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A Longitudinal Speed Controller for Autonomous Multi-Trailer Articulated Heavy Vehicles

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Abstract**—**This paper presents an automated longitudinal speed controller for multi-trailer articulated heavy vehicles (MTAHVs). A 6 degrees of freedom (DOF) yaw-plane vehicle model is generated to represent a MTAHV with the configuration of A-train double. A vehicle states prediction approach and a Mamdani fuzzy interface system are utilized to devise the automated driving controller for forward speed control of the MTAHV. Due to multiple articulation joints and heavy and long architectures, MTAHVs exhibit low high-speed lateral stability. They often experience amplified lateral motion of trailing units in transient curved path negotiations. Most of the speed planning schemes and control strategies introduced in the literature have been proposed for single unit vehicles. To enhance the automated speed control performance of the MTAHV, an anticipatory/compensatory lateral acceleration controller strategy considering the states of all the vehicle units and the MTAHV performance envelope is proposed. This speed controller distinguishes itself from others with several features. To evaluate the effectiveness of the innovative speed control strategy, co-simulations are carried out by combining the nonlinear A-train double model generated in TruckSim with an integrated controller designed in MATLAB/ SIMULINK.

Keywords- multi-trailer articulated heavy vehicles; autonomous driving control; speed planner; speed controller; fuzzy control; co-simulation.

I. INTRODUCTION

Around 1.25 million people are killed per year in road vehicle accidents all around the world [1]. To tackle the safety problem, active vehicle safety systems (AVSSs) have been commercialized [2]. However, these systems don't normally consider the impact of human driver errors, while human drivers' mistakes causes (around 94%) of road collisions [3]. One solution to this problem is to eliminate the human factors from the control loop, i.e., applying autonomous driving [4].

However, the vast majority of research activities dedicated to autonomous driving have been focused on single-unit vehicles (SUVs), e.g., cars [5]. Due to their environmental and economic advantages, MTAHVs are increasingly used worldwide [6-8]. Notwithstanding the pros, MTAHVs exhibit low lateral stability and poor maneuverability due to high center of gravity (CG), big sizes and multi-unit structures [9-10]. Moreover, in highway operations, MTAHVs represent as 7.5 times higher risk than SUVs [11]. Albeit, few attempts have been made to study predictive safety systems (PSSs) for MTAHVs. In recent years, some studies have focused on autonomous driving of construction trucks [12], articulated construction vehicles [13–14], and automated reverse parking for articulated vehicle [14]. However, these studies only consider low-speed motion planning and path tracking while ignoring high-speed features of MTAHVs, e.g. rollover, jack-knifing and trailer sway.

The main objectives of autonomous vehicles designs are to increase the transportation efficiency and to enhance the vehicle safety. MTAHVs demonstrate poor dynamic performance in high-speed evasive maneuvers, under which trailing vehicle units often experience larger lateral motions than the leading unit. Rearward amplification (RWA) is an indicator for evaluating the amplified lateral motion of the rearmost trailer. It is indicated that the ideal value of RWA should be 1.0 [15-16]. High RWA values imply high safe risk in highway operations. MTAHVs with B- and C-train configurations typically have an RWA measure around 1.5. The situation would be even worse for an A-train double with RWA of 2 [17]. Any attempt of designing autonomous MTAHVs disregarding the high-speed lateral stability features of these vehicles may not be acceptable.

Attempts have been made for longitudinal motion control for SUVs. In a study [18], the driving behavior considering the driver capability envelope was analyzed during braking a car in a turn through a closed-loop simulation. This approach can be inferred as an automated braking scheme for a car, but it did not consider throttling. In another research [19], the driver's steering and speed control performance was investigated for a SUV while negotiating a curved path. It was assumed that the vehicle should reach a predefined speed at the circular part of the road. Indeed, the proposed methodology was not autonomous and the speed was not adjusted based on road curvatures and driving strategies.

A convex optimization-based speed planning strategy for a heavy truck was studied considering both the acceleration and deceleration demand for a path with varying curvatures [20]. A two-level control strategy for longitudinal motion control of a truck was proposed [21]. The authors used the engine and brake system states for the speed control. The model introduced a reference speed to follow and speed was not planned based on road curvatures. A clothoid-based speed profiler and control using a receding horizon fashion was introduced for a SUV during a low-speed S-curve path negotiation, but the vehicle performance envelope and high-speed transient maneuvers functionality of the designed controller were disregarded [22]. Unfortunately, there is no published autonomous speed controller in the literature dedicated to MTAHVs considering their unique dynamics features.

