SYSTEM DEVELOPMENT FOR HYDRAULIC TILT ACTUATION OF A TILTING NARROW VEHICLE

Benjamin Drew, Kevin Edge, Matt Barker, Jos Darling & Geraint Owen
Centre of Power Transmission and Motion Control
University of Bath
Bath, BA2 7AY, UK
Phone: +44 (0)1225 384048
Email: b.drew@bath.ac.uk

Heiko Johannsen
Technische Universität Berlin
Institut für Land- und Seeverkehr
Fachgebiet Kraftfahrzeuge
TIB 13, Gustav-Meyer-Allee 25
13355 Berlin, Germany

ABSTRACT

Narrow tilting vehicles have been identified as a novel concept for individual urban transport providing car-like levels of comfort, safety and convenience, with the low emissions, noise and road envelopes of motorcycles. This paper discusses the development and simulation of the hydraulic circuit and the necessary control system to actively control the tilting mechanism of a narrow tilting vehicle. A preliminary system was defined based on the specifications of the tilt system, which was then modelled in MatLab-Simulink using a hydraulic blockset. Various severe input demands were used to select and size the components and assess the predicted performance of the complete system. Using this simulation, a final system was established. Real performance will be examined by constructing and implementing this system on a rolling prototype of the vehicle integrated with the active tilt control system.

KEYWORDS: Tilting vehicle, Active control, Tilting system

1 INTRODUCTION

Compact Low Emission Vehicle for Urban Transport (CLEVER) is an EC funded research project [1] to develop a novel concept vehicle for urban transport aiming to overcome the problems of congestion, emissions and inconvenient public transport while satisfying the needs of consumers. One possible concept aims to marry the comfort, safety and convenience of a conventional car with the small, narrow road footprint and efficiency of a motorcycle. In order to be accepted by the consumer, important areas have been identified as safety, efficiency, emissions, and vehicle size. Car-like controls, as opposed to motorcycle controls, will be used.

The main problem associated with narrow vehicles is their tendency to overturn in corners. By tilting the centre of mass towards the centre of a curve, the likelihood of roll over is reduced.

There are two methods of controlling the tilting action of the vehicle: passive and active. Passive tilt control necessitates driver skill, motorcycle controls and counter-steering [2]. Active control, despite its inherent complexity, alleviates the requirement of counter steering and allows direct steering inputs, as is expected when using car-like controls. Active control also allows for a fully enclosed cabin, since the system can maintain the vehicle in an upright position when stationary.

The challenge with such a vehicle is to design an efficient tilting system that behaves appropriately. Important aspects to consider include the increased power demand on the engine, the stability of the control, and the additional weight and cost such a system could introduce. An electric solution was considered for its easy method of energy storage and high efficiency. However the disadvantages of cost, weight and packaging ruled this option out. A hydraulic system was considered a better solution owing to its higher power density, flexible integration, and easier position control. This paper discusses the design, development and simulation of an efficient hydraulic system for this unusual application.

2 SYSTEM DESIGN

2.1 Analysis of system

The CLEVER vehicle prototype has a main tilting structure and a separate rear module that remains upright with respect to the ground. The hydraulic system fits between these two structures providing the actuation necessary to rotate the tilting cabin relative to the stationary base. Assuming the tilt axis is on the ground, the system can be considered as a position control system for an inverted pendulum, with the length of the pendulum being the distance between the tilt axis and the centre of mass. A schematic is shown in figure 1.

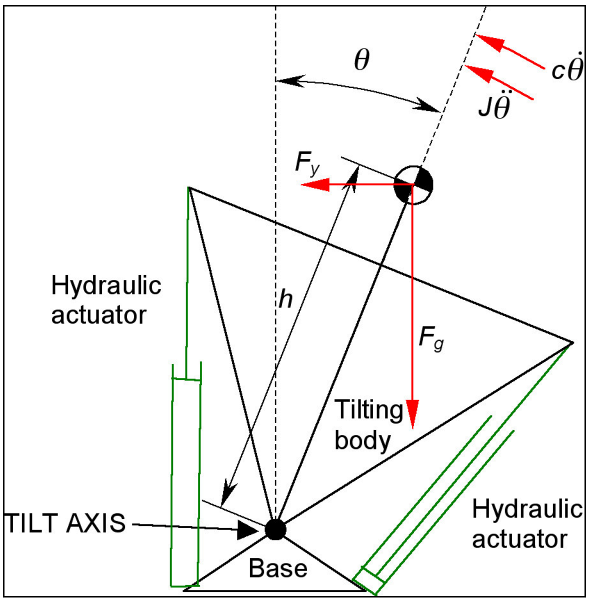


Figure 1: Representation of the system as position control of an inverted pendulum

The total torque the actuators have to provide is the sum of the torque due to gravity on the mass, the torque due to the centripetal reaction force on the mass, the torque required to accelerate the mass and overcome any damping and friction in the system, and the torque due to external forces on the mass. Thus, the torque required to tilt the vehicle can be expressed as:

$$T = -F_g h \sin(\theta) + F_a h \cos(\theta) + J\ddot{\theta} + c\dot{\theta} + T_{dist} \tag{1}$$

where F_y is the centripetal reaction force, F_g is the force due to gravity, h is the distance from the tilting axis to the centre of mass, J is the inertia of the tilting body, c is the damping of the tilting body, and T_{dist} is the torque due to external forces on the mass.

