Automated Vehicle Stability Control for Articulated Vehicles

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Abstract

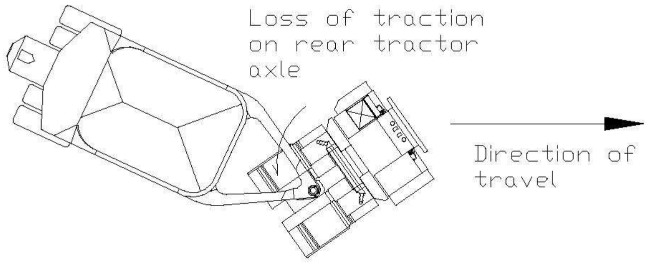
This work investigates the dynamics of heavy articulated vehicles and the problem of yaw-instability caused by loss of traction. A mathematical model of the system is derived and applied to a typical vehicle used in mine haulage applications. Automatic control, based on differential wheel braking, is explored by simulation and shown to be potentially valuable in improving the safety of such vehicles in adverse road conditions.

1 Introduction

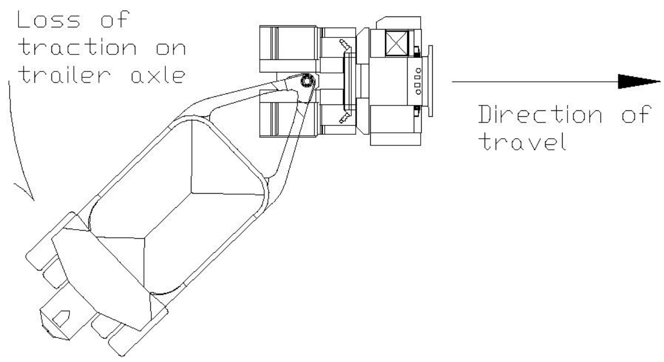
Articulated vehicles have the potential to lose yaw stability under adverse road conditions. If the driver cannot take corrective action quickly enough then the vehicle may slew into the path of oncoming vehicles or leave the roadway. Tractor-trailer combinations used for haulage on mine sites are one example of applications where this safety problem is gaining attention. The belly-dumper shown in Figure 1is typical of such vehicles.

Figure 1: Caterpillar 776D tractor and trailer. Engine power 700KW, 56 tonnes (empty), load 160 tonnes.

Two unstable dynamic modes are possible. These are referred to as "jack-knife" and "trailer-swing". Loss of traction of the rear wheels of the tractor causes a jack-knife, whereas traction loss on the trailer wheels results in trailer swing. When traction is lost on both sets of wheels simultaneously the tractor and trailer remain in-line and the entire vehicle will experience yaw.
The aim of this project is to i) identify when unstable operation is likely to occur, ii) evaluate what action is required to prevent the vehicle from reaching an unstable state, iii) examine an automatic control strategy which allows recovery from yaw instability.



(a) Jack-knife (see video Classic Jackknife.mpg)



b) Trailer swing (see video Trailer Swing.mpg)

Figure 2: Unstable yaw modes: a) "jack-knife" b) "trailer swing"

2 Automatic control strategies

It is generally agreed that anti-lock braking systems are useful in preventing yaw instability because they prevent brake application beyond the peak friction coefficient on the tyre characteristic. In the literature it is also noted that correct design of the coupling between the tractor and trailer ("fifth wheel") is important in prevention of instability. [Kaneko and Kageyama, 2003]

The next step in improving resistance to yaw instabilities is to take corrective action through application yaw moments. Human drivers apply corrective yaw moments through steering. This leaves differential braking (right / left side) as the means by which automatic correction may be applied. This strategy is well advanced in passenger vehicles [Kin et al 2003] , [Esmailzadeh et al 2003] but not currently used for articulated vehicles.

3 Tyre mechanics

Success in predicting the likelihood of yaw instability in a vehicle comes from an understanding of tyre mechanics. These principles are explained in various texts [Gilespi 1992] , [Genta, 1997], [Pacekja, 2002].

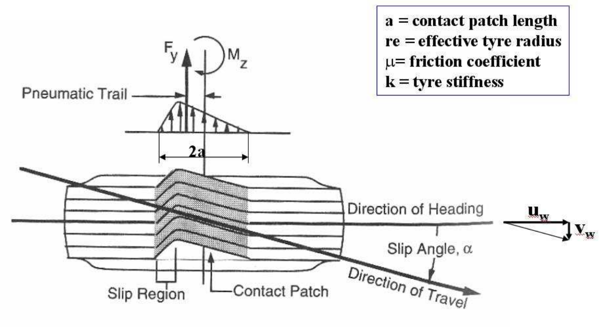


Figure 3: Tyre parameters and state variables.