This paper proposes a speed profiler and control strategy considering curvature of the target path and all the vehicle units' states. The proposed controller works using two anticipatory and compensatory lateral acceleration control approaches via adjusting acceleration/deceleration to form a speed profile in order to reach the safe maximum speed and minimum trip time.

The rest of the paper is organised as follows. Section II introduces the vehicle modelling and the virtual driver model for path-following. The proposed speed planning and control strategy is described in section III. A fuzzy speed controller is introduced in section IV. Simulation results are discussed in section V. Finally, the conclusions are drawn in section V.

II. Vehicle And Driver Modelling

A. *Six DOF yaw-plane model*

Fig.1 illustrates a 6-DOF yaw-plane vehicle model. The motions considered include the tractor's longitudinal, lateral and yaw motions as well as trailers' and dolly's yaw motions. For deriving the governing equations of motion, the following assumptions are made: 1) the longitudinal dynamics equations have been neglected, but the vehicle units' velocities are variables and are updated with real-time values generated by the nonlinear TruckSim model during closed-loop simulation at each time step; 2) dynamics of the wheels are neglected; 3) roll motion and pitch motions are ignored; 4) side-slip angles of all units are small; 5) lateral tire forces have linear relation to the tire sleep angles; 6) products of variables are neglected; 7) articulation angles and leading unit steering angle are small.

The state-space representation of the dynamics equations is:

$$\begin{aligned} \dot{x} &= A_d x + B_d u \\ y &= C_d x \end{aligned} \tag{1}$$

where x is state vector and u is the steering input. Ad, Bd and Cd are matrices. The matrices are then used for driver model design.

B. *Driver Model*

To mimic the driver steering functionality to control the lateral motion of the MTAHV, a predictive driver model is used. MacAdam driver model is one of the most reputed and verified ones in the literature, and is based upon optimal preview control for linear systems [23]. This driver model is used in the current study for vehicle steering control, in which varying longitudinal speed is updated at each time step using the tractor speed of the virtual A-train double, i.e., the nonlinear TruckSim model and the relevant matrices coded in MATLAB are updated with the

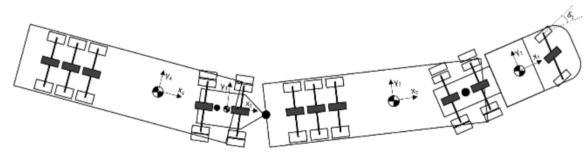


Figure 1. Schematic representation of the A-train double.

new speed value. Furthermore, the body-fixed coordinate system of the tractor is applied to find the vehicle states over the prediction horizon which is considered to be 1 second herein.

With (1), the following quadratic cost function (2) should be minimized to find the optimal tractor front axle steering demand for following a target path.

$$J \triangleq \frac{1}{T}\int_{t}^{t+T} \{[f(\eta) - y(\eta)]^2 W(\eta - t)\} d\eta \tag{2}$$

where f(η) is the previewed desired path, y(η) is the predicted lateral position of the vehicle over the preview time and W(η-t) is the weighting factor over the preview interval.

III. Longitudinal Motion Control Methodology

A driver is gathering the visual cues from the surroundings and combining the gained information with the knowledge that he/she has about the dynamics of the vehicle to make decisions based on all the data collected. Those decisions are then compiled into predictive and corrective reactions by choosing appropriate steering and throttling/braking to follow a path safely.

The lateral driver model introduced in section II demonstrated reliable performance in different maneuvers and at various vehicle speeds considering all vehicle units' motion cues and can be used as the lateral control strategy during the automated driving operation of the MTAHV. To design an autonomous/semi-autonomous MTAHVs, longitudinal motion control shouldn't be ignored. To control the acceleration of the A-train double by applying appropriate brake force or throttling for adjusting the vehicle speed, a fuzzy logic control strategy is employed.

