STABILITY CONTROL OF ARTICULATED VEHICLES USING BRAKE-BASED TORQUE VECTORING

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ABSTRACT

The stability of articulated vehicles is a growing concern due to the fact that transport is moving towards larger and longer truck and trailer combinations for better efficiency which may possibly lead to an increased number of safety risks. This study focuses on the yaw stability of articulated vehicles which include jack-knifing and snaking. The main steps that were taken in this study consist of developing a bicycle model and a simulation model built using MSC ADAMS. The simulation model is fully validated with experimental handling tests using a loaded trailer. This simulation model works with co-simulation through SIMULINK, and it is through this co-simulation that a simple controller is implemented. This controller uses yaw moment control that is achieved via brake-based torque vectoring and simulates the differential braking of the trailer. This study shows that a simple yaw moment control applied to the trailer can significantly improve the stability hence increasing the safety of articulated vehicles and reducing fatalities. This study highlights the possibilities this area of research has with great promise for future results.

Keywords: Articulated vehicles, Stability, Yaw moment control, Torque vectoring.

NOMENCLATURE:

Symbols		Greek		Subscripts	
F	Force, [N]	δ	Steering angle, [rad]	1	SUV parameters
v	Velocity, [m/s]	α	Slip angle, [rad]	2	Trailer parameters
a	Acceleration, [m/s^2]	ψ	Yaw angle, [rad]	f	Front
I	Moment of inertia, [kgm^2]	θ	Hitch angle, [rad]	r	Rear
Y	Hitch force, [N]	$\dot{\psi}$	Yaw rate, [rad/s]	H	Hitch
M	Moment, [Nm]	$\dot{\theta}$	Hitch rate, [deg/s]	t	Trailer
a,b,c,e,l	Length, [m]	$\ddot{\psi}$	Yaw acceleration, [rad/s^2]	x	Longitudinal plane
C	Cornering Stiffness, [N/rad]	$\ddot{\theta}$	Hitch acceleration, [rad/s^2]	y	Lateral Plane
				z	Vertical Plane

1. INTRODUCTION

An articulated vehicle is described as a vehicle that has a pivoting joint, which is either permanent or semi-permanent, connecting two or more parts of the vehicle (Azad, 2006). The stability of these types of vehicles have become more and more important over the recent years. Since road freight transport is continually growing and industry seeks to improve productivity and efficiency, articulated vehicles are getting longer as well as larger with more articulations. This can potentially lead to an increase in safety risks. According to the state of Road Safety Report for January-March 2018, 3.6% of major crashes in

South Africa were due to jack-knifing (Road Trafffic Management Corporation, 2018). It was found that in the last 20 years, over 400 fatalities have occurred every year in road accidents that involved trailers being towed by passenger vehicles (Koenigsberg, 2008). In a 12-month survey that was completed in the UK (Farr & Neilson, 1968), it was found that jack-knifing occurs in over half of handling incidents and snaking plays the second largest cause. These statistics clearly show that stability problems are one of the major hazards regarding articulated vehicles and that providing a method of stabilisation will significantly reduce articulated vehicle accidents. The majority of other handling-related accidents that are not due to jack-knifing or snaking occur due to the inability to negotiate corners, most likely due to excessive speed but also because of high loads leading to a higher center of gravity. The risk of suffering injuries or fatalities are also ten times higher for other road users than that of the driver of the articulated vehicle (Farr & Neilson, 1968). This highlights the severity of an articulated vehicle accidents as more people are being placed in harm's way. Since there is a connection between the two components of an articulated vehicle, the dynamics and kinematics of the trailer and towing vehicle, are coupled. A Car-Trailer Combination (CTC) has a dynamic critical speed which is used to determine whether the system is stable or not. A single vehicle also has a critical speed, but it is not a concern since, at very high velocities, the stability of the system remains intact (Zhang, 2015). This proves how much more complicated an articulated vehicle is in terms of vehicle dynamics.