The dynamic requirement is to tilt the body from −45° to +45° in 1.5 seconds (cycle frequency of 0.33 Hz). This is the 'worst case' duty cycle. The tilt angle, θ, as a function of time is therefore taken to be:

$$\theta(t) = \frac{\pi}{4} \sin\left(\frac{2\pi}{3}\theta\right) \tag{2}$$

For initial evaluation, a simple model of a stationary vehicle will be used, assuming no damping in the tilt action. The torque, T, required to tilt the system is:

$$T = -mgh\sin(\theta) + mh^2\ddot{\theta} \tag{3}$$

Power, P, can be calculated from torque:

$$P = T\dot{\theta} \tag{4}$$

With current estimates suggesting m is 295 kg and h is 0.49 m, a plot of equations 2, 3, and 4 is shown in figure 2. The maximum torque requirement is approximately 1.25 kNm, and the maximum power requirement is 1 kW.

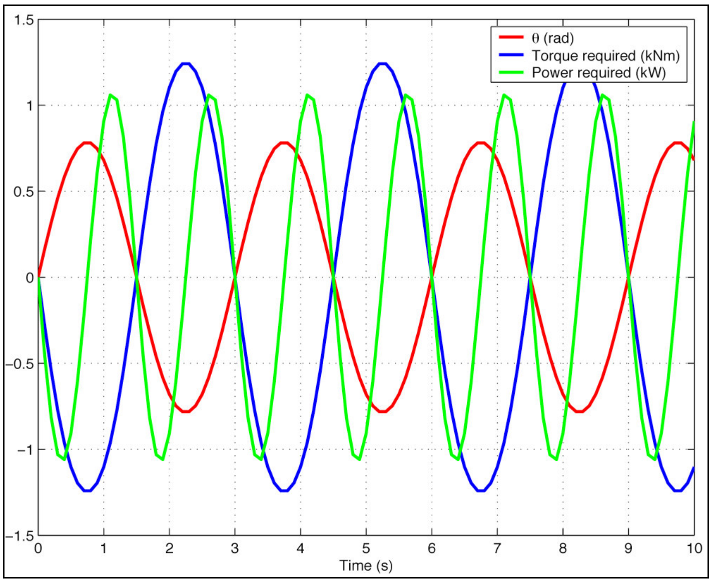


Figure 2: Tilt Angle, Torque, and Power Requirements

2.2 *Hydraulic circuit*

2.2.1 Proposed Design

The hydraulic circuit has been designed to control the position of the tilting part of the vehicle with two single acting linear hydraulic actuators. When pressurised, these cylinders control the lean angle of the tilting cabin by rotating it with respect to the upright rear module. A proportional directional control valve with a closed centre position controls the position of these actuators and locks the cylinders when no command is given.

The system is pressurised by a gear pump driven directly from the engine crankshaft, providing adequate flow for sinusoidal tilting from maximum to minimum tilt angle at 0.33 Hz. In order to unload the pump, augment the flow and provide flow in the event of pump failure, an accumulator is added to the circuit in conjunction with an unloading valve. When the desired system pressure is reached, the unloading valve opens, allowing flow generated by the pump to return to tank, decreasing the torque demand from the engine. When the accumulator has discharged and the pressure in the system falls below a preset value, the unloading valve closes, directing flow from the pump back to the system to charge the accumulator and pressurise the cylinders until maximum system pressure is reached. Figure 3 shows the proposed hydraulic circuit.

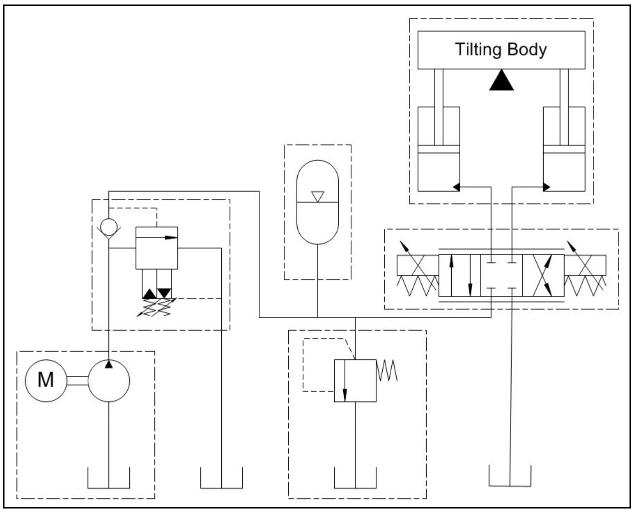


Figure 3: Proposed hydraulic circuit for tilting vehicle

2.2.2 *Actuators*

The hydraulic actuators have been positioned in the vehicle between the tilting body and the upright rear module. They are positioned to optimise the torque generation within the package design constraints. As a consequence, the lever arm varies with respect to tilt angle, as shown in figure 4.