In regular highway driving scenarios, adjusting forward speed is not such crucial. This issue becomes vital during obstacle avoidance scenarios or curve negotiations. Upon approaching a curve or confronting a transient lane-change maneuver, the driver needs to make decision well in advance as the strategy will be used for manipulating the steering-wheel and throttle/brake pedals. Drivers usually consider the preview curvature of the road over a preview distance relative to the current speed of the vehicle. Over a curved path negotiation, the vehicle's lateral acceleration is correlated with the forward speed; a high forward speed will lead to a high lateral acceleration, which may trigger a rollover. Thus, drivers try to adjust vehicle forward speed considering road curvature. Actually, the precision and efficiency of vehicle speed adjustment is based on the driver's experience and knowledge about the handling characteristics and throttle/brake response delay of the vehicle.

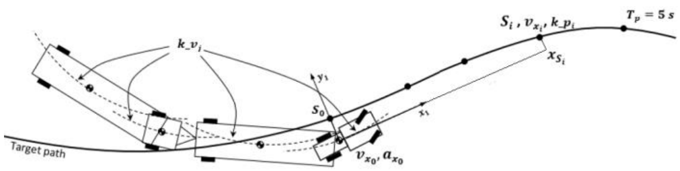


Figure 2. Schematic diagram representing the anticipatory and compensatory control points for MTAHV speed planning

However, even skillful drivers may make a mistake when estimate the braking demand for curve negotiation due to external disturbances, e.g., poor visibility and harsh weather condition. Hence, a compensatory approach is necessary to mitigate inappropriate vehicle state estimation over the preview distance. This is done using a supplementary compensation braking/throttling once the vehicle is on the previewed spot of the curve and the driver has the realistic estimation of the lateral acceleration, thereby applying the brake force by a reasonable value may assure that the lateral acceleration does not exceed a threshold.

To mimic the predictive and corrective actions of the driver, the acceleration controller comprises an anticipatory and a compensatory approach. For the anticipatory approach, 10 preview points are considered in front of the Tractor's CG. The preview distances are tractor's speed-dependent and they are evenly distributed over a 5-second prediction time, i.e. 0.5 second time gap between two adjacent points. The reason for choosing 10 preview points is as follows: if only one point in front of the tractor is selected, at high vehicle speeds, the preview distance would be long; as a result, the controller may probably ignore any path curvature alteration between the current position of the vehicle and the preview point due to either a sudden change in planned motion or an obstacle avoidance maneuver. The selection of 10 points assures that at any simulation time step the whole preview distance is observed, and the best speed control scheme will be adopted according to the previewed states of the vehicle. This will be explained in details later in the paper.

Based on the global x and y coordinates of the target path, a curve is fitted on the path ahead of the tractor's CG, then, the curvature of the road section associated with each preview point is separately estimated. Fig.2 shows the schematic diagram of the anticipatory acceleration control approach. The current states of the tractor include longitudinal speed V_{x0}, longitudinal acceleration a_{x0}, current station of the tractor's CG on the path S_0. The current states' values are treated as their initial values at next simulation time step. It is assumed that the longitudinal acceleration is constant over the preview distance. The anticipated states over the preview time T_p can be estimated by (3).

$$\begin{aligned} V_{x_i} &= a_{x_0}.T_{p_i} + V_{x_0} \\ S_i &= \frac{1}{2}.a_{x_0}.\left(T_{p_i}\right)^2 + V_{x_0}.T_{p_i} + S_0 \end{aligned} \tag{3}$$

where $i = 1, 2, \dots, 10$ represents the i^{th} preview point on the target path, V_{xi} is the predicted speed and S_i is the previewed station of the tractor's CG on the i^{th} point of the trajectory. As mentioned earlier using curve fitting on the target path coordinates in front of the tractor's CG, curvature for each preview point, k_p_i is estimated. It is important to mention that since only the road coordinates in front of the tractor are used for anticipatory speed control strategy, the path global coordinates ahead of the tractor's CG are mapped to its body-fixed coordinate system (x,y) to find the fitted curve y = f(x). Note that curve-fitting function in MATLAB is used for the purpose. Equation (4) is used to transform the global coordinates to the local coordinate system of the tractor.

