An Investigation of Roll Control System Design for Articulated Heavy Vehicles

D.J.M. Sampson and D. Cebon
Cambridge University Engineering Department

Department of Engineering, University of Cambridge,
Trumpington Street, Cambridge, CB2 1PZ, United Kingdom.
Phone : +44-1223-332600
Fax : +44-1223-332662
E-mail : djms3@eng.cam.ac.uk

This paper investigates modelling requirements and design issues for developing active roll control systems for heavy commercial vehicles. A flexible, general methodology suitable for generating models of the yaw-roll dynamics of multiple-unit articulated vehicles is presented. A state feedback roll control system for a tractor semi-trailer featuring torsionally flexible tractor and trailer units is designed using the linear quadratic regulator method. The trade-offs involved in the control system design are discussed. Active roll control is shown to offer worthwhile improvements in roll performance, and the use of more sophisticated roll control schemes promises additional gains in future.

Keywords : heavy vehicle, articulated vehicle, rollover, active roll control, active suspension

1. INTRODUCTION

1.1 Background

Rollover of heavy goods vehicles is a serious problem worldwide. In 1993, 545 heavy goods vehicles were reported to be involved in rollover accidents in the UK [1]. The average cost of each of these accidents to vehicle operators has been estimated at between £75,000 and £100,000 [2]. These costs include recovery and repair of the vehicle, product loss, and road repair and re-surfacing. There are also costs attributable to expenditure on hospitals and emergency services, and to social security benefits payed as a result of these accidents, that are in addition to the figures quoted above.

Recent studies indicate that most rollover accidents involve articulated vehicles, and occur on highways [3]. 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.

There has been significant research activity into using advanced suspension systems to control and improve the ride, roll and handling dynamics of automobiles. However, the application of advanced suspension systems to heavy goods vehicles, particularly to control roll and handling dynamics, has been researched to a relatively small degree [3,4].

Yaw rates and lateral accelerations of automobiles can be considerably greater than those of heavy goods vehicles. Heavy vehicles typically feature large payloads, high centres of gravity and multiple vehicle units, and their cornering performance is limited by the vehicle's rollover threshold, rather than by the limit of adhesion of the tyres. Thus, conclusions from research into the use of advanced suspension systems on cars can not simply be applied to trucks.

1.2 Previous research

Dunwoody [5] 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 required the measurement of the trailer lateral acceleration and the relative roll angle between the tractor and the trailer. The study stated that such a system could raise the static rollover threshold by 20-30%.

Lin et al. [6,7] 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 feedback, lateral acceleration feedback and load transfer feedback were investigated. Control gains were selected by pole placement. The authors recommended using 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 reponse; 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. [6,7] then investigated roll control system design 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. [6,8] also simulated the performance of a rigid tractor semi-trailer equipped with a roll control system based on lateral acceleration feedback, using a linear vehicle model and control gains selected using an ad hoc approach. The study found that such a system can reduce steady-state and transient load transfer for a range of manoeuvres. The authors recommended investigating the influence of vehicle frame flexibility on control system performance, and noted that more rigorous roll control system design methodologies for articulated vehicles are required.

2. MODELLING THE YAW-ROLL DYNAMICS OF ARTICULATED VEHICLES

2.1 Model requirements

In order to investigate roll control strategies for articulated commercial vehicles with arbitrary numbers of vehicle units, it was necessary to develop a modelling methodology for deriving the equations of motion of vehicle models with suitable complexity.

The vehicle models must be capable of capturing the essential handling and roll dynamics of the vehicle. Other vehicle motions, such as bounce and pitch, are of secondary importance.

The models must be capable of representing the dynamics of a range of vehicle couplings – the A-coupling (“pintle hitch”), the B-coupling (“fifth wheel”) and the C-coupling (“converter dolly”) – as well as the torsional flexibility of vehicle frames.

The model should be simple enough that the roll control system designer retains sufficient physical insight into the behaviour of the system.

2.2 Model formulation

The vehicle modelling method is based on the linear single unit yaw-roll vehicle model developed by Segel [9], adapted to account for the interaction between connected vehicle units. It is effectively a generalisation of the rigid tractor semi-trailer model used by Lin [6,8].