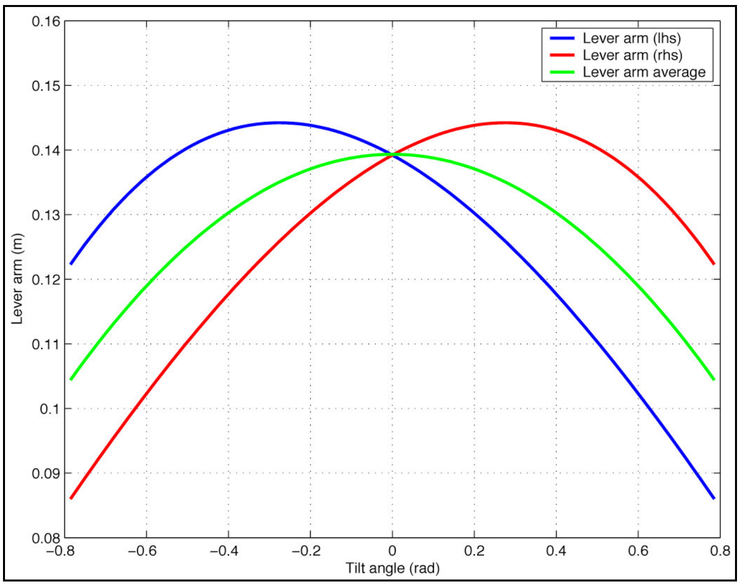


Figure 4: Actuator lever arm against tilt angle

The hydraulic actuators are single acting, being pressurised on the piston side only. When the body tilts to the right to make a right hand turn (θ is positive), the left hand cylinder provides the necessary force and vice versa.

The relationship between actuator extension and tilt angle is shown in figure 5, for the case of the left actuator. The stroke requirement is 200 mm.

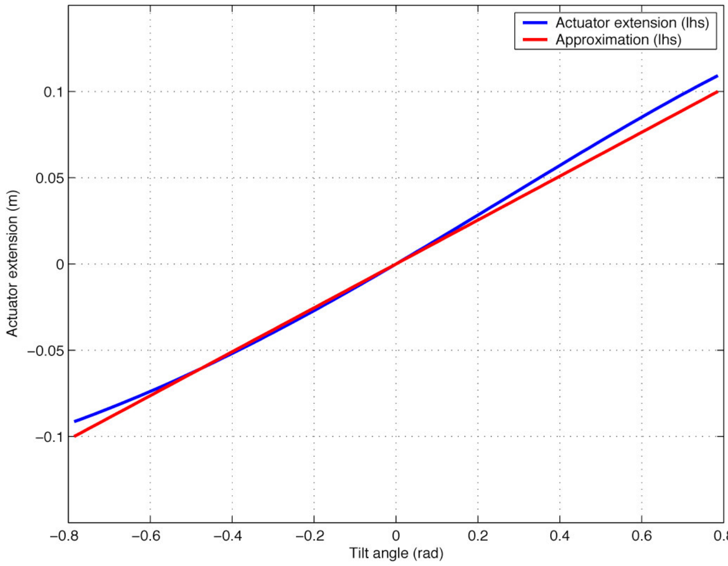


Figure 5: Actuator length against tilt angle

The actuators need to be sized according to the force requirement. The piston diameter can be determined from the torque equation of a pressurised actuator:

$$T = pAb \rightarrow A = \frac{T}{pb} \rightarrow d = 2\sqrt{\frac{T}{\pi pb}} \tag{5}$$

where p is the system pressure, b is the lever arm, A is the piston area, and d is the piston diameter. The highest load on each actuator occurs when fully contracted, so from figure 4, b is 0.122 m. With the maximum torque requirement of 1.25 kNm and a system pressure of 160 bar, the piston diameter needs to be at least 28.5 mm. Commercial cylinders fulfilling this requirement have a piston diameter of 32 mm, and can therefore provide a torque of 1.57 kNm with a system pressure of 160 bar. Approximately 0.16 litres of oil is required for one complete tilt from −45° to +45°.

2.2.3 Control Valve

The control valve for the system is a 4-port, 3-position proportional directional control valve with a closed centre. A proportional valve is necessary to achieve the system's dynamic requirements and continuous regulation during normal driving. The size of the valve can be determined by the maximum flow requirement. Assuming the same 0.33 Hz sinusoidal duty cycle shown in figure 2, the maximum piston speed, and therefore the peak flow requirement can be determined: this is calculated to be 10.1 l/min.

2.2.4 Pump

The pump choice is determined by considering the flow requirement and the IC engine used to drive the pump. Analysis of the CVT transmission [3] suggests that when cruising, the engine will spend much of the operating time at around 5500 rpm. Engine idle speed is 1700 rpm, and maximum engine speed is limited to 8500 rpm.