$$\begin{aligned} x &= (Y - Y_0)\sin\psi + (X - X_0)\cos\psi \\ y &= (Y - Y_0)\cos\psi - (X - X_0)\sin\psi \end{aligned} \tag{4}$$

where (X_0, Y_0) are the global coordinate of the tractor's CG at each simulation time step. Having the previewed stations S_i calculated by (3) and the equation of trajectory y = f(x), which is recursively calculated in MATLAB at each time step, the local coordinates of the preview points x_{Si} are estimated by solving (5).

$$S_i = \int_{S_0}^{x_{S_i}} \sqrt{1 + \left(\dot{f}(x)\right)^2}\,dx \tag{5}$$

It is assumed that at each time step the origin of tractor's body-fixed coordinate is on the S_0, hence in the above equation S_0=0. Then the attained longitudinal local coordinates are used to calculate the curvatures as follows.

$$k\_p_i = \frac{\left|\ddot{y}(x_{S_i})\right|}{\left[1 + \left(\dot{y}(x_{S_i})\right)^2\right]^{\frac{3}{2}}} \tag{6}$$

As noted earlier, since the preview distances are speed-dependent, when the vehicle speed is high even the closest preview point may be rather far from the tractor's CG for the controlling purposes, and as a result, the anticipatory approach may introduce some errors to the current demanding acceleration/deceleration and when MTAHV reaches to the previewed position on the path, the speed and consequent lateral acceleration will be different from what expected.

Hence, a corrective approach should also be deemed for the acceleration controller design. The corrective reaction of the driver is modelled using the same kinematic equations as for the predictive approach; however, in this case the real-time lateral acceleration of all the MTAHV units' CGs as well as the estimated real instantaneous curvatures of the paths followed by towing and trailing units' CGs are employed to calculate the demand acceleration/deceleration. Fig.2 also illustrates the instantaneous curvatures of the vehicle units. These curvatures are different from the corresponding ones on the road, because trailing units will not be able to exactly follow the trajectory due to the articulation points as well as the high-speed steady-state and transient off-tracking or the low-speed path-following off-tracking especially in sharp turns negotiations. The so-called instantaneous curvatures are estimated by (7).

$$k\_v_j = \frac{a_{y_j}}{{V_{x_j}}^2} \tag{7}$$

where j=1 to 4 denotes the vehicle unit's number beginning with the tractor up to the second trailer respectively. a_{yj} and V_{xj}

represent the real-time values of the lateral accelerations and longitudinal velocities for all the units' CGs.

The main strategy for acceleration/deceleration automated control in the current paper is to drive the MTAHV as speedily and safely as possible while following a trajectory with variable curvatures. To reach this goal it is presumed that the maximum lateral acceleration of the vehicle should be kept around a safe threshold during carrying out a wide range of maneuvers and negotiating various road curvatures. As there is a direct relation between the lateral acceleration and longitudinal speed i.e. $a_{y_i} = V_{x_i}{}^2 . k_i$, accomplishing this objective ensures the highest speed while having a safe ride.

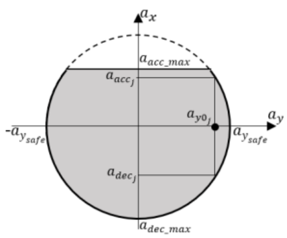


Figure 3. MTAHV performance envelope diagram

Static rollover threshold which is usually demonstrated by lateral acceleration in gravitational unit is a valuable measure to investigate the rollover stability. The mentioned parameter can be as low as 0.25 g in some harsh driving scenarios and load conditions for an AHV which is not equipped with any roll stability control system like active trailer steering (ATS), active roll control (ARC) or trailer differential braking (TDB) but drivers can often drive their vehicles with a lateral acceleration around 0.2 g [17]. Besides, as the linear vehicle dynamics model of MTAHV is utilized for the purpose of steering and speed control, the maximum lateral acceleration of all the vehicle units should be less than 0.35 g to prevent non-linear dynamics excitation [9]. A typical A-train double which is studied in the current paper usually demonstrates a maximum rearward amplification of 2 [17]. Hence, a maximum lateral acceleration of 0.25 g was considered as the safe lateral acceleration $a_{y_{safe}}$, for all the segments of the road with curvatures greater than zero.