3.1 Tyre parameters

Critical parameters which determine a tyre's ability to maintain traction on the road are described in Figure 3.
i) Contact patch length ($2a$) : determined by the normal wheel loads and inflation pressure,
ii) Effective tyre rolling radius (r_e): determined by the tyre diameter and inflation pressure,
iii) Coefficient of friction (μ) : determined by road conditions and
iv) Stiffness (k): determined by tyre material properties.

3.2 Variables

Variables which determine the longitudinal (F_x) , lateral friction (F_y) and moment (M_z) acting on the tyre are the longitudinal (u) and lateral (v) velocities at the wheel axle. Wheel slip angle is defined as $\alpha = \tan^{-1}(u/v)$. Figure 4 shows typical tyre force versus slip characteristics. Slip is defined as $f = 1 - \frac{r_e \omega}{u}$, where ω is the angular velocity of the wheel. A positive value of f designates braking and a negative value applies during application of drive torque to the wheels.

3.2 Tyre model

Using the "brush model" developed by Pacekja [2002], lateral tyre characteristics, shown in Figure 5, were developed for the Caterpillar 776D tractor and trailer combination assuming the following operating conditions.

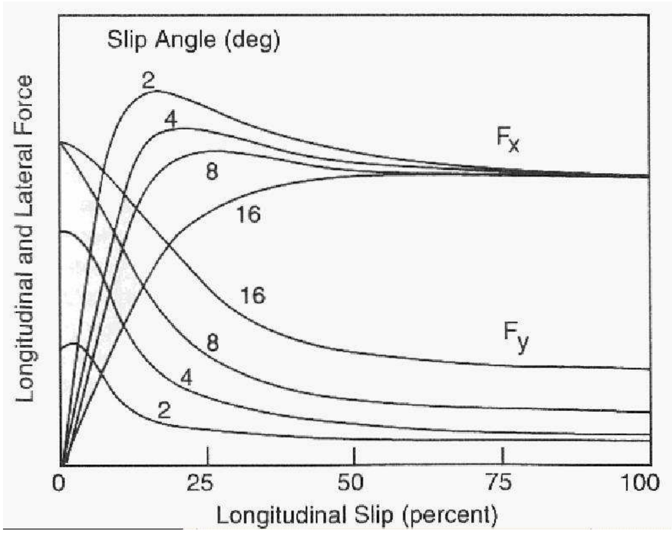


Figure 4: Typical tyre characteristics during braking.

	Tractor	Trailer
Coefficient of friction μ	0.5	0.5
Tyre tread stiffness k [N/m^2]	5e6	7e6

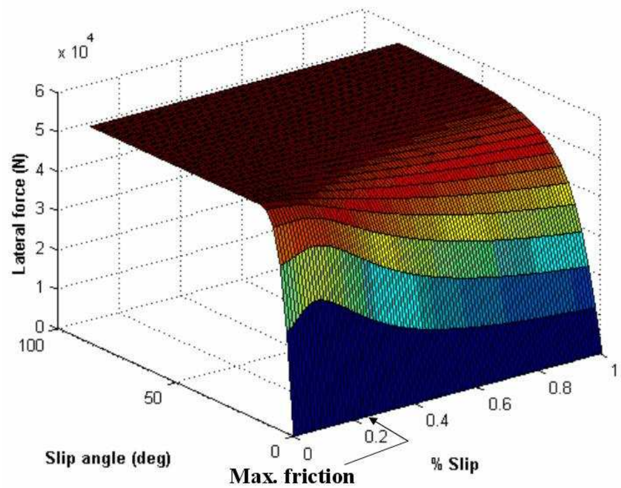


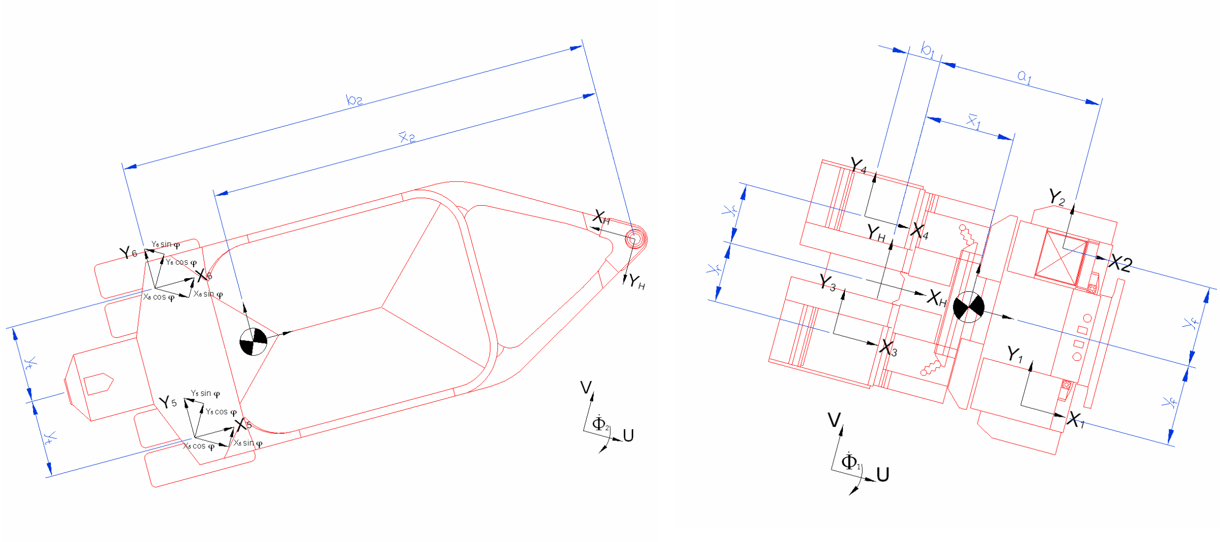
Figure 5: Modelled tyre characteristics Caterpillar 776D tractor and trailer during braking.