The vehicle of interest is decomposed into generic vehicle units, each representing a section of the vehicle. The sprung and unsprung masses of each vehicle unit are lumped into a single mass, with yaw, sideslip and roll freedoms. The axles of each vehicle unit are considered to be a single rigid body, with flexible tyres that can roll with respect to the roll centre. The sprung mass rolls about the roll centre, and is restrained by the torsional stiffness and damping of the suspension. A control torque, representing the torque applied by the active roll control system, also acts on the sprung mass. Vehicle units are joined together with couplings that have roll stiffness and yaw stiffness that can range from zero to infinity. Thus, A-couplings, B-couplings, C-couplings and torsional frame flexibility can all be modelled by selecting the appropriate coupling stiffnesses.

Each physical vehicle unit of an articulated vehicle is represented by one or more generic vehicle units in the model. For example, a tractor unit with a flexible frame is represented by two generic vehicle units – one for the steer axle and front structure of the tractor, and another for the drive axle(s) and rear structure. These two vehicle units are coupled with a torsional spring representing the flexibility of the chassis between the steer and drive axles.

Each generic vehicle unit has four equations of motion:

Lateral force equation:

$$\begin{aligned} m_{s,i} h_i \ddot{\phi}_i + m_i v \dot{\beta}_i &= \\ &Y_{\beta,i}\beta_i + (Y_{r,i} - m_i v)\dot{\psi}_i + Y_{\delta,i}\delta_i + F_{i-1} - F_i \end{aligned} \tag{1}$$

Yaw moment equation:

$$\begin{aligned} -I_{xz,i}\ddot{\phi}_i + I_{z,i}\ddot{\psi}_i &= \\ &N_{\beta,i}\beta_i + N_{r,i}\dot{\psi}_i + N_{\delta,i}\delta_i + x_{f,i}F_{i-1} + x_{r,i}F_i \\ &+ K_{\psi,i-1}(\psi_{i-1} - \psi_i) - K_{\psi,i}(\psi_i - \psi_{i+1}) \end{aligned} \tag{2}$$

Sprung mass roll moment equation:

$$\begin{aligned} I_{x,i}\ddot{\phi}_i + m_{s,i}h_i v\dot{\beta}_i - I_{xz,i}\ddot{\psi}_i &= \\ &m_{s,i} g h_i \phi_i - K_i(\phi_i - \phi_{t,i}) - L_i(\dot{\phi}_i - \dot{\phi}_{t,i}) \\ &- m_{s,i} h_i v \dot{\psi}_i + u_i + z_{f,i}F_{i-1} - z_{r,i}F_i \\ &+ K_{\phi,i-1}(\phi_{i-1} - \phi_i) - K_{\phi,i}(\phi_i - \phi_{\phi i+1}) \end{aligned} \tag{3}$$

Unsprung mass roll moment equation:

$$K_{t,i}\phi_{t,i} = K_i(\phi_i - \phi_{t,i}) + L_i(\dot{\phi}_i - \dot{\phi}_{t,i}) - u_i \tag{4}$$

(Notation is detailed in the Appendices.)

Furthermore, there is a kinematic constraint at each coupling: the velocity of the articulation point, whether viewed from the vehicle unit fore or aft of that point, must be the same.

Kinematic constraint equation:

$$\begin{aligned} &\beta_i - \beta_{i+1} + \frac{z_{r,i}}{v}\dot{\phi}_i - \frac{z_{f,i+1}}{v}\dot{\phi}_{i+1} \\ &\quad - \frac{x_{r,i}}{v}\dot{\psi}_i - \frac{x_{f,i+1}}{v}\dot{\psi}_{i+1} + \psi_i - \psi_{i+1} = 0 \end{aligned} \tag{5}$$

These equations feature lateral coupling forces F_i. It is possible to eliminate these internal constraint forces automatically from Eqs. 1-3, thereby generating the equations of motion of a vehicle system with any number of units. The equations of motion can be written in state-space form:

$$\mathbf{M}\dot{\mathbf{x}} = \mathbf{N}\mathbf{x} + \mathbf{G}\mathbf{u} + \mathbf{G_d}\delta \tag{6}$$