The chosen pump would have to operate and function between the two extremes, while providing mean flow at the normal operating point. In order for a gear pump to operate at these engine speeds, the pump will need to be driven through a gear ratio to operate within the limiting speeds of the pump.

The 0.33 Hz sinusoidal duty cycle can be used to determine flow requirements. Mean flow required for this duty cycle is 6.434 l/min. Commercial gear pumps operate between 750 rpm and 4000 rpm, so choosing a 3 cc/rev gear pump, matching that to 6.434 l/min, results in a pump speed of 2150 rpm. In order to match this pump speed to an engine speed of 5500 rpm, the necessary gear ratio between the pump and the engine is 2.56:1.

2.2.5 Accumulator

The accumulator is used within the system for three purposes:

1. To provide the necessary flow during periods when the pump is unloaded.
2. To provide additional flow when the pump is loaded and harsh manoeuvres are being undertaken by the vehicle.
3. To provide emergency flow if the pump or engine fails.

Diaphragm accumulators were considered, owing to their lighter weight and reduced size compared to bladder accumulators. The higher pressure and flow capabilities of the latter were not necessary for this application.

Sizing the accumulator not only involves examining the simulation results, but also has to take into account design constraints. Larger accumulators would be able to provide flow for more tilt cycles, allowing the pump to be unloaded for longer periods of time. This benefit has to be balanced with the longer charging time required while the pump is loaded, charging the system, and integrating the increased weight and size of a larger accumulator within the vehicle design. A smaller accumulator reduces the number of tilting movements available while the pump is unloaded and increases the frequency of charging cycles, arguably impairing the drivability. However, smaller accumulators would be quicker to recharge, and integration within the vehicle would be easier owing to their lighter weight and smaller size.

2.2.6 Unloading valve

The engine used in the CLEVER vehicle is a small 213 cc 4-stroke internal combustion engine running on compressed natural gas with a maximum power of 12.5 kW. For this reason, it is beneficial to unload the pump when the system is pressurised, and hence reduce the power consumption from the engine. To accomplish this, an unloading valve needs to be integrated into the circuit.

When the system starts, the valve is initially closed, sending flow from the pump through a check valve, charging the system. When the maximum system pressure is reached, the valve opens allowing flow from the pump to return to tank, unloading the pump and reducing the torque load on the engine. As the vehicle tilts, the pressure within the system is reduced until it reaches a threshold pressure, at which point the valve closes again, enabling the pump to recharge the system.

One challenge with this situation is the transition between loading and unloading the system and the effect of this on the drivability of the vehicle. With a standard two-position valve, the loading can be quite harsh, but by using a proportional or soft-start [4] valve, it would be possible to smooth the transition when loading the pump. When unloading, there is little point in gradually opening the valve, since the throttling that occurs will waste energy. A proportional valve would also require careful control, including pressure measurement to predict when the system will need charging, and hence start to close the valve. A proportional valve was nevertheless tested in simulation (see section 3.3.5).

3 SIMULATION

3.1 MatLab-Simulink Environment

A model representing the hydraulic system of CLEVER has been developed in MatLab-Simulink using a third party hydraulic blockset [5]. The organisation of the blocks follows the organisation of a real hydraulic system. A block diagram of the model is shown in figure 6.

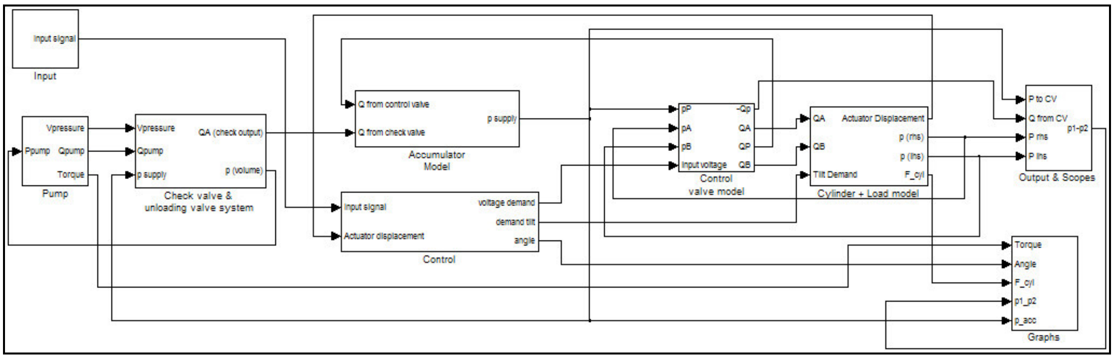


Figure 6: Hydraulic Simulation Block Diagram

3.2 Assumptions/Simplifications

The following assumptions and simplifications were made in order to implement a representative hydraulic system in the model.