Stabilising an articulated vehicle using a suitable control system has been explored. (Zanchetta et al., 2018) proposed a torque vectoring formulation that made used of the combined hitch angle and yaw rate of an articulated vehicle. The stabilization was achieved by applying a yaw moment on the towing vehicle. The control system was based on a single-input single-output (SISO) feedback control structure where the yaw rate of the towing vehicle is altered when instability is detected using a hitch angle sensor. It was found that this controller was insufficient in stabilising the vehicle under extreme conditions but provides safe trailer behaviour during the comprehensive set of manoeuvres used in the study. (Azad, 2006) proposed a classical PID controller for an active steering system on the towing vehicle. The controller worked by measuring the hitch angle, comparing it to the desired hitch angle and then altering the valve displacement of the steering system. (Azad, 2006) also proposed a robust feedback controller for a torque vectoring system. In which the controller is a full state feedback system that is used to find the required torque to stabilise the vehicle. The control is generated by applying an equal but opposite torque to the rear wheels of the towing vehicle. (O'Neal Arant, 2013) proposed a controller that uses Model Predictive Control (MPC). The MPC controller optimizes the control of the vehicle by predicting the vehicle response a finite time into the future. The output of the controller is a yaw moment applied to the towing vehicle which is obtained by using differential braking forces. The controller made use of the anti-lock brake system (ABS) to achieve differential braking. It was found that the MPC approach was able to predict future stability risk quite elegantly. (Mokhiamar, 2015) proposed a control design concept, that uses sliding control law, for an optimum distribution of longitudinal and lateral forces of the four tyres of a towing vehicle.

The majority of these concepts focus on the control of the towing vehicle. The concept of applying control to the trailer has been explored before in systems such as anti-jackknife systems, modern ABS and roll stability. Modern trailers are also equipped with ABS controllers and it is possible that stability could be achieved by braking the trailer instead of the towing trailer. This study shall aim to introduce yaw moment control that is achieved via brake-based torque vectoring and simulates differential braking of the trailer. The controller shall be designed and tested using an MSC ADAMS model of a Land Rover Defender 110 Tdi. and testing trailer on a simulation basis.

2. MATHEMATICAL MODEL

The mathematical model that was derived for this study is a Single-Track Model (STM) that only takes the yaw dynamics of the system into account. The main purpose of this model is to be used as the reference model for the controller. The model takes the steering angle and speed of the towing vehicle as the input. The schematic of an articulated vehicle in the yaw-plane is portrayed in Figure 1, showing both the free body diagram and kinetic diagram. The assumptions made to generate this model are highlighted below.

2.1 Assumptions

- Assume the effects of aerodynamics are negligible.
- Assume the effects of deceleration on the lateral dynamics are negligible.
- Assume pitch and roll motion effects are small.
- Assume that tyres are linear.
- Left and right tyres can be approximated to single equivalent tyre at the centre of the axle.
- Assume small angles for the steering angle, therefore $\sin\delta \approx \delta$ and $\cos\delta \approx 1$, slip angle and articulation angle.
- Assume constant longitudinal velocity where the velocity of the towing vehicle v_{x1} and the trailer v_{x2} are equal therefore, $v_{x1} = v_{x2} = v_x$.