4 Rigid body dynamics

Rigid body dynamic, equations of motion (Equation 1) can be formulated using the analysis described by Ellis [1994]. Origin of the frame of reference is located at the hitching point. The x-axis points forward, y-axis to the left and the z-axis vertically upward. Forward and lateral velocities of the origin are designated U and V and the yaw angles of the front and rear of the vehicle, relative to an absolute reference direction are ϕ_1 and ϕ_2. The hitch angle is φ.

Vehicle sideslip angle $\beta = \tan^{-1}\left(\frac{V}{U}\right)$.

Figure 6: Vehicle parameters



		Tractor		Trailer
Dimension				
Distance from hitch to front axle [m]	a_1	3.808		
Distance from hitch to rear axle [m]	b_1	0.762	b_2	14
Distance from hitch to COG [m]	$\bar{x}_1$	1.563	$\bar{x}_2$	10.365
Distance from longitudinal axis to front wheel centres [m]	y_f	2.086		
Distance from longitudinal axis to rear wheel centres [m]	y_r	1.788	y_t	2.29
Mass [kg]	m_1	57000	m_2	69000
Mass moment of inertia [kgm^2]	I_{z1}	3.5e5	I_{z2}	19.4e5
Effective tyre radius [m]	r_e	1.2	r_e	1.2
Area of contact patch [m^2]	a	0.5	a	0.44

Equations of motion : **Equation (1)**

$$\begin{bmatrix} m & 0 & 0 & m_2\bar{x}_2\sin\varphi \\ 0 & m & -m_1\bar{x}_1 & -m_2\bar{x}_2\cos\varphi \\ 0 & -m_1\bar{x}_1 & I_{z1}+m_1\bar{x}_1^2 & 0 \\ m_2\bar{x}_2\sin\varphi & -m_2\bar{x}_2\cos\varphi & 0 & I_{z2}+m_2\bar{x}_2^2 \end{bmatrix} \times \begin{bmatrix} \dot{U} \\ \dot{V} \\ \ddot{\phi}_1 \\ \ddot{\phi}_2 \end{bmatrix}$$

$$= \begin{bmatrix} -mV\dot{\phi}_1 + m_1\bar{x}_1\dot{\phi}_1^2 - m_2\bar{x}_2\dot{\phi}_2^2\cos\varphi + X_f + X_r + X_t\cos\varphi + Y_t\sin\varphi \\ mU\dot{\phi}_1 - m_2\bar{x}_2\dot{\phi}_2^2\sin\varphi + Y_f + Y_r + Y_t\cos\varphi + X_t\sin\varphi \\ -m_1\bar{x}_1 U\dot{\phi}_1 - a_1Y_f + b_1Y_r - y_f(X_1 - X_2) - y_r(X_3 - X_4) + M_1 + M_2 + M_3 + M_4 \\ -m_2\bar{x}_2(U\dot{\phi}_1\cos\varphi + V\dot{\phi}_1\sin\varphi) + b_2Y_t - y_t(X_5 - X_6) + M_5 + M_6 \end{bmatrix}$$

where: $m = m_1 + m_2$ and
X and Y are the longitudinal and lateral tyre forces and M is the aligning moment acting on the tyres.

5 Vehicle model

5.1 Identifying when instability occurs.

An appreciation of the transition from stable to unstable behaviour can be obtained from the steady state, constant speed, outputs of the mathematical model. Predictions of the required steering angle are plotted against lateral acceleration of the centre of gravity. Figure 7 shows this characteristic for the tractor.