The performance of a MTAHV can be depicted by a graph named "g-g" diagram [24]. Maximum attainable lateral acceleration and longitudinal acceleration/deceleration synthesize a curve which envelopes the g-g diagram and is called performance envelope of the vehicle. The capability envelope's shape relies on a number of factors comprising vehicle speed, driver's driving skill, tire/road friction condition and so forth. But, in the current study a theoretical operating envelope is applied and it is assumed that the autonomous speed controller should react similar to a skilled driver's capabilities. In the current study the capability envelope during acceleration/deceleration is described using (8) [18].

$$\left(\frac{|a_x|}{a_{x_{\max\_dec}}}\right)^2 + \left(\frac{|a_y|}{a_{y_{safe}}}\right)^2 \leq 1 \qquad (8)$$

Fig.3 illustrates the exaggerated graphical representation of the capability envelope in this paper (the grey area). Under forward acceleration the grip between road surface and tire is not the only effective factor similar to the braking process and the engine power would also be a limiting factor. That is why the limit for the forward acceleration a_{acc_max}, is different from that of for the braking i.e. a_{dec_max}. The maximum longitudinal acceleration/deceleration are defined based on the TruckSim built-in A-train double model engine capability and road friction condition. Fig.4 shows a graph including a flowchart for the acceleration/deceleration demand calculation. The final value $a_{x\text{-}ctrl}$ is then used as the input for the fuzzy controller to estimate the required brake pedal force or throttle which will be later sent to the TruckSim MTAHV model via SIMULINK S-function as the speed control signal.

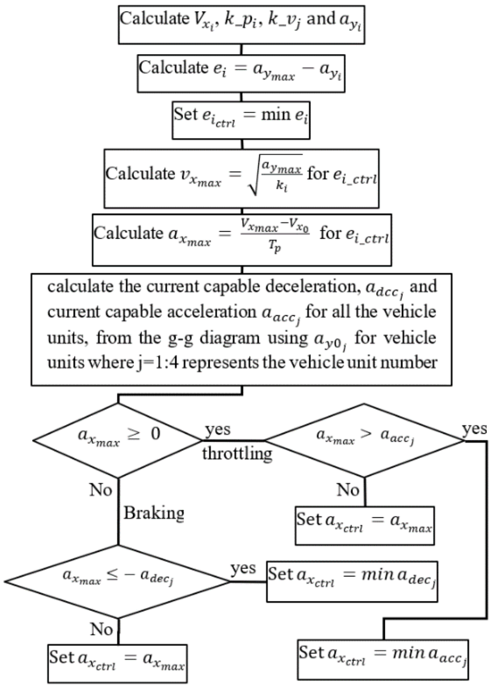
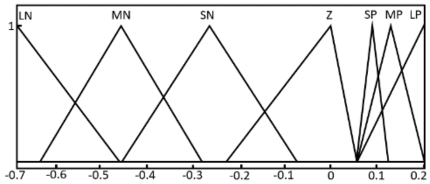


Figure 4. Speed planning and control strategy flowchart

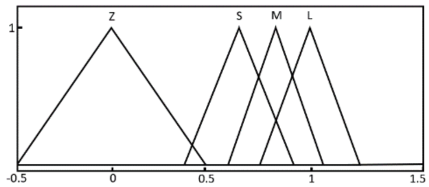
Based on the performance envelope demonstrated in Fig.3 if the lateral acceleration of the MTAHV reaches to the a_{ysafe}, the capable deceleration or acceleration would be zero. Hence, to ensure that there always exists a reserved amount of throttle or braking capability for the vehicle when reaches to the previewed station of the road, the desired lateral acceleration a_{ymax} is deemed to be a bit less than the $a_{y_{safe}}$ for anticipatory acceleration calculation approach. In the current study a_{ymax} = 0.24g, $a_{acc\text{-}max}$ = 0.06g and $a_{dec\text{-}max}$ = -0. 6g.

IV. FUZZY CONTROLLER

Typical control strategies usually employ an input-output mathematical model of a vehicle. However, fuzzy controller doesn't use the mentioned model or precise plant parameters. Instead, it relies on the mapping the inputs to the outputs via membership functions and fuzzy rules [25]. Fuzzy systems don't necessitate using linear plant model and provide a model-free estimation of a non-linear plant. The precision and validity of control outputs highly depend upon choosing correct membership functions and fuzzy rules among numerous options which are defined based on the experience and knowledge of the designers [26].