In Eq. 6, $\mathbf{u}$ is the control torque vector, δ is the steering input vector, and $\mathbf{x}$ is the state vector:

$$\mathbf{x} = \begin{bmatrix}\mathbf{x_1} & \mathbf{x_2} & \dots & \mathbf{x_n}\end{bmatrix}^\mathbf{T} \quad (7)$$

$$\mathbf{x_i} = \begin{bmatrix}\phi_i & \dot{\phi}_i & \beta_i & \psi_i & \dot{\psi}_i & \phi_{t,i}\end{bmatrix}^\mathbf{T} \quad (8)$$

With the equations of motion in state-space form, it is possible to perform numerical simulations of the transient, harmonic and steady-state responses of the system, and to design and simulate roll control systems for articulated heavy vehicles.

The linearised tyre and suspension models used in the derivation of the equations of motion are approximate. However, it is possible to include non-linear tyre and suspension characteristics within this modelling framework, and to perform simulations of vehicle transient responses using more realistic component models.

3. ACTIVE ROLL CONTROL FOR A FLEXIBLE TRACTOR SEMI-TRAILER

This section details an investigation into the design of an active roll control system for a tractor semi-trailer, using a vehicle model generated using the technique detailed in Section 2.2.

3.1 Vehicle description

The vehicle used for these simulations was a fully-laden four-axle tractor semi-trailer, with a gross mass of 32.5 tonnes. The vehicle consisted of a tractor unit with a single drive axle, coupled by a fifth wheel coupling to a flat-bed twin-axle trailer unit. Vehicle parameters were taken from [10]. The vehicle was modelled using three generic vehicle units (two for the tractor and one for the trailer) or two generic vehicle units (tractor and trailer), depending on whether the torsional compliance of the tractor unit was included.

3.2 Steering and speed inputs

Two steering inputs were selected to investigate the steady-state and transient performance of roll control system designs:

- a ramp steering input, applied over 4 seconds and then held constant;
- a lane change steering input.

Both steering inputs featured peaks in steered wheel angle of 1°. The vehicle speed was 40 km/h.

3.3 Control system design methodology

A basic state feedback controller was designed by finding an optimal controller based on a linear quadratic regulator (LQR).

For zero steering input*, the state space representation of the vehicle system (Eq. 6) can be rewritten by defining matrices **A** and **B**:

$$\dot{\mathbf{x}} = \mathbf{M}^{-1}\mathbf{N}\mathbf{x} + \mathbf{M}^{-1}\mathbf{G}\mathbf{u} \equiv \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u} \quad (9)$$

The aim of the control system is to minimise lateral load transfer in response to steering inputs, since it is excessive lateral load transfer that causes vehicles to roll over.

* Within the LQR framework, the steering input is modelled as a disturbance to the vehicle system.

Lateral load transfer has components from several sources. Some of these terms – from centripetal acceleration during cornering, and lateral coupling forces from adjacent vehicle units – are set by the vehicle dimensions and the trajectory of the vehicle around a corner. Other terms – from vehicle body roll, torques applied by adjacent vehicle units through stiff couplings, and roll inertia terms – are influenced strongly by the performance of the suspension and the active roll control system.

The lateral load transfer of a vehicle unit can be normalised by dividing by half of that vehicle unit's total axle load. This gives an indication of the proximity of the vehicle unit to rollover; rollover occurs when the normalised load transfer reaches ±1. It is then possible to form a vector **y** from the normalised lateral load transfers of each vehicle unit:

$$\mathbf{y} = \begin{bmatrix}LT_1 & LT_2 & \cdots & LT_n\end{bmatrix}^\mathbf{T} \quad (10)$$

This vector can be expressed in terms of the the state vector **x** and the control torques **u**:

$$\mathbf{y} = \mathbf{C}\mathbf{x} + \mathbf{D}\mathbf{u} \quad (11)$$

The LQR problem is to find a control vector **u** that minimises the quadratic performance index J:

$$J = \int_0^t (\mathbf{y}^\mathbf{T}\mathbf{Q}\mathbf{y} + \mathbf{u}^\mathbf{T}\mathbf{R}\mathbf{u})dt \quad (12)$$

Q and **R** are weighting matrices chosen by the control system designer. **Q** penalises the output **y** (the normalised load transfers), and **R** penalises control action **u** (the control torques applied to each vehicle unit). By carefully varying the elements of **Q** and **R**, it is possible to balance performance and control action requirements at each axle. The LQR methodology ensures that the optimal system will keep load transfer **y** "small" without "excessive" control action **u**.