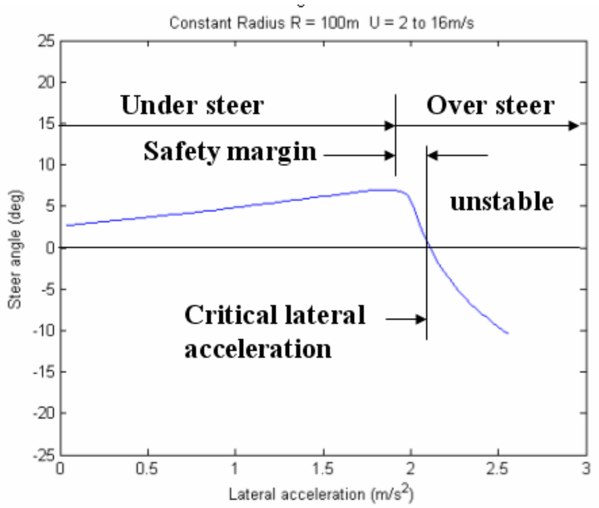


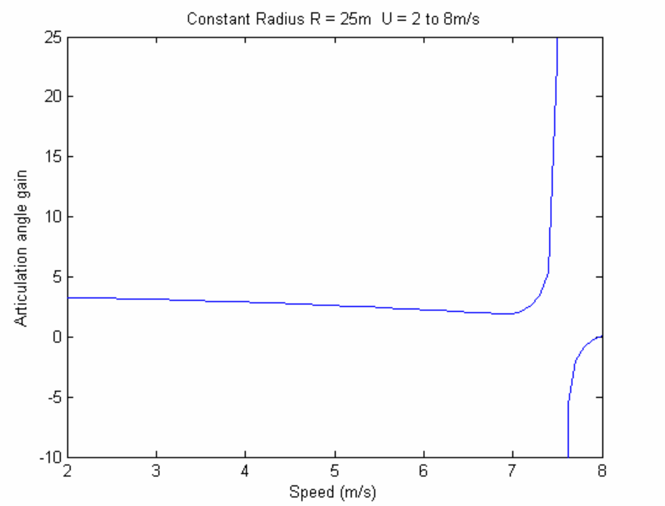
Figure 7: Predictions of steering angle (δ) required on the tractor, for a constant radius (100m) turn , 2 < U <16 m/s.

The initial gradient of the tractor's characteristic curve is constant and positive because the front wheels are relatively heavily loaded compared with the rear.

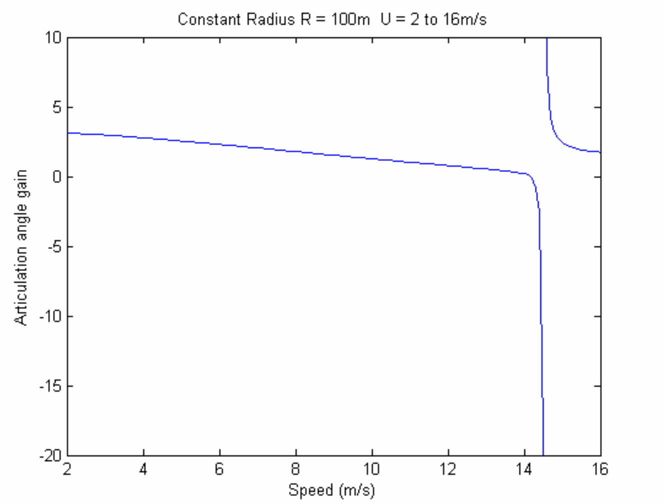
Figure 7 predicts that instability in the tractor motion will occur at lateral accelerations of approximately 2.1 m/s^2. The graph also shows that the precursor to instability is the transition from understeer (positive gradient $K_{us}>0$) to oversteer (negative gradient $K_{us} < 0$) and that an identifiable safety margin exists during which corrective action could be taken to prevent instability.

For the trailer, the rear wheels of the tractor become its steering wheels. A similar characteristic curve to Figure 7, will exist for the trailer. Articulation angle φ is the trailer's effective steering angle and, in this case, the lateral acceleration is that of the trailer's centre of gravity. Because the rear wheels of the trailer are relatively heavily loaded compared with the tractor's rear wheels, then the characteristic of the trailer is to oversteer. Gradient of the characteristic curve starts negative and remains a falling gradient as lateral acceleration increases.

Figures 8 show an example of the articulation gain curve ($K_{\varphi\delta} = \varphi / \delta$) for the Caterpillar 776D tractor and trailer. The first frame corresponds to operating conditions where the critical speed of the tractor has been reached first, causing the tractor to lose stability and jack-knife occurs. The second frame shows that the trailer critical speed was reached first and the vehicle experiences trailer swing. Whether the vehicle experiences jack-knife or trailer swing depends on which critical speed is reached first. When the radius of turn is small, the tractor characteristic curve will cut zero (δ=0) first and the articulation gain will go to plus infinity. For large radius turns the articulation angle will reduce to zero (φ=0) first before the articulation gain heads to minus infinity.



a) Jack-knife condition: articulation gain first approaches positive infinity.



b) Trailer swing condition: articulation gain first crosses zero then approaches minus infinity.

