valves) and analogue outputs (servo-valve demand voltages);
- Voltage input cards, with 8 analogue input channels per card;
- Strain gauge cards, with 8 strain gauge input channels per cards;
- Communications card, which performs CAN related routines.

Each microprocessor requires its own separate program. The timing between the main CPU card and the controller card, and between the main CPU card and the GC has to be synchronised for correct operation. All cards communicate with the main CPU card via a dual-port RAM on the system bus. There is an RS232 interface on both the main CPU card and controller card for communications with a PC.

The main CPU card cycles and sends/receives CAN messages at 250Hz, which means that the data is logged on the GC at 250Hz. The controller card cycles at 50Hz. There is no benefit in cycling any faster on the controller card because the roll motion of the vehicle cannot respond any faster. In addition, the controller card has to perform a substantial number of control computations per cycle, which limits its cycle frequency. The analogue filters on the voltage input cards are equivalent to second order low-pass Butterworth filters with a cut-off frequency of 33Hz. The CAN bus operates at 500 kBaud, but has the potential to operate at 1 MBaud.

2.3. Local Controller programming

The code for both the main CPU card and controller card have to be written in C and then compiled and assembled into two C165 compatible ‘hex’ files for download on the ICON 2000. The download process is done through the RS232 interface with a bootstrap loader. The tasks required to be performed by the main CPU and the controller card can be summarised as shown in Fig. 4.

A hardwired stop line has been installed on the vehicle for safety reasons. Emergency stop buttons can also cut the power off to the junction boxes and therefore to the servo-valves. If the GC or LC goes into shutdown mode, a CAN message is logged with error source and description for future analysis and debugging.

![Fig. 4: Main CPU card & controller card program architecture.](image)

In Fig. 4, control related tasks are represented by the one step “Compute PID output & servo-valve demand voltage”. All calculations are done using integers, the sensor signals being represented by the number of ‘counts’ from the Analogue to Digital Converter (ADC), which converts -10/+10V into 4095/0 counts in order to maximise computation speed. All the control algorithms have to be converted from real number arithmetic to integer arithmetic.

In order to investigate more advanced control strategies (such as LQG or LQR), it is planned to use the GC (which possess a Matlab Simulink/xPC interface) to do all the state feedback and demand computation, while the LC will only do the local (PID) control as it currently does.

3. SIMPLE YAW/ROLL MODEL OF EXPERIMENTAL VEHICLE

3.1. Introduction

The model in Fig. 5 consists of:

- A bicycle model of the tractor unit, with the speed and steering angle as inputs and lateral acceleration, yaw rate and sideslip angle as outputs;
- A roll plane model of the trailer unit, including the active ARBs and hydraulic actuators, It has lateral acceleration and spool valve displacement as inputs, and body roll angle, roll rate and actuator torque as outputs;
- A controller, with any chosen states as inputs and servo-valve spool displacements as outputs.
- A small sub-model to calculate load transfer from body roll angle and lateral acceleration.

![Fig. 5: Simple yaw/roll model architecture.](image)
The roll and yaw degrees of freedom are not coupled. This assumption simplifies the controller design. Only the roll plane model of the trailer unit is presented here, as it is the most relevant part.

3.2. Roll plane model of trailer unit

The roll plane model of the trailer unit shown in Fig. 6 consisted of two degrees of freedom (body roll angle and anti-roll bar angle). The axle was assumed to be rigid, therefore there is no unsprung mass roll angle.

Fig. 6: Roll plane model of trailer unit tilting towards the centre of the turn.

The equations of motions are obtained by writing the moment equilibrium of the sprung mass and the anti-roll bar about the roll centre of the suspension:

\[
\begin{align*}
I_s \ddot{\phi} + C_s \dot{\phi} + (K_b - mgh)\phi + ma_y h - T_{act} &= 0 \\
I_{arb} \ddot{\phi}_{arb} + K_{arb} \phi_{arb} + T_{act} &= 0 \\
\frac{d}{dt} \left( ma_y h \right) &= \phi - \phi_{arb}
\end{align*}
\]

(1)

The roll plane model of the trailer unit can therefore be reduced to a simple linear 2DOF model. The anti-roll bar inertia is light compared to the sprung mass inertia and can be neglected. For control purposes, the active hydraulic elements (actuators and servo-valves) need to be modelled, as shown in Fig. 7, using the previous 2 DOF linear model.