- The two single acting actuators have been represented as one double acting double rod cylinder with the annulus area the same as the single acting piston area and a rod diameter of zero.
- As can be seen from figure 5 and discussed below, a straight line can approximate the relationship between actuator extension and tilt angle.
- A specific unloading valve is not available in the model, therefore it has been represented as two-port, two-position servo valve with a relay used to open and close the valve based on system pressure.
- For most of the simulations, the unloading valve is modelled as a bang-bang valve.
- The engine is assumed to be running at 5500 rpm throughout the simulations.
- The numerical data used in the simulation is listed in table 1 below.

Item	Symbol	Value	Units
Mass of tilting body	m	295	kg
Distance from tilting axis to CoM	h	0.49	m
Diameter of actuators	d	32	mm
Stroke of cylinders	l	0.2	m
Accumulator pre-charge pressure	–	80	bar
Lean angle limits	θ	+45 to −45	degrees
Oil compressibility	β	10000	bar

Table 1: Numerical data used in the hydraulic simulation model

3.2.1 Cylinder force calculation

Since the simulation of the hydraulic system reported here is in isolation of a vehicle model, the force the hydraulic system has to overcome needs to be calculated from the input demand signal and the system outputs. Recalling equation (1), the torque required to tilt the vehicle is

$$T = -F_g h \sin(\theta) + F_a h \cos(\theta) + J\ddot{\theta} + c\dot{\theta} + T_{dist}$$

where $J = (mh^2 + I)$. T_{dist} is initially taken to be zero. The torque applied by each hydraulic actuator in the system is:

$$T = F_a b \tag{6}$$

where b is the lever arm and F_a is the force provided by each hydraulic actuator. From the geometry of the tilt mechanism, the lever arms for the actuators vary with respect to tilt angle, as shown in figure 4.

When the tilting body is tilted to the right, the left hand cylinder provides the necessary force and vice versa. While approximations of each curve could be used, the average of the two curves adequately represents the real lever arm. This average curve, as shown in figure 4, can be represented by the following quadratic equation:

$$b = -0.0566\theta^2 + 0.1393 \tag{7}$$

When the vehicle is tilted in the balanced position, the only forces governing θ are the lateral centripetal reaction force, F_y, and the gravitational force, F_g; both $\dot{\theta}$ and $\ddot{\theta}$ are zero. Since the aim of the tilting system is to balance the vehicle, the tilt angle in this situation can be considered as the demand angle, θ_d. So, from figure 1:

$$\tan\theta_d = \frac{F_y}{F_g} \tag{8}$$

This assumes Ackermann steering on a flat road [6] with no tyre, suspension, or road camber effects included, and the tilt axis at ground level. In this situation, the centripetal reaction force is dependent on a specific steer input at a certain vehicle speed. Because these two values are used to calculate the demand tilt angle, centripetal reaction force is a function of tilt demand:

$$F_y = mg \tan\theta_d \tag{9}$$

Because one of the main outputs from the hydraulic model is actuator extension, it is necessary to have an equation relating actuator extension to tilt angle to include in the cylinder force calculation. The relationship between actuator extension and tilt angle can be approximated as a straight line:

$$x = 0.1273\theta \rightarrow \theta = 7.85x \tag{10}$$

where x is the actuator extension; $x = 0$ when $\theta = 0$, and $x = 100$ mm when $\theta = 0.785$ radians (45°).

The combination of equations 1, 6, 7, 9 and 10 results in equation 11. Each actuator has to provide this force:

$$F_a = \frac{mgh \tan\theta_d \cos(7.85x) - mgh \sin(7.85x) + 7.85mh^2\ddot{x} + 7.85c\dot{x} + T_{dist}}{-3.488x^2 + 0.1393} \tag{11}$$

3.3 Results

3.3.1 Harsh Ramp Input

This situation is unlikely to occur in normal driving. It is used to test the response of the system to sudden harsh inputs, especially the central ramp that is a tilt requirement of −45 to +45 degrees in 1 second (this is beyond the performance required by the system). The simulation results of demand and real tilt angle, torque load on the IC engine crankshaft, force in the cylinders, and system pressure are shown in figure 7.