(a)



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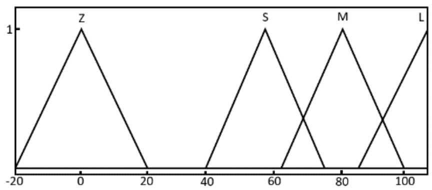


(c)

Figure 5. Membership functions (a) Acceleration demand as input variable (b) Throttle as first output and (c) Pedal brake force as second output

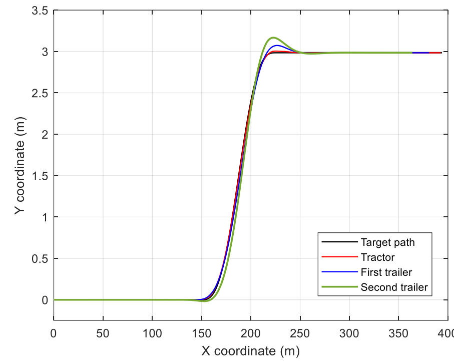
The driver model explained in the previous section of this study takes advantage of a linear vehicle dynamics model. However, since during the vehicle acceleration control strategy the velocity changes continuously, a non-linearity will be introduced to the state-space matrices. To tackle this issue, at every time step the state-space matrices coded in MATLAB are updated by new vehicle's speed provided by the TruckSim. Fig.5 illustrates the fuzzy input and output membership functions.

V. SIMULATION RESULTS AND DISCUSSION

To evaluate the performance of the proposed speed controller, co-simulations using MATLAB/SIMULINK and TruckSim were conducted. The controller was codded in MATLAB and a VS S-function block in SIMULINK was utilized to receive the real-time states from and send the control signals including the brake force and engine throttle to the TruckSim.

Two driving scenarios i.e. high-speed lane change (HSLC) and a 180-degree curve negotiation were designed to verify the controller. In the first scenario the MTAHV is moving with an initial speed of 108 km/h. Then it performs a single-lane-change maneuver with the maximum lateral displacement of 2.984 m. To have a good estimation about the controller performance, two simulations for the SLC maneuver with and without the controller were carried out and the results were compared. The achieved results are shown in the following graphs.

Form Fig.6 it is inferred that using the speed controller the transient off-tracking has been remarkably reduced for all the vehicle units. The highest decline is seen for the dolly and the second trailer. This result is achieved because the speed controller considers the lateral acceleration of all the trailing units and not just the tractor. This interesting result proves that the proposed speed controller can also operate as a reactive safety system (RSS) for autonomous MTAHV in some high-speed transient maneuvers taking advantage of the anticipatory/compensatory lateral acceleration control approach and compromising the speed.



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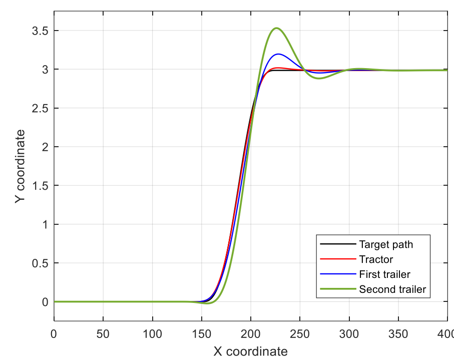
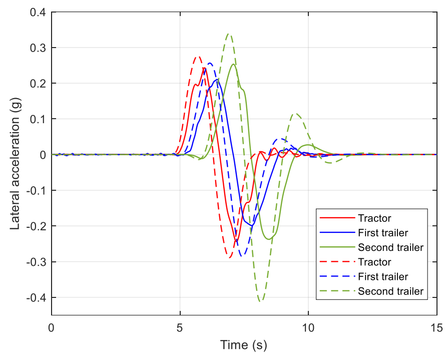


(b)

Figure 6. Lateral displacement for HSLC maneuver (a) with controller (b) without controller

The lateral acceleration for all the MTAHV units is illustrated in Fig.7 (a). From the graph it is deduced that the designed controller performs very well to keep the maximum lateral acceleration within the desired range i.e. 0.24 g in the current study. Among the trailing units second trailer has the highest lateral acceleration which is exactly the same as the target value for the controlled situation. In comparison, when the controller is deactivated the second trailer's lateral acceleration is a about -0.42 g which means a 75% rise. Another interesting point to be mentioned is that when the speed controller is off the rearward amplification is 1.5, while when the controller is activated the rearward amplification is 1.0 which assures a stable obstacle avoidance maneuver.