The introduction of the active hydraulic elements into the roll plane model brings four extra equations to Eq. 1, which are derived from [10]:

\[
\begin{align*}
T_{act} &= 2F_a d_a \\
F_a &= A\Delta P \\
Q_L &= A\dot{\phi} + C_p \Delta P + \frac{V}{4\beta_p} \frac{d\Delta P}{dt} \\
Q_L &= K_x x_v - K_p \Delta P
\end{align*}
\]

Fig. 7: Roll plane model with active elements.

In total, Eq. 1 and 2 constitute 7 equations for 9 unknowns \((x_v, a_y, \phi, \phi_{arb}, y_v, F_a, T_{act}, \Delta P, Q_L)\), so that two ratios (or transfer functions) can be determined. The system has two inputs, \(x_v\) and \(a_y\), and one output, which can be \(\phi\) for roll angle control or \(T_{act}\) for roll moment control.

The active system on the CVDC experimental vehicle can be switched off to simulate a passive vehicle or locked in position to simulate a passive vehicle with extra roll stiffness. In those two cases, the trailer tilts in the same direction as the centrifugal force, so that the first equation in Eq. 1 is replaced by:

\[
I_s \ddot{\phi} + C_s \dot{\phi} + (K_b - mgh)\phi - ma_y h + T_{act} = 0
\]

(3)

The condition for a passive vehicle is \(F_a = T_{act} = 0\) (the actuators do not generate any force as they are free to move) and the condition for a passive vehicle with locked ARBs is \(y_v = 0\) (the actuators are locked in position and therefore do not extend or contract).

3.3. Model validation

The model was validated both in steady state and transient conditions by experimental data gathered on the CVDC experimental vehicle. Only the most relevant data for active roll control, namely roll angles and ARB moments, is presented here.

Steady state results for a laden vehicle are shown in Fig. 8 and 9. A negative angle indicates the vehicle tilting towards the outside of the turn. The increase in roll stiffness from locking the anti-roll bars can clearly be seen as it reduces the suspension roll angles from approximately 5° to 3° at 0.3g lateral acceleration. A small amount of residual torque can be noticed in the passive case due to friction, whereas in the locked case the torque is simply given by \(-K_{arb}\phi\).
Transient results are presented in Fig. 10 and 11 for a laden vehicle with the ARBs locked, during a slalom manoeuvre. The input to the model was the measured trailer lateral acceleration. The measured roll angles for the three trailer axles are fairly similar to one another but the experimental vehicle is not symmetric: it leans more one way than the other. The roll-plane model, being an ideal model, does not take into account this, but the match between the predicted and measured roll angles is still acceptable despite that asymmetry.

The asymmetry is even more obvious in the ARB moments in Fig. 11: the measured roll moments on the trailing and leading axles are offset positively while the middle axle is offset negatively. The roll-plane model is symmetric and generates the same moment on all three axles. It fits the measurements reasonably well despite the asymmetry.

4. CONTROL SYSTEM DESIGN

4.1. Introduction

So far, two different control strategies have been considered, implemented and tested:

- Roll angle control: the body roll angle of the vehicle is controlled to a specified angle. This is normally used for commissioning.
- Roll moment control: the torque provided by the actuators is controlled and the torque demand is proportional to the measured lateral acceleration. The lateral acceleration gain is chosen so that maximum lateral acceleration corresponds to maximum tilt inwards in the turn.

Bar centre control maintains the ARB in its centre position (corresponding to mid-actuator extension) with a slow time constant. This is necessary because of the ‘floating’ nature of the ARBs, whose position is determined by the wheel and actuator positions. The roll angle control and bar centre control are not detailed here as they do not relate to vehicle performance.

4.2. Roll moment control

The architecture of the roll moment control system, including bar centre control, is shown in Fig. 12. The (torque) demand is proportional to the lateral acceleration, rather than being specified manually. The feedback comes from the sum of the ARB moments on the three axles (through the load cells mounted on top of the actuators).