3.4 Control for a rigid tractor semi-trailer

In this section, the performance of a vehicle fitted with an active roll control system is compared with that of a passively-suspended vehicle. A passive vehicle leans out of the turn during cornering and transfers load to the outside tyres due to the effect of the centripetal acceleration.

Excessive trailer load transfer on the trailer axles is the cause of most rollover accidents in articulated vehicles. Therefore, using the LQR methodology, a controller was designed initially to penalise (by choosing the elements of **Q**) trailer load transfer more heavily than tractor load transfer. The response of the vehicle fitted with this controller is shown in Figs. 1-4. The active control system can be seen to tilt the tractor and trailer into turns (Figs. 1 and 3), reducing the lateral load transfer (Figs. 2 and 4). The tractor is tilted into the turn more than the trailer. Thus, a negative overturning moment is applied to the trailer through the fifth wheel coupling, reducing lateral load transfer. (This technique is known as 'roll moment co-operation'.) This particular scheme reduces trailer load transfer significantly, as specified by the weighting matrices. In this case the

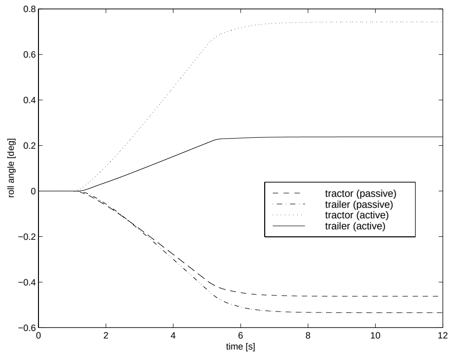

Fig. 1: Roll angle vs. time for passive and active anti-roll suspensions, subject to a ramped steering input over 4 seconds.

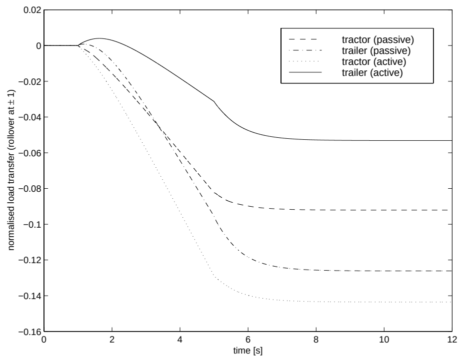

Fig. 2: Variation of lateral load transfer with time for passive and active anti-roll suspensions, subject to a ramped steering input over 4 seconds.

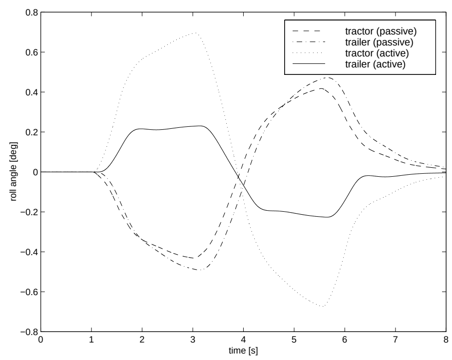

Fig. 3: Roll angle vs. time for passive and active anti-roll suspensions, subject to a lane change steering input.

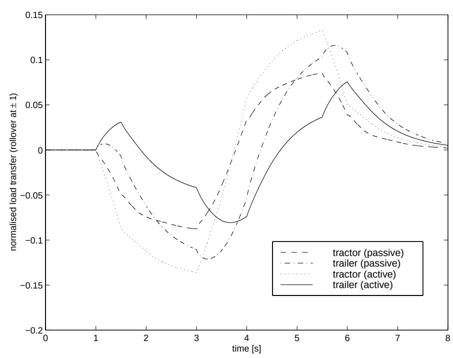

Fig. 4: Variation of lateral load transfer with time for passive and active anti-roll suspensions, subject to a lane change steering input.

control torques at each vehicle unit depended on the vehicle states of both the tractor and trailer units because the tractor and trailer roll motions are strongly coupled.