Fig.7 (b) depicts the longitudinal speed manipulation to reaching the desired lateral acceleration based on the target path. When the controller is off, it is assumed that the speed is kept almost constant during the maneuver as seen in the graph. When the vehicle starts the SLC maneuver, the speed is decreased as required by the controller in order to prevent the lateral acceleration goes beyond the threshold. Once all the vehicle units pass the curvilinear path and enter the straight part of the maneuver, the speed begins increasing again.



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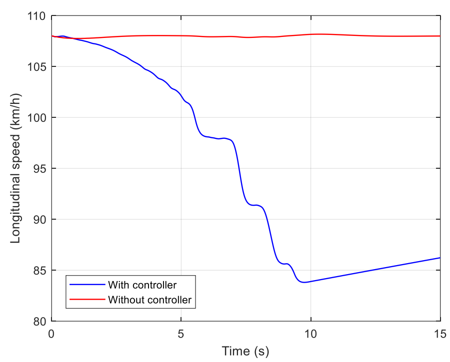
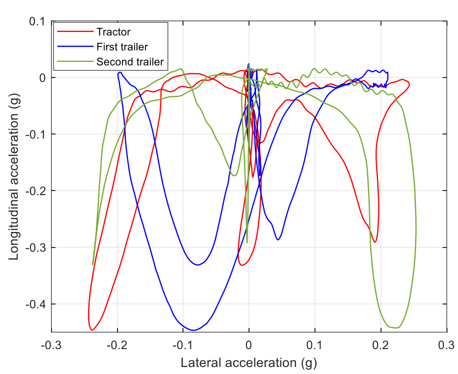


(b)

Figure 7. A-train double HSLC maneuver results (a) lateral acceleration with controller (solid line) and without controller (dashed line) (b) Longitudinal speed

Fig.8 (a) and (b) show the g-g diagram for situations in which the speed controller is activated and deactivated respectively. When the controller is on, the vehicle performance envelope is used more efficiently just like what a skilled driver tries to do to drive safely while exploiting the capabilities of the vehicle as much as possible. However, when the controller is off the vehicle capability is not used appropriately and there is a risk of rollover because of an RWA value of more than 1.5, very similar to a novice driver performance during an urgent situation.



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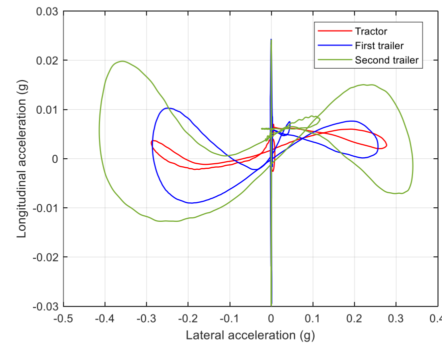


(b)

Figure 8. g-g diagrams for HSLC maneuver (a) with controller (b) without controller

The next driving scenario is a rather sharp curve negotiation with an initial speed of 100 km/h [18]. Fig.9 demonstrates the curvature of the path with respect to the road station. The road involves five segments including two straight part with zero curvature, a circular bend which has constant curvature, and two clothoids which have linearly varying curvatures and connect the bend to the straight path smoothly.

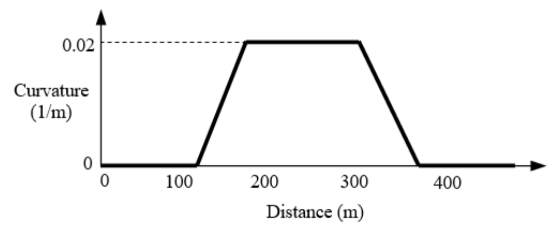
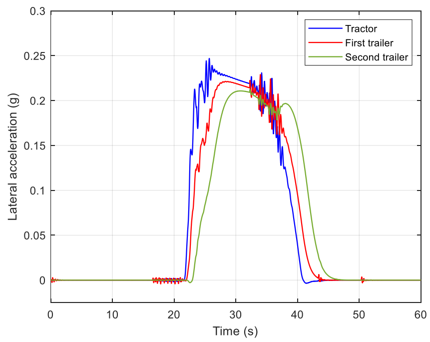