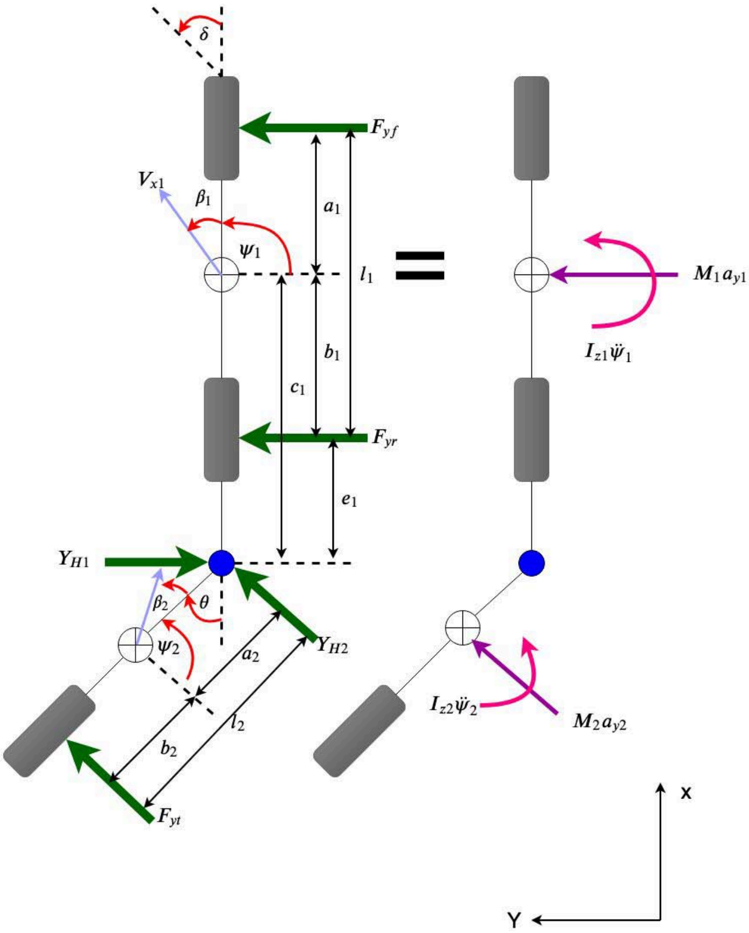

Figure 1: Free body diagram for a single axle trailer for an articulated vehicle

2.2 Equations of Motion

The equations defining the yaw motions for the towing vehicle and trailer are represented in Equations 1 and 2 respectively and the lateral equations of motion for the towing vehicle and trailer are defined in Equations 3 and 4. All moments are taken about the center of gravity (CG).

$$I_{z1}\ddot{\psi}_1 = F_{yf}a_1 - F_{yr}b_1 + Y_{H1}c_1 \tag{1}$$

$$I_{z2}\ddot{\psi}_2 = Y_{H2}a_2 - F_{yt}b_2 \tag{2}$$

$$m_1 a_{y1} = F_{yf} + F_{yr} - Y_{H1} \tag{3}$$

$$m_2 a_{y2} = F_{yt} + Y_{H2} \tag{4}$$

2.3 Kinematic Relationships

With the combination of the towing vehicle and the single axle trailer it was found that certain kinematic relationships hold. These relationships are defined in Equations 5, 6 and 7.

$$\dot{\psi}_2 = \dot{\psi}_1 + \dot{\theta} \tag{5}$$

$$a_{y1} = \dot{v_{y1}} + v_x\dot{\psi}_1 \tag{6}$$

$$a_{y2} = \dot{v_{y1}} + v_x\dot{\psi}_1 - c_1\ddot{\psi}_1 - a_2\left(\ddot{\psi}_1 + \ddot{\theta}\right) \tag{7}$$

2.4 Lateral Tyre Forces

The linearized tyre forces for the front and rear tyres of the towing vehicle as well as the tyre force for the trailer tyres are defined in Equations 8, 9 and 10.

$$F_{yf} = -C_{yf}\alpha_f = -Cyf\left(\frac{v_{y1}+a_1\dot{\psi}_1}{v_x} - \delta\right) \tag{8}$$

$$F_{yr} = -C_{yr}\alpha_r = -Cyr\left(\frac{v_{y1}+b_1\dot{\psi}_1}{v_x}\right) \tag{9}$$

$$F_{yt} = -C_{yt}\alpha_t = -Cyt\left(\frac{v_{y1}-(c_1+l_2)\dot{\psi}_1-l_2\theta}{v_x} - \theta\right) \tag{10}$$

2.5 Linear System of Equations

Equations 1 to 4 were then combined to form a linear set of equations represented by Equation 11. The equations were combined using the assumption that the hitch force at the vehicle is equal to the hitch force at the trailer, hence $Y_{H1} = Y_{H2} = Y_H$.

$$M\dot{x} = Dx + E \tag{11}$$

The matrix parameters can be found in Appendix A. The state vector **x** contains the lateral velocity of the towing vehicle, the yaw rate of the towing vehicle, the hitch rate and the hitch angle as defined in Equation 12.

$$\left[v_{y1}\ \dot{\psi}_1\ \dot{\theta}\ \theta\right]^T \tag{12}$$

This model was set up in MATLAB for two different loading conditions: unloaded and fully loaded. The differential equations are solved using an ODE solver. The tyre cornering stiffnesses were found for each static loading condition using the Pacjeka tyre model. This model proved to have a good correlation with the simulation model that was constructed for this study. It can therefore be stated that this model is a realistic representation of the yaw dynamics of an articulated vehicle.