Figure 8: Predictions of articulation gain versus forward speed for a constant radius turn .

Hence the articulation gain curve is a valuable of way of showing which instability is dominant. For either the tractor or trailer to be unstable the vehicle must be travelling at a speed which takes it beyond its critical lateral acceleration. If the tractor is unstable and the trailer stable, the vehicle will jack-knife. If the tractor is stable and trailer is unstable then the tailer will swing.

5.2 Transient response for step steering input.

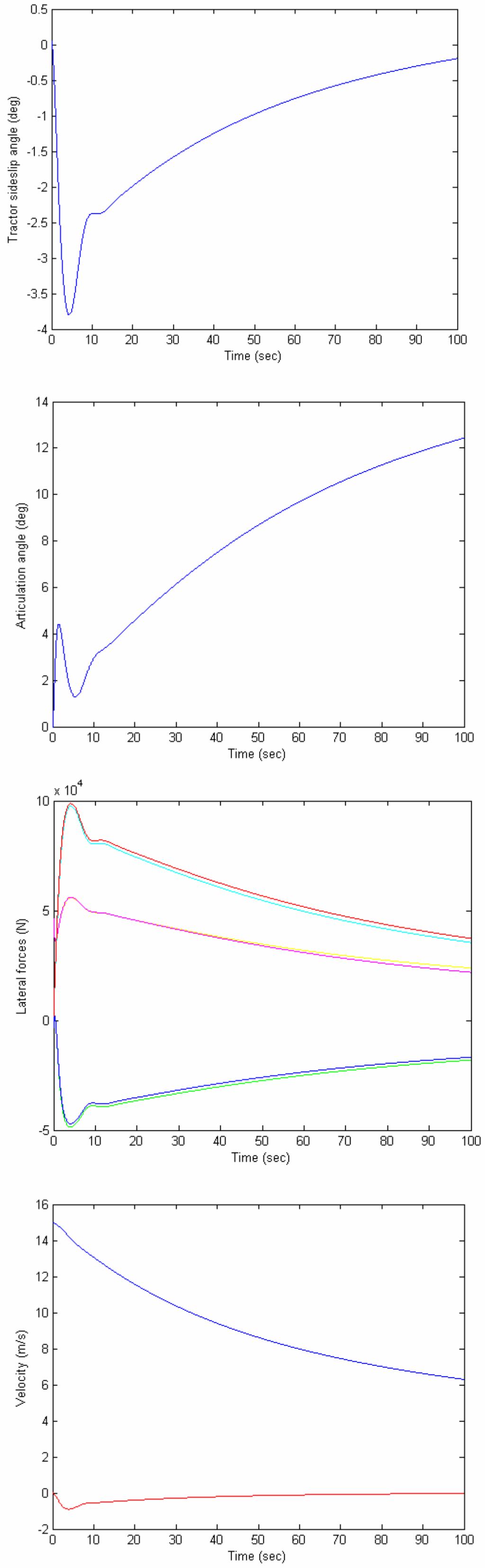


Figure 9: Transient responses for step steering input of δ=6 degrees with forward velocity 15 m/s.