It is similarly possible to penalise tractor load transfer more heavily than trailer load transfer by selecting a different weighting matrix **Q**. Such a system was found to tilt both the tractor and trailer into the turn, but the trailer was tilted more steeply than the tractor. Thus, a negative overturning moment was applied to the tractor through the coupling, and its load transfer was reduced. In practice, the control system designer must balance load transfer requirements at all vehicle units by selecting the elements of **Q**.

It is notable that lateral load transfer did not converge to zero in response to a steady non-zero steering input. This performance limitation is a consequence of the way the LQR method treats the steering input as a disturbance. However, it is possible to add integral action to the controller to reduce this effect for low lateral acceleration manoeuvres.

The roll control system attempts to nullify load transfer by tilting the vehicle units into the turn. In practice, the angle of tilt can not be increased arbitrarily due to limits on suspension travel and actuator power. Within the LQR methodology, excessive control action is penalised by **R**. Figure 5 shows that, when the weighting on control action in the performance index J is increased, the system reduces power consumption by tilting the vehicle units less. This also reduces performance, although the improvements over the passive case are still worthwhile.

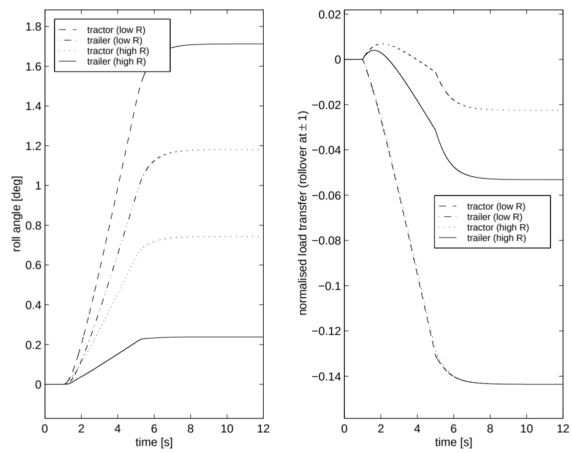

Fig. 5: Variation in the roll angle and lateral load transfer with an increased weighting **R** in the controller design.

3.5 Influence of frame flexibility on control system design

The effect of the torsional flexibility of the trailer frame was investigated by adjusting the stiffness K_ϕ from the rigid frame value (1 MN·m/rad) to 100 kN·m/rad and 10 kN·m/rad. K_ϕ represents the combined stiffness of the structural elements between the tractor drive axle and the trailer axles, i.e. the coupling and the trailer frame. An LQR roll controller was designed for the flexible vehicle. Figs. 6-7 show the variation of roll angle response and load transfer response with K_ϕ.

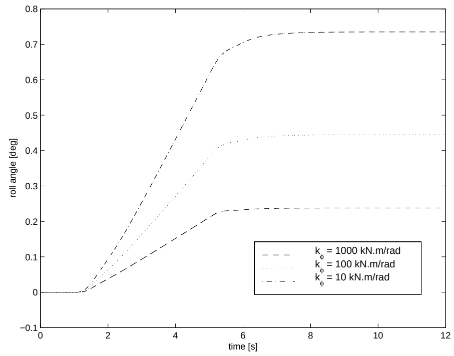

Fig. 6: Variation of trailer roll angle response for various values of the torsional stiffness of the trailer frame. As the torsional flexibility of the trailer increases, the control system is forced to tilt the trailer further into the turn to reduce load transfer.

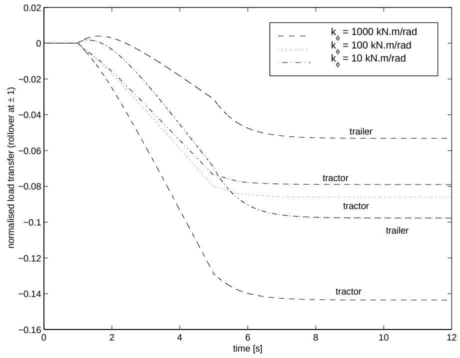

Fig. 7: Variation of lateral load transfer response for various values of the torsional stiffness of the trailer frame. The increased torsional flexibility of the trailer reduces the roll moment at the fifth wheel, lowering the potential for using roll moment co-operation. The result was an increase in the load transfer of the trailer and a decrease in the load transfer of the tractor.