3. MODEL CONSTRUCTION AND VALIDATION

3.1 Model Construction

The articulated vehicle model was built by combining two already existing validated models, that of the SUV and the testing trailer. The test trailer model was built by (van der Merwe, 2018). The experimental setup showing the SUV and the fully loaded test trailer is depicted in Figure 2. The towbar of the SUV was built and joined to the trailer using a spherical joint. The co-simulation with SIMULINK is used to control the steering path as well as the suspension forces within the vehicle and the trailer. The displacements and velocities at the attachment points are read from the ADAMS model into SIMULINK which then calculates the required suspension forces that are then sent back to ADAMS. The suspension and damping of the SUV is more complicated since a controllable suspension has been implemented on it. This suspension is known as the 4 State Semi-active Suspension System (4S4) (Els, 2006). This system enables switching between low and high damping as well as between soft and stiff springs. The final model is portrayed in Figure 3.

Figure 2: Experimental setup with a fully loaded trailer

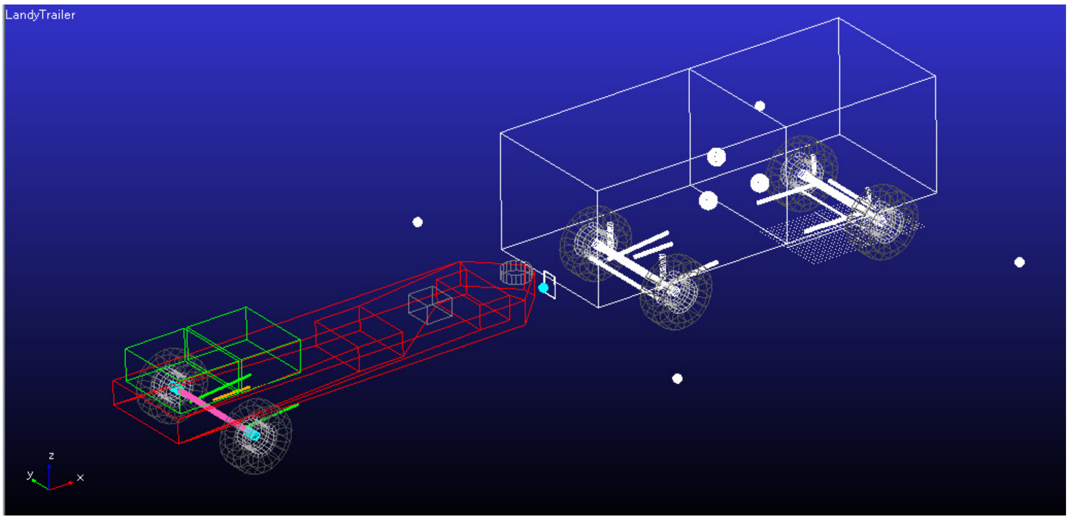

Figure 3: Final ADAMS model of the articulated vehicle

Two types of tyre models were used in the ADAMS model; these include a Pacejka tyre model and a Ftire model. The Pacejka tyre model is a non-linear tyre model that models the contact patch as a single point load. The Ftire model is a non-linear model of the Michelin LTX A/T2 235/85R16 SUV tyre. The Ftire model is a far more accurate representation of a realistic tyre since it takes tyre parametrization data such as the footprint, hardness, vertical stiffness as well as lateral, longitudinal and torsional stiffness into account and it can handle intricate geometry.

3.2 Model Validation

The model described above has been fully validated using experimental tests done on a loaded trailer. The tests that were performed include a Double Lane Change (DLC) manoeuvre and a constant radius. The DLC manoeuvre shall be used as the focus for this paper due to the fact that the emphasis has been placed on yaw dynamics. The double lane change was performed at 40 km/h, 50 km/h and 55 km/h on a soft and hard suspension setting and was setup according to the ISO 3888-1 standard. The layout of a double lane change can be seen in Figure 4 where the lengths are defined according to the same standard (International Standard, 1999). The dimensions of a DLC can be seen in Table 1.