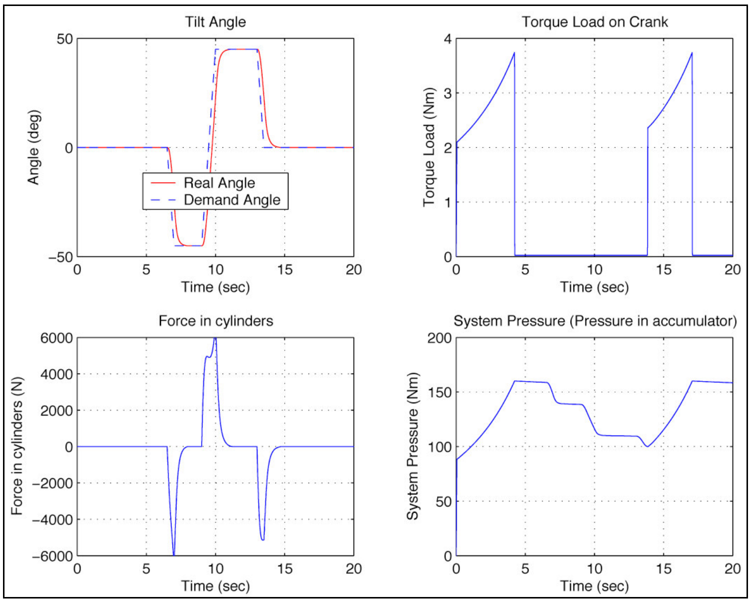


Figure 7: Simulation results for harsh ramp input

These results show that the response of the hydraulic system closely matches the demand signal. During the ramp inputs, the response lags the demand by approximately 0.25 seconds. After the initial startup—pressurising the system by charging the accumulator—the pump unloads and the manoeuvre is almost completed using only the flow from the accumulator. As can be seen, when the pump is unloaded, it is taking almost no torque from the engine. When the pump is loaded, the maximum torque requirement is approximately 3.7 Nm. With the engine speed running at 5500 rpm, this torque requirement can be accommodated and a charging cycle takes just over 3 seconds. The maximum power required at this engine speed is approximately 2 kW.

3.3.2 Slalom

This input signal represents a slalom manoeuvre from the straight-ahead position. The tilting frequency is 0.25 Hz (i.e. full tilt in 2 seconds). It would be possible to replicate this manoeuvre in the prototype vehicle. The simulation results are shown in figure 8.

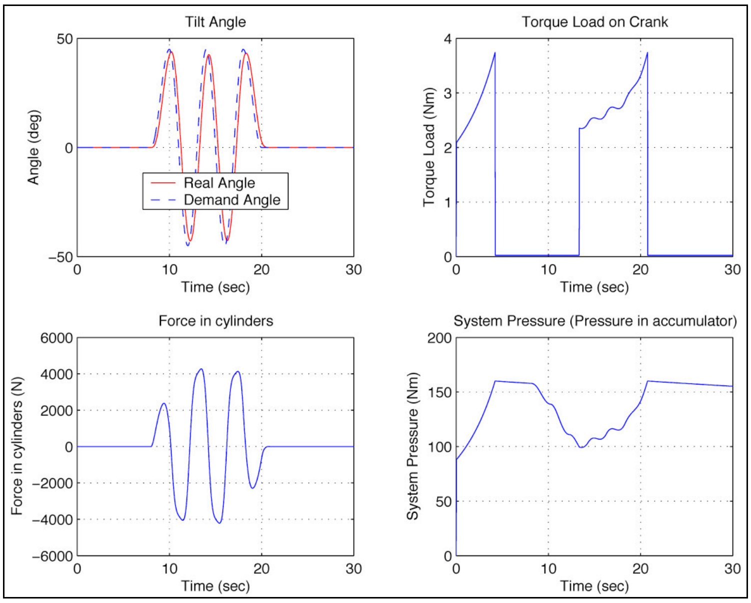


Figure 8: Simulation results for slalom (0.25 Hz)

Again, the system shows acceptable behaviour, with the accumulator providing the necessary flow for approximately half of the manoeuvre.

3.3.3 Harsh Slalom

This input signal represents the worst case possible with the CLEVER vehicle. The tilting frequency is 0.33 Hz (i.e. full tilt in 1.5 seconds). The simulation results are shown in figure 9.

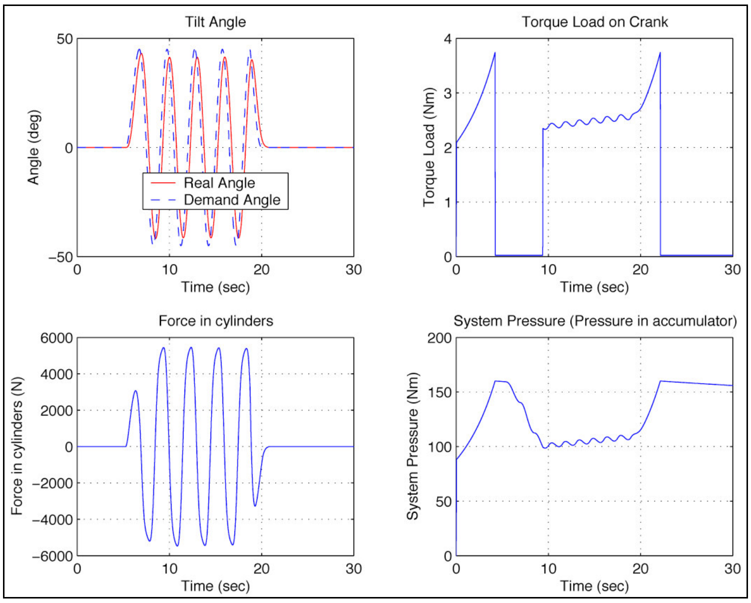


Figure 9: Simulation results for harsh slalom (0.33 Hz)

As can be seen, when the pump is charging the system, it can provide the mean flow, with the accumulator supplementing the flow to provide the peak requirements. The system cannot recharge until this harsh manoeuvre is completed, as signified in the rise in system pressure at 20 seconds.