![Fig. 12: Roll moment control architecture.](image)
The lateral acceleration gain $K_{xy}$ is selected so that at maximum lateral acceleration (in steady state), the inward roll angle is maximum at approximately 6 degrees and therefore the lateral load transfer is minimum. Fig. 13 shows the lateral acceleration gain selection graph. In order to allow for some overshoot, a value of $K_{xy} = 80000 \text{ kg.m}$ was chosen.

![Fig. 13: Lateral acceleration gain selection graph.](image)

The PID gains were selected using root-locus and pole placement techniques in order to obtain a satisfactory step response. The root locus plot of the closed loop system is shown in Fig. 14. The servo-valve poles have an open-loop frequency of approximately 66 rad/s or 10.5Hz. No derivative gain was included as it brought little benefit.

![Fig. 14: Root locus plot for roll moment control.](image)

Very little integral gain is necessary to remove the steady state error and to avoid too much overshoot, as shown in the step response for the selected gains in Fig. 15. The maximum overshoot is 11.72%, the rise time (to 90% of the steady-state value) is approximately 0.8s and the settling time (to within +/-2% of the steady-state value) is approximately 1.9s. The performance of roll moment control very good, the response times are very fast for little overshoot.

![Fig. 15: Step response for roll moment control.](image)

The roll moment control strategy was tested with the vehicle at speed, performing different types of manoeuvres. Fig. 16 shows both measured and predicted results when the (unladen) vehicle was performing a slalom manoeuvre. Rather than using the signal from the lateral accelerometer for control, an estimated value of lateral acceleration, based on speed and steering angle readings was used. This was computed on the GC and fed back to the LC in real time. The reason for this is that the signal from the accelerometer has a roll acceleration component in addition to the cornering component. This generates feedback between the plant output (ARB moment) and the reference (torque demand) which is not included in the model and can lead to instability.

![Fig. 16: ARB moments under roll moment control.](image)

The agreement between predicted and measured results is very good, the ARB moments on both leading and middle axle responding very well to the torque demand. The predicted ARB moment is an average of the moments on the three axles, as expected.
5. CONCLUSION

An active roll control system has been designed and implemented on an experimental vehicle. The system uses active anti-roll bars with hydraulic actuators controlled by servo-valves.

Roll moment is used to lean the vehicle into the corner, thereby reducing the lateral load transfer. The control system was designed using a simple yaw-roll model combining a bicycle model of the tractor unit and a roll-plane model of the trailer unit. Although the model does not take into account the coupling between the yaw and roll degrees of freedom, it was validated using experimental data logged on the vehicle, with which it agreed very well.

The control strategy was implemented on the experimental vehicle with a distributed control architecture between a global controller and 4 local controllers. More sophisticated control strategies will be investigated using the GC Simulink/xPC interface for implementation.

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REFERENCES


APPENDIX

Notations for roll plane model of trailer unit

- $m$: sprung mass of trailer unit
- $I_b$: sprung mass moment of inertia about roll centre
- $C_b$: suspension roll damping
- $K_b$: suspension roll stiffness
- $\phi$: body roll angle of trailer sprung mass
- $\phi_{arb}$: anti-roll bar angle
- $h$: CG height of sprung mass above roll centre
- $T_{act}$: torque provided by actuators on ARB
- $I_{arb}$: anti-roll bar moment of inertia about roll centre
- $K_{arb}$: anti-roll bar roll stiffness
- $y_a$: actuator extension
- $d_a$: distance to actuators from suspension
- $d_d$: distance to dampers from suspension
- $d_s$: distance to air springs from suspension
- $d_t$: half track width
- $F_a$: actuator force
- $A$: piston area
- $\Delta P$: differential pressure across piston
- $Q_L$: load flow
- $C_{lp}$: total leakage coefficient (internal and external) of piston
- $V_i$: total volume of trapped oil in system
- $\beta_e$: effective bulk modulus of hydraulic oil
- $K_z$: servo-valve flow gain coefficient
- $K_{pe}$: servo-valve total flow pressure coefficient ($K_{pe} = K_p + C_{lp}$)