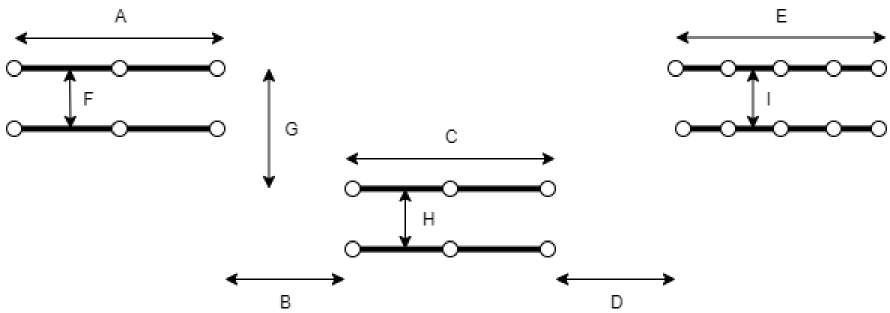

Figure 4: Basic schematic of a double lane change (International Standard, 1999)

The results that were selected for this paper include the hitch angle and trailer yaw rate for the 55 km/h tests with the vehicle on soft and hard suspension settings. These parameters were chosen as they are the parameters that highlight the relationship between the tow vehicle and trailer that make up the articulated vehicle. The results for a soft suspension can be seen in Figure 5 and Figure 6 respectively. The results for a hard suspension are depicted in Figures 7 and 8.

Table 1: Double lane change dimensions

Parameter	Dimension [m]
A	15
B	30
V	25
D	25
E	30
F	1.1 x vehicle width
G	1.2 x vehicle width
H	1.3 x vehicle width
I	1.3 x vehicle width

From Figures 5, 6, 7 and 8, the simulation results and the experimental tests produced a good correlation, and the overall results are very promising. It can therefore be stated that the simulation model is an accurate representation for a loaded trailer for both the hard and soft suspension at low speeds. It can also be stated that the model is accurate enough for the development of the controller.