3.3.4 Disturbance Force

The hydraulic tilt system has to take into account disturbances on the tilting body such as wind forces. To test this, a side wind force can be added to the forces applied in the cylinder associated with the tilting force. Side wind force, F_w, can be defined as:

$$F_w = \frac{1}{2}\rho C_d A V^2 \qquad (12)$$

where ρ is the air density (taken as 1.226 kg/m^3), C_d is the drag coefficient, A is the projected area facing the wind, and V is the velocity of the wind. Taking, for example, a wind speed of 100 kph, a C_d value of 1.3, and an area estimation of 2.5 m^2, F_w is 1537 N. The C_d value used in this calculation is particularly pessimistic. The correct value for the side of the CLEVER vehicle is unknown, so a value of 1.3 is taken as worst case. See table 2 for typical C_d values for certain objects.

Taking a wind force of 1600 N, this translates to a disturbance torque around the tilt axis of 784 Nm. This is introduced as a step input (simulating a gust of wind) between 7 and 11 seconds during the harsh slalom tilting manoeuvre (see section 3.3.3). The results for the simulation are shown in figure 10.

	Drag coefficient, C_d
Conventional cars (frontal)	0.3–0.4
Rough sphere (Re = 10^6)	0.4
Square flat plate 90° to flow	1.17
Articulated lorry	0.6–0.9
Wire cables 90° to flow	1.0–1.3

Table 2: Typical drag coefficients [7]

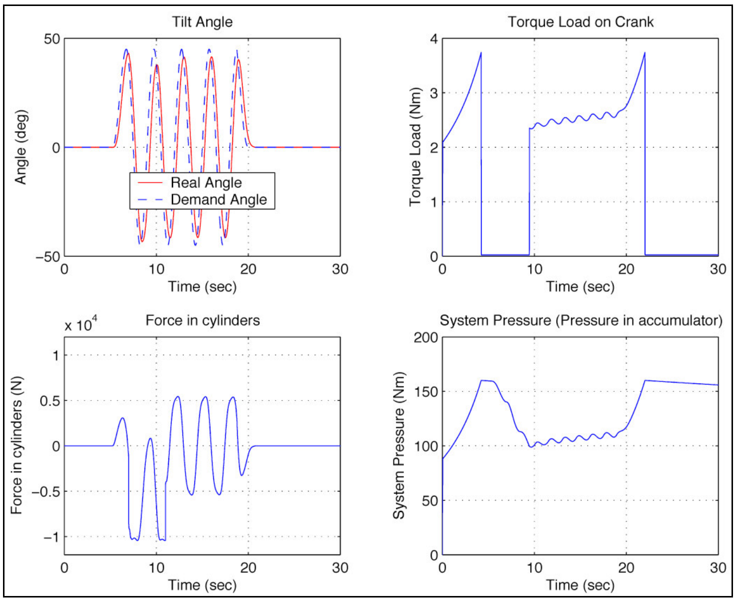
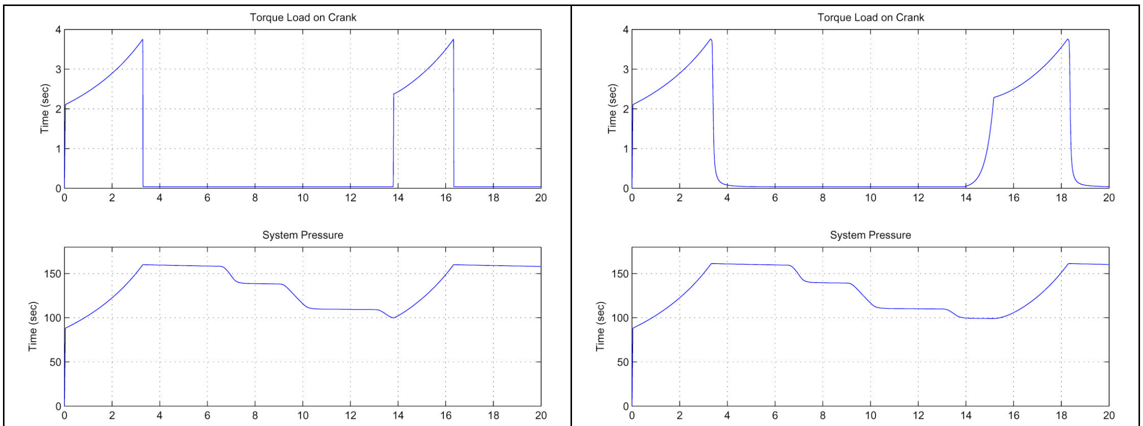


Figure 10: Simulation results for harsh slalom with side force introduced

Looking at the plot showing the force in the cylinders, the effect of the side force can be clearly seen. However, comparing the tilt angle plot with the previous result with no disturbance (figure 9), there is little difference in behaviour. Closer analysis of the results shows that the amplitude of the tilting motion is reduced by less than 3.5° with this force. At wind speeds above 130 kph, this reduction would be greater; however it is likely that the CLEVER vehicle would be blown over, irrespective of the tilt angle.