5.3 Transient response where brakes were applied during a turn.

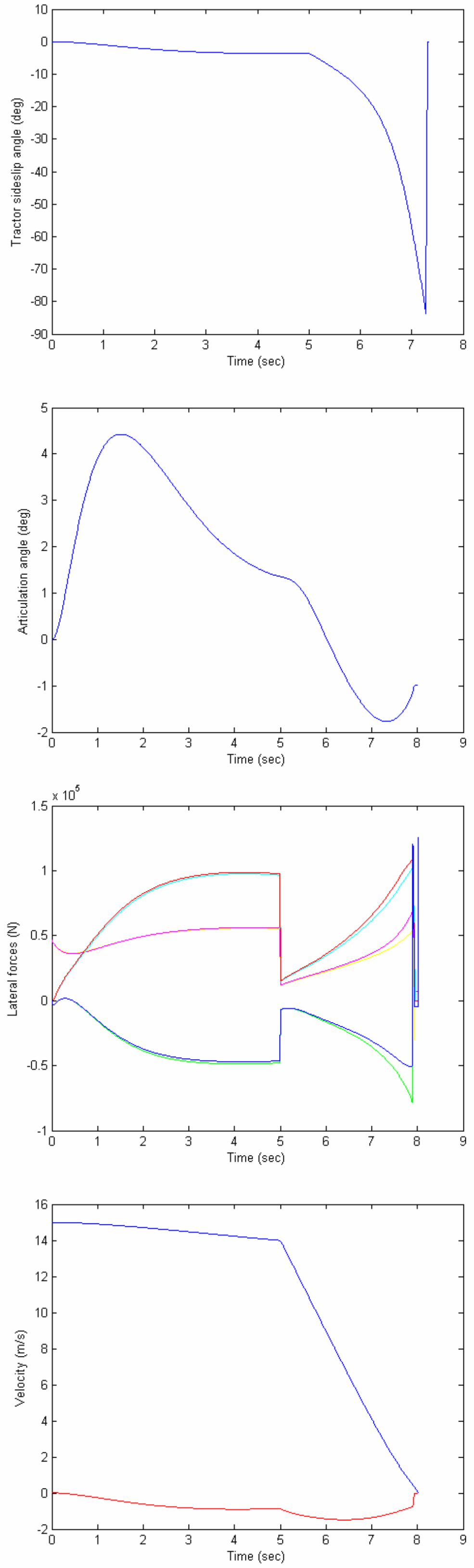
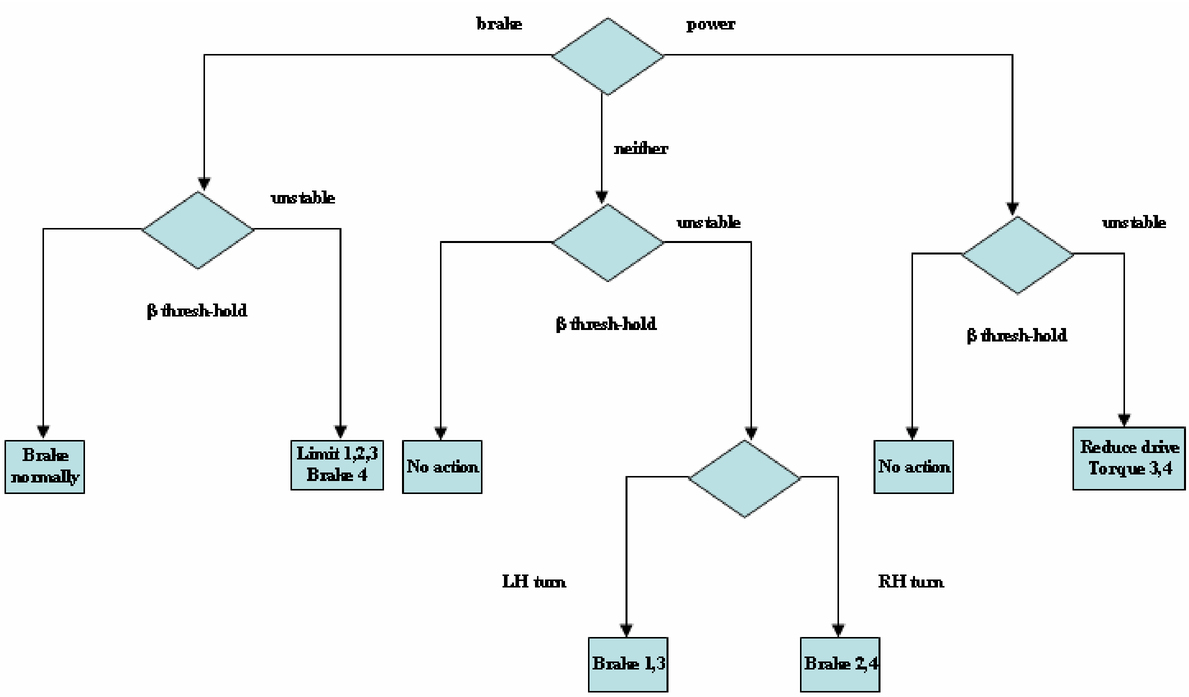
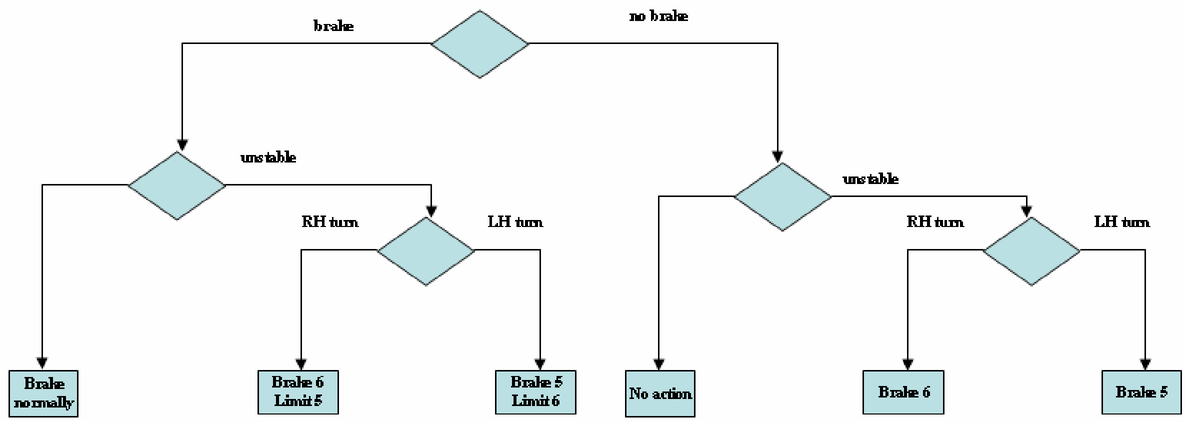