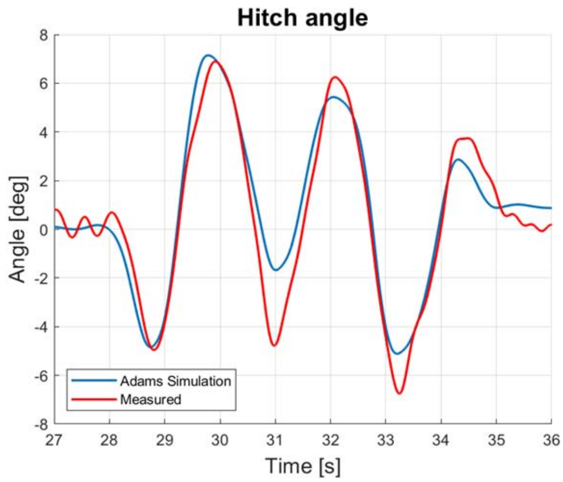

Figure 5: Hitch angle through a DLC at 55 km/h for a soft suspension

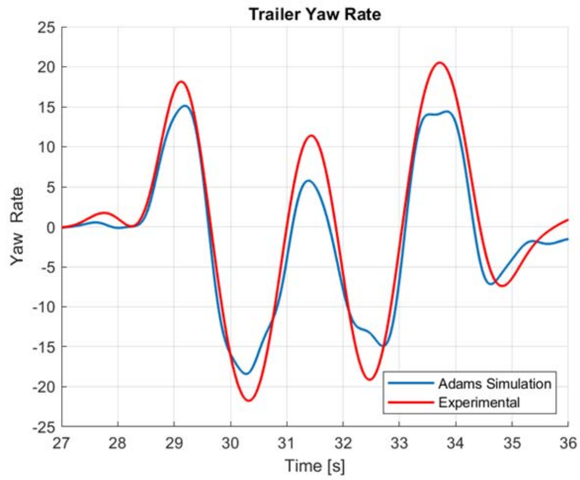

Figure 6: Trailer yaw rate through a DLC at 55 km/h for a soft suspension

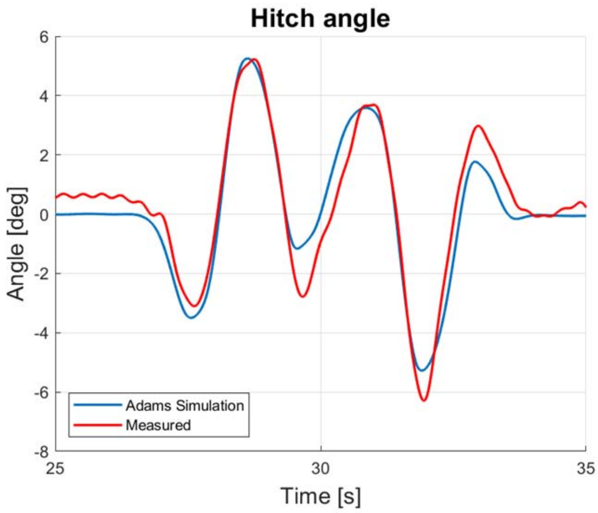

Figure 7*:* Hitch angle through a DLC at 55 km/h for a hard suspension

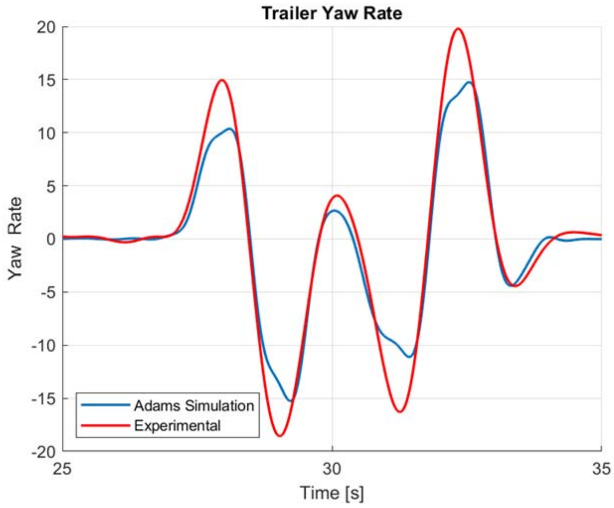

Figure 8: Trailer yaw rate through a DLC at 55 km/h for a hard suspension

4. YAW MOMENT CONTROL

This section serves to prove that the developed ADAMS model can be controlled in some way to improve the handling of the articulated vehicle. The focus is in the yaw plane to attempt to decrease the yaw rate of both the vehicle and trailer as well as to decrease the hitch angle. These parameters are controlled by implementing a yaw moment control on the trailer. The controller is kept as simple as possible since it is only necessary to be used as a proof of concept and therefore, only a gain controller is implemented. The controller was implemented onto the ADAMS model through co-simulation with SIMULINK. The controller uses the Single-Track Model as a reference model and works by reading the current yaw rate on the trailer from ADAMS and taking the trailer yaw rate from the Single-Track Model to determine the error between the two, seen in Equation 13. This error is then multiplied by some gain K to produce the control output defined by Equation 14. The gain was selected using an iterative process and was selected to be 1000. This control output u is then sent back to the ADAMS model as a reverse control torque that acts at the trailer centre of gravity. The braking is simulated this way as this controller served as a proof of concept that the developed model can be controlled using trailer braking. It is therefore kept as simple as possible. This control torque is representative of a braking system on the trailer. This control torque is what is applied to the trailer in order to counteract the snaking of the system, thus stabilising the vehicle. The controller schematic is depicted in Figure 9.