3.3.5 Unloading valve

As can be seen in previous results, the torque loading on the engine is quite sudden, and the effect on the drivability of the vehicle is currently unknown. In an effort to smooth the transition between the loaded and unloaded state, a 2-port, 2-position electronically controlled proportional valve was substituted for the unloading valve in the model. By implementing a first order lag between the relay switch and the valve control voltage (this can be achieved in the control by the use of a simple RC circuit, ignoring the inductance of the solenoid), the control signal is 'smoothed', gradually closing the valve, which gradually loads the engine. Although shown in simulation, repeating this when unloading the engine is not beneficial, since energy would be wasted throttling the flow back to tank. Figure 11 below illustrates the difference between a standard valve (as used in previous simulations) and the first order lag controlled proportional valve.



$\tau = 0$ seconds

$\tau = 0.6$ seconds

Figure 11: The effect of adding a first order lag in the unloading valve control

Although this method has the effect of smoothing the transition, the actual loading time is extended with the proportional valve. It must also be noted that the time constant, τ, of the first order lag requires careful tuning, since if it is too great, the system pressure would drop too low, affecting the overall response of the tilt system.

4 CONTROL

4.1 Control Methods

Driving a tilting vehicle employing passive tilt control necessitates driver skill and motorcycle controls. CLEVER must have car-like controls and be fully enclosed, meaning an active control system must be employed to maintain vehicle stability. There are three main methods of tilt control:

- Direct tilt control (DTC): the vehicle is controlled using actuators to tilt one portion of the body relative to another. Advantages are low speed stability and simple implementation, but a high roll moment is required at the initiation of the corner.
- Steer Tilt Control (STC): the vehicle counter-steers automatically in a similar manner to motorcycle control. Advantages are good high speed stability, but a

costly steer-by-wire steering system has to be implemented. At lower speed, such a system requires high gains to maintain stability.

- Dual mode: this method aims to marry the two methods above. Previous studies [8, 9] have generally implemented DTC at low speed and STC at high speed.

For CLEVER, a DTC system has been chosen as it offers the best combination of achieving engineering practicality and project objectives.

4.2 Basic Control Strategy

The controller objective is to balance the centripetal reaction force and gravitation force to prevent roll over.

The first stage of the control algorithm is to use open loop control based on the steer angle and velocity. This gives a first estimate at the demand tilt angle, but is based on Ackermann geometry, so it is not an exact demand. Ackermann geometry is not a true representation because tyre slip angles are not considered, and the real geometry of CLEVER is non-linear.

Closed loop control develops upon this first stage by estimating the demand angle using the steer angle and velocity, then refining this demand angle by using the perceived lateral acceleration as a feedback at high speed, and actuator position at low speed. The yaw rate may also be used as additional feedback to check for errors. The control functions as follows:

1. The initial lean angle demand is from the open loop controller, but this will be improved to account for non-linearities (vehicle geometry, tyre model). This initial demand signal will be calculated using look up tables.
2. The perceived lateral acceleration and actuator position provides closed loop correction.
3. Yaw rate is used to check correct functioning of feedback.

Sensors on the CLEVER vehicle will measure the following: velocity, steer angle, lean angle (actuator length), yaw rate, and perceived lateral acceleration. This control strategy will be constructed and evaluated in simulation using a vehicle model and the hydraulic model presented here before integration with the prototype vehicle.

5 CONCLUSION

An enclosed narrow tilting vehicle with car-like controls requires an active tilting system to maintain stability in corners. A hydraulic system has been identified as the best solution for actuation. A preliminary hydraulic system was designed and a model of the system was constructed in MatLab-Simulink to assess the design concept, and select and size components. This model aimed to simulate the behaviour of the hydraulic system in the prototype vehicle. A series of input demands were used and the resulting behaviour was assessed, looking specifically at the system response, the torque load on the engine, the force in the cylinders, and the resulting system pressure. Acceptable performance was achieved for all input demands using the final component specifications.

A control strategy has been designed and, after model evaluations, the active control system will be integrated on the prototype vehicle to provide experimental results of the complete active tilt control system.

5.1 Further Work

Control model development is the focus of current work, with attention being placed on system behaviour in specific scenarios. These include startup and idle conditions, different driving modes (coasting, cruising), and parking, to ensure appropriate function and acceptable performance. The hydraulic system control will be integrated with the engine ECU in order to ensure that the hydraulic system can only be charged when the engine can provide the necessary torque. This control logic needs careful consideration.

Following this, the work will concentrate on integration of the hydraulics and control system within the prototype vehicle and experimental evaluations can begin to assess real world performance, tune the control system, and ensure suitable behaviour of the hydraulic system for all possible scenarios.

6 ACKNOWLEDGEMENTS

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