Figure 10: Transient responses for step steering input of δ=6 degrees with forward velocity 15 m/s. Brakes applied to all wheels after 5 seconds.



a) **Tractor control strategy**



b) **Trailer control strategy**

Figure 11 Control strategy: Wheels on the right hand side are labelled 1,3,5 and on the left hand side are labelled 2,4,6, from the from of the vehicle.

5.4 Simulations

Movie simulations of the vehicle are shown in the following clips:

- mov1.avi : shows stable cornering for a steering angle of 6 degrees and speed of 15 m/s.
- mov3.avi: shows unstable cornering with (excessive) steering angle of 11 degrees and speed of 15 m/s. Both tractor and trailer lose stability. They slew together, remaining in-line.
- mov5.avi: shows combined cornering and braking with steering angle 11 degrees and speed 20 m/s. The trailer losses stability.

5.5 Conclusions from the vehicle model

Tractor oversteer begins at the same lateral acceleration for different radii and vehicle speeds. Transition from understeer to oversteer is a leading indicator of unstable behaviour in the tractor. This instability thresh-hold is reduced when friction coefficient reduced.

At low speeds the mode of instability is the jack-knife whereas at high speeds trailer swing appears. The crossover point from jack-knife to trailer swing corresponds to the case where the articulation angle gain is zero at the critical speed of the tractor.

Vehicle response is extremely sensitive to small changes in steering input once a steering threshold was reached.

Figure 9 shows that a step steering input of 6 degrees results in stable behaviour. However a change of steering input from 6 degrees to 7 degrees causes the vehicle to lose stability at 15 m/s. This small change in steering angle corresponds to the transition between understeer and oversteer shown in Figure 7.

Application of brakes during steering exacerbates the stability problem. Figure 10 shows that for 50% braking applied on all wheels the response is stable. However in instances where one pair of wheels locks, unstable behaviour occurs. The mode of instability depends on which pair of wheels lock.

50% braking : front wheels locked – trailer swing
50% braking : rear wheels locked – jack knife
50% braking : trailer wheels locked – trailer swing.

Excess torque on rear tractor wheels, causes slip which results in a jack-knife. Spinning the rear wheels has the same effect as locking them.

6 Control system design and simulation

6.1 Control strategy

Control logic used to prevent yaw instability is shown in Figures 11 a) and b). Both tractor and trailer are monitored separately and corrective differential (left / right side) braking, one either body, is only taken when it approaches instability.

i) Tractor control

Figure 11 a) shows the control strategy applied to the tractor depends on whether brakes, power or neither are being applied. The decision as to whether the tractor is approaching instability is made on the basis of side-slip angle (β) exceeding a predetermined threshold.

- If instability is reached and the driver has not applied the brakes (and therefore does not want to slow the vehicle), the controller only applies braking torques to the wheels on the outside of the turn.
- If the driver has applied the brakes, then the outside wheels are given maximum braking force whilst the inner wheels have their braking limited to achieve maximum lateral force. This allows for overall braking of the vehicle without losing traction or reducing lateral stability.
- To prevent a power-slide the drive wheel torque is reduced.

When control is applied to both tractor wheels on one side, the rear wheel is activated first, and if sufficient stabilising moment has not been achieved, control is then applied to the front wheel.

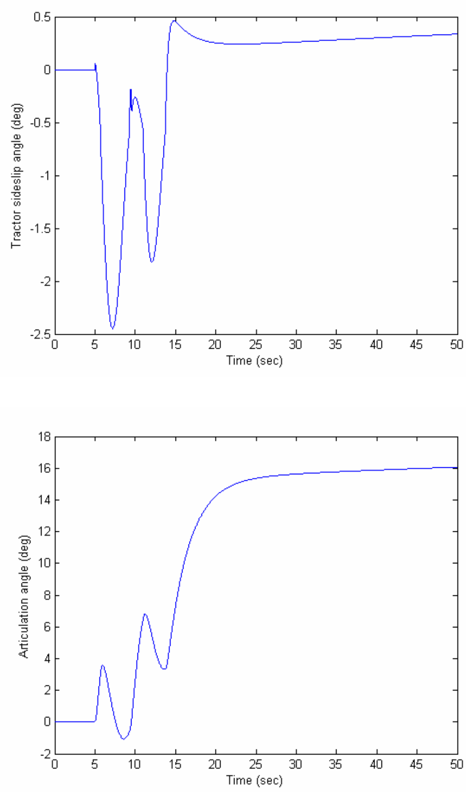
ii) Trailer control

Figure 11 b) shows that the control strategy on the trailer depends on whether the driver is applying the brake.