$$e(t) = \dot{\omega_{ref,2}} - \dot{\omega}_2 \tag{13}$$

$$u(t) = Ke(t) \tag{14}$$

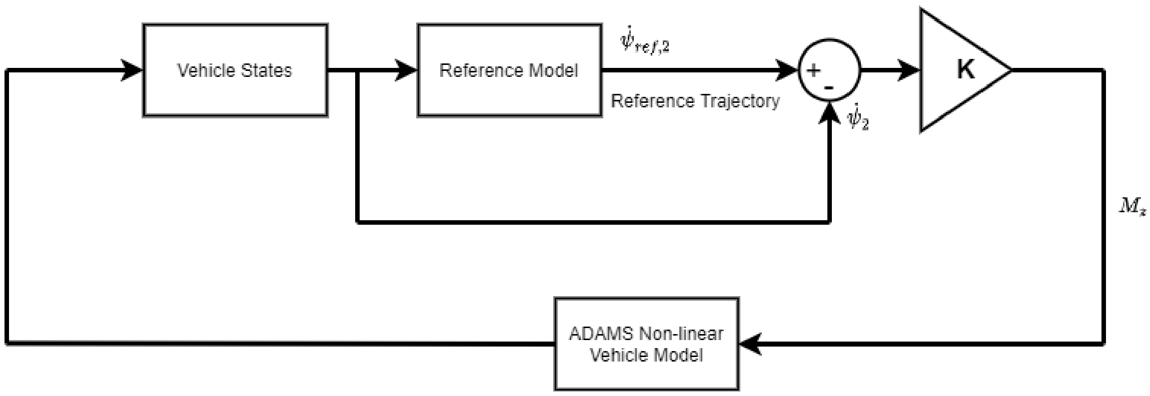

Figure 9: Schematic of the controller

5. SIMULATION RESULTS AND DISCUSSION

The controller described above was reviewed by running a double lane change simulation on a flat road at 40 km/h using a loaded trailer. A loaded trailer is used due to the fact that it is more unstable than that of an unloaded trailer. All simulations were ran using an Ftire tyre model. These simulations were run both with and without the controller in order to compare the two. The initial states given to the Single-Track Model are all zero and are updated continually as the articulated vehicle moves through the manoeuvre. The simulations were also performed using a soft suspension only as a softer suspension does not handle as well as a hard one and therefore the instability of the vehicle is increased. The yaw rates of both the tow vehicle and the trailer as well as the hitch angle are analysed. The simulation results for 40 km/h are depicted in Figure 10.