The potential for exploiting roll moment co-operation is greatly diminished by the decrease in vehicle stiffness. The ability of the tractor control torques to influence the roll response of the trailer (and vice versa) is reduced.

A similar analysis investigated the effect of tractor frame flexibility on control system design and performance, assuming a torsional stiffness of 100 kN·m/rad. The effect of tractor frame flexibility was found to be less pronounced than the effect of trailer frame flexibility. It reduced the ability of tractor steer axle control torques to influence the roll response of the rear section of the tractor, and hence the trailer.

3.6 Consequences for control system design

Active roll control systems can provide useful reductions in lateral load transfer for articulated heavy vehicles, although the performance gains are limited by practical considerations. A trade-off between power consumption and performance is achieved by varying the weighting matrices **Q** and **R** in Eq. 12.

The LQR methodology is a powerful and convenient design framework for selecting control system gains for a state feedback controller.

For vehicles with stiff frames and stiff couplings (e.g. fifth wheel couplings) between the vehicle units, the roll motions of the vehicle units are strongly coupled, and a centralised roll control system is appropriate. As frame stiffness and coupling stiffness decrease, vehicle units can rely less on adjacent units to provide roll torques through the vehicle couplings, and a more decentralised roll control system is required. In the limiting case of negligible coupling stiffness (e.g. A-coupling, or C-dolly), no roll moment co-operation is possible.

3.7 Future enhancements

While the simple state feedback controller described above provides worthwhile reductions in lateral load transfer, a more sophisticated feedback control scheme would further enhance the performance of the system. Integral control action would reduce steady-state load responses to non-zero steering inputs, while enhanced derivative action would give better transient performance. A limited state feedback controller, with observers to estimate unmeasured states, could reduce instrumentation requirements.

4. CONCLUSIONS

1. A general, systematic technique for developing linear models of the yaw-roll dynamics of articulated vehicles has been developed. This technique can be used to assemble the equations of motion for an arbitrarily long vehicle, and can represent a wide range of vehicle couplings as well as the torsional flexibility of vehicle frames.
2. Although the linear vehicle model has limitations, it is particularly suitable for control system design as it allows the designer to retain a physical insight into the behaviour of the vehicle.
3. A basic state feedback roll control system was designed for a flexible tractor semi-trailer. An active roll control system can reduce steady-state and peak transient load transfer compared with the passive case for a series of manoeuvres, improving the rollover safety of the vehicle.
4. The influence of frame flexibility on controller design was investigated. Vehicles with flexible tractor and trailer frames require more decentralised control systems, as frame flexibility reduces the opportunity for using roll moment co-operation between vehicle units.
5. While relatively simple active roll control schemes can provide improvements in roll performance, the use of more sophisticated roll control schemes promises additional benefits.
6. The practical design issues of instrumentation requirements and the effect of hardware performance limitations on achievable roll response are the subject of current investigations.

ACKNOWLEDGEMENTS

The authors wish to thank Dr D.E. Davison of Cambridge University Engineering Department for his advice on control system design.

The work described was funded by EPSRC and Cambridge Vehicle Dynamics Consortium. At the time of writing, Cambridge Vehicle Dynamics Consortium consists of the Universities of Cambridge, Cranfield and Nottingham together with the following industrial partners from the European heavy vehicle industry: Tinsley Bridge Ltd, Meritor HSV, Koni BV, DERA, Dunlop Tyres, Shell UK Ltd, Volvo Trucks and Crane Fruehauf.

Mr Sampson would also like to thank the Cambridge Australia Trust and the Committee of Vice-Chancellors and Principals of the Universities of the United Kingdom for their financial support.

REFERENCES

[1] Anon., "Road accidents in Great Britain." Department of Transport, HMSO, 1994.

[2] Harris R., "Cost of roll-over accidents." Personal communication, 1995.