In the event that the driver applies the brake and the trailer approaches instability, then the brake on the inside of the turn is fully applied and the outside wheel has its braking torque limited to the value which provides maximum lateral force on the tyre.

If no brake has been applied and the trailer approaches instability then full braking load is applied to the wheel on the inside of the turn.

6.2 Simulation



Figures 12: Transient responses for step steering input of δ=20 degrees applied after five seconds with forward velocity 15 m/s.

Figures 12 show that, with the control strategy in place, the stability limits with large steering inputs are greatly enhanced. Figure 13 gives a pictorial representation of the motion of the vehicle under the action of the controller.

Similarly the control system was demonstrated to be effective in instability during braking.

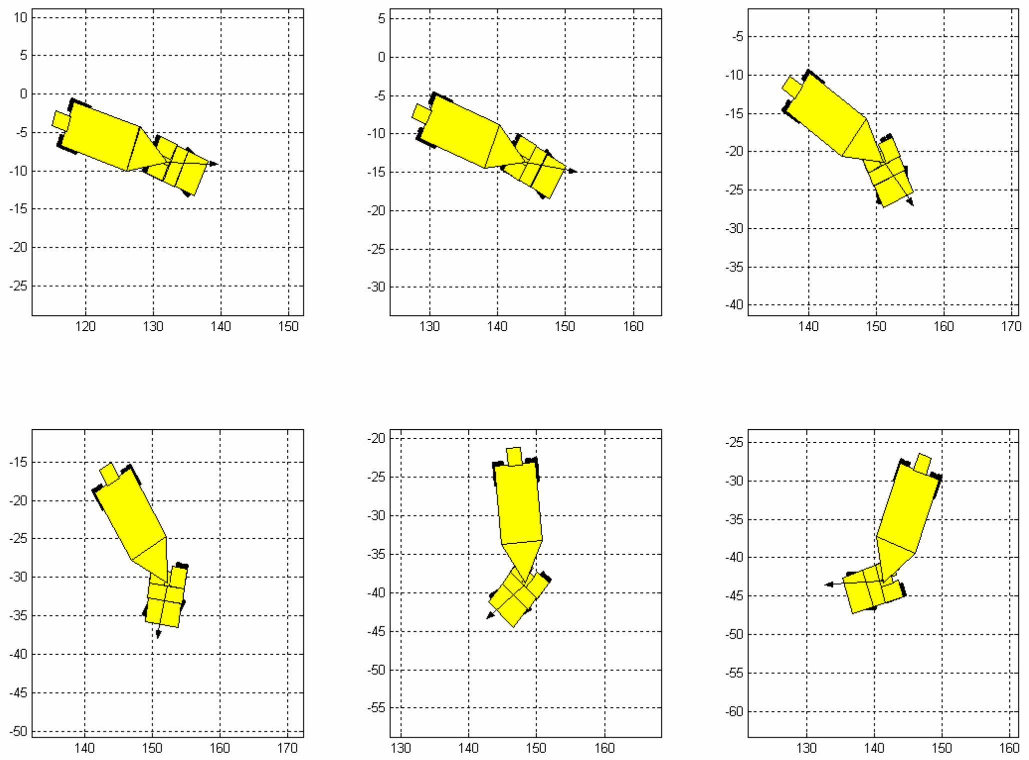


Figure 13: Responses for step steering input of δ=20 degrees applied after five seconds with forward velocity 15 m/s. Vectors show the direction of the velocity vector.

The following movie clips show the improvement to vehicle response under the action of automatic control.

- mov2.avi: shows stable cornering and braking with a steering angle of 6 degrees and speed 15 m/s.
- mov4.avi: shows stable cornering and braking with a steering angle of 11 degrees and speed 15 m/s.
- mov6.avi: shows stable cornering and braking with a steering angle of 11 degrees and speed of 20 m/sec.

7 Conclusion

The simulations show that the control strategy, in theory has some potential. However one significant practical hurdle (at least) needs to be overcome during implementation. Accurate estimation of the state variables, (especially side-slip velocity *V*) on which the control strategy relies, is not a simple task. This will rely on our ability to take outputs from a variety of sensors (GPS, rate-gyroscope, accelerometers) and to filter and extract the state variable estimates.

Plans are currently in progress to undertake an experimental program to develop the yaw stabilising control strategy.

8 Acknowledgements

This work [Stevenson 2004] was carried out by Bradley Stevenson as a final year QUT undergraduate student in 2004, under the supervision of Dr Peter Ridley. Bradley Stevenson is currently employed by Hastings Deering as a Mechanical Design Engineer.

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