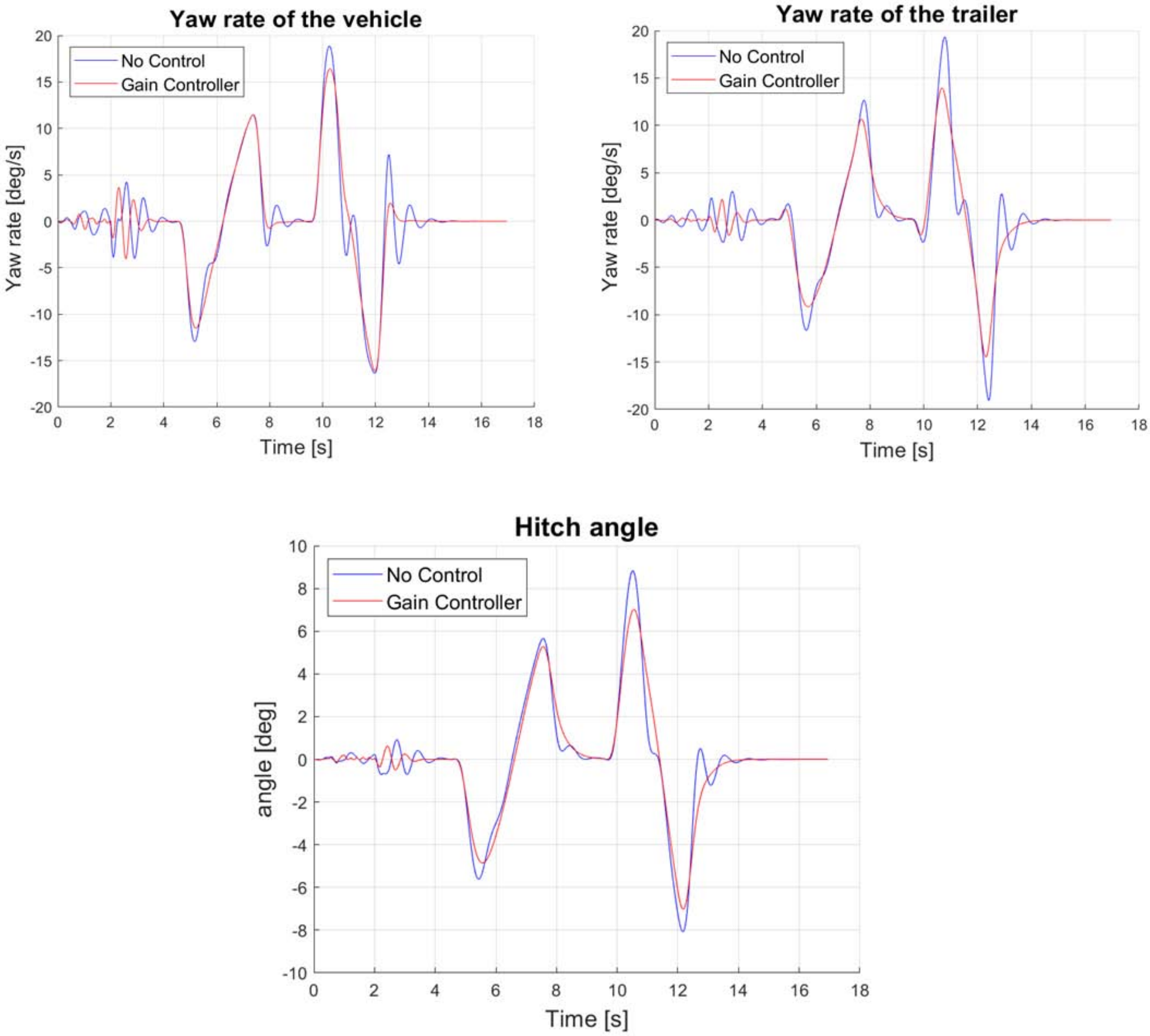

Figure 10: Results for a fully loaded trailer on a flat road at 40 km/h with control

From Figure 10, it can clearly be seen that the yaw rates, hitch rate and hitch angle decreased due to the gain controller in comparison to the results without a controller. It was also noticed for every situation considered that the controller removed a lot of the small peaks seen in the plots and replaced them with a smoother function. This simple controller has ultimately proven its worth as it can now be confidently stated that a brake-based control system on the trailer can be used to stabilise an articulated vehicle system. These results serve to prove that trailer braking is worth investigating and that a more complex and elegant control system should be developed.

6. CONCLUSION

In this paper a Single-Track Model was derived to be used as a reference model for the controller. The knowledge gained through the derivation of the mathematical models was applied when constructing the full multi-body dynamics simulation model. The results from the validation tests show that a good correlation is achieved between the model and the articulated vehicle therefore proving that the model is a good representation of the articulated vehicle. The main objective or focus for this study is to find a way to stabilise an articulated vehicle autonomously. In order to do so, a control system is needed. For a proof of concept, it was found that brake-based yaw moment control on the trailer can significantly improve the stability hence improving the safety of articulated vehicles and reducing fatalities. This study highlights the possibilities and indicates that significant improvement can be achieved using control. Since we now have a model with the ability to be controlled, the controller itself can be greatly improved by adding complexity to the design and further increasing its ability to stabilise all types of articulated vehicles. Since the work in this paper has proved that instability can be improved for a vehicle-trailer combination, it opens the door for this type of braking control to be applied to all articulated vehicle including those used for freight and logistics. This shall be a new area to explore but the current research shows promise. Torque vectoring can be applied to all systems that have the ability to be braked as these systems can be easily modified to do so. This type of control is more suited for modern articulated vehicles that have some form of braking system already existing. A braking system would have to be created for trailers that do not have brakes which shall introduce an extra expense.

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APPENDIX A – STM Matrix Parameters

This appendix contains the matrix parameters of the STM that was derived in Section 2.

$$\mathbf{M} = \begin{bmatrix} m_1 + m_2 & -m_2(c_1 + a_2) & -m_2 a_2 & 0 \\ m_1 c_1 & I_{z1} & 0 & 0 \\ -m_2 a_2 & I_{z2} + m_2 a_2 (c_1 + a_2) & I_{z2} + m_2 a_2^2 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}$$

$$\mathbf{D} = \begin{bmatrix} -\frac{\mathrm{C_{yf} + C_{yr} + C_{yt}}}{\mathrm{v_x}} & -\frac{C_{yf} a_1 + C_{yr} b_1 + C_{yt}(c_1 + l_2) - (m_1 + m_2) v_x^2}{v_x} & \frac{C_{yt} l_2}{v_x} & C_{yt} \\ -\frac{C_{yf}(a_1 + c_1) + C_{yr} e_1}{v_x} & -\frac{C_{yf}(a_1 + c_1) + C_{yr} b_1 e_1 - m_1 c_1 v_x^2}{v_x} & 0 & 0 \\ \frac{C_{yt} l_2}{v_x} & -\frac{C_{yt} l_2 (c_1 + l_2) + m_2 a_2 v_x^2}{v_x} & -\frac{C_{yt} l_2^2}{v_x} & -C_{yt} l_2 \\ 0 & 0 & 1 & 0 \end{bmatrix}$$

$$E = \begin{bmatrix} C_{yf} \\ C_{yf}(a_1 + c_1) \\ 0 \\ 0 \end{bmatrix}$$