[3] Kusters L.J.J., "Increasing roll-over safety of commercial vehicles by application of electronic systems." "Smart Vehicles." Pauwelussen J.P. and Pacejka H.B. ed., Swets and Zeitlinger, 1995, pp. 362-377.

[4] Besinger F.H., Cebon D. and Cole D.J., "Force control of a semi-active damper." VSD, Vol. 24 (No. 1), 1995, pp. 695-723.

[5] Dunwoody A.B., "Active roll control of a semi-trailer." SAE Transactions, SAE 933045 (Nov), 1993.

[6] Lin R.C., "An investigation of active roll control for heavy vehicle suspensions." PhD thesis, Cambridge University Engineering Department, 1994.

[7] Lin R.C., Cebon D. and Cole D.J., "Optimal roll-control of a single-unit lorry." J. Auto. Eng., Proc. IMechE, D05294, 1996, pp. 45-55.

[8] Lin R.C., Cebon D. and Cole D.J., "Active roll control of articulated vehicles." VSD, Vol. 26 (No. 1), 1996, pp. 17-43.

[9] Segel L., "Theoretical prediction and experimental substantiation of the response of an automobile to steering control." Proc. IMechE Automobile Division, 1956-1957, pp. 310-330.

[10] Cole D.J., "Measurement and analysis of dynamic tyre forces generated by lorries." PhD thesis, Cambridge University Engineering Department, 1990.

APPENDICES

Notation

ψ_i	Heading angle
$\dot{\psi}_i$	Yaw rate around z axis
ϕ_i	Sprung mass roll angle
$\phi_{t,i}$	Unsprung mass roll angle
β_i	Sideslip angle at centre of mass
δ_i	Steer angle
u_i	Control torque
F_i	Lateral coupling force
m_i	Total mass
$m_{s,i}$	Sprung mass
$I_{x,i}$	Roll moment of inertia
$I_{z,i}$	Yaw moment of inertia
k_i	Suspension roll stiffness
l_i	Suspension roll damping
k_t	Tyre roll stiffness
$c_{i,j}$	Combined tyre cornering stiffness
v	Vehicle speed

The tyre coefficients in Eqs. 1-3 are given by:

$$Y_{\beta,i} = \sum_j c_{i,j} \qquad Y_{r,i} = \sum_j \frac{a_{i,j}c_{i,j}}{v} \qquad Y_{\delta,i} = -c_{i,1}$$

$$N_{\beta,i} = \sum_j a_{i,j}c_{i,j} \qquad N_{r,i} = \sum_j \frac{a_{i,j}^2 c_{i,j}}{v} \qquad N_{\delta,i} = -a_{i,1}c_{i,1}$$

The subscript i denotes vehicle unit i or coupling i. Coupling i is the coupling between vehicle units i and $i+1$. The subscript j denotes axle j. Vehicle units, couplings and axles are numbered from front to rear.

Vehicle axis systems and dimensions

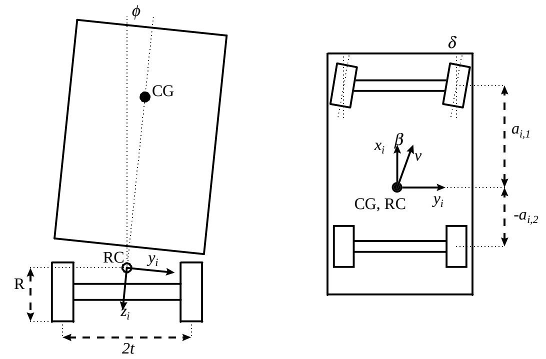

Fig. 8: Axis system for a generic vehicle unit.

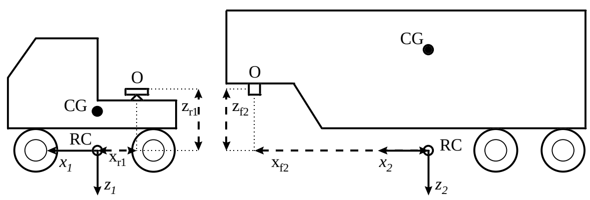

Fig. 9: Axis system for an articulated vehicle.