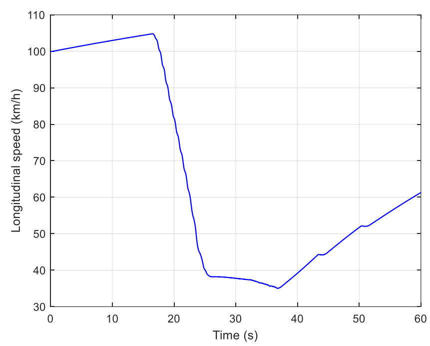
Figure 9. The target path curvature diagram for U-turn negotiation maneuver

Simulation results are yielded in Fig.10. From Fig.10 (a) it is well understood that the speed controller successfully kept the maximum lateral acceleration which in this case belongs to the tractor, within the expected range. The trailing units have also similar lateral acceleration during negotiating the bend segment of the road. It is really promising for an A-train double that can achieve and keep the desired lateral acceleration while having a high speed and enters a rather sharp turn for such a long vehicle with three articulation points.

Fig.10 (b) represents the speed change graph to reaching the previously mentioned valuable goal. Speed value initially goes up because the vehicle moves on the straight part of the path. Then the controller begins reducing the speed while arriving the clothoid up to somewhere near 35 km/h where the road has the highest curvature. This speed is well kept between 35 km/h and 40 km/h during passing the circular course. Upon exiting the bend and arriving the second clothoid, speed gradually increases based on the engine power limits and feasible MTAHV acceleration. The graph shows just 60 second of the simulation to show the general speed control process.



(a)



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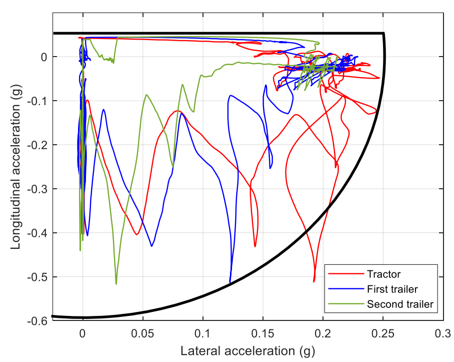

(c)

Figure 10) U-turn negotiation results (a) Lateral acceleration (b) Longitudinal speed and (c) g-g diagram

The g-g diagram for the mentioned driving scenario is demonstrated in Fig.10 (c). Since whole the MTAHV maneuver is performed either straight or while turning to the left, almost all the lateral and longitudinal accelerations are depicted in one quarter of graph. The thick black lines show the limits of the vehicle performance envelope. First of all, it is seen that the controller has kept the lateral and longitudinal acceleration between the thresholds i.e. 0.06 g for longitudinal acceleration, 0.6 g for longitudinal deceleration and 0.24 g for the lateral acceleration. Furthermore, similar to the HSLC maneuver, in this case the controller has exploited almost all the feasible acceleration/deceleration taking into account the predefined constraints very similar to a professional human driver which tries to keep the MTAHV dynamics operating close to the performance limits. This will increase the vehicle speed and minimize the trip time especially during long journeys.

VI. Conclusion And Future Improvements

Findings acquired through this study are summarized as follows:

i) A new autonomous speed planner/controller was proposed for an A-train double MTAHV considering realistic operation and decision-making process of a skilled human driver for driving over different roads with various curvatures.

ii) The main objective of the autonomous speed controller is to maintain the lateral vehicle acceleration within the maximum safe range when negotiating curvilinear paths which assures the highest capable vehicle speed as well.

iii) The acceleration/deceleration autonomous control strategy uses a compensatory approach in collaboration with the anticipatory strategy to tackle the vehicle inaccurate states prediction over the preview horizon which may even happen for a skilled driver due to some unwanted conditions such as poor visibility.

iv) The compensatory approach utilizes all the vehicle units' lateral accelerations to plan the best speed control scenario to decline the lateral instability occurrence due to inappropriate vehicle throttling or braking effort while following a curved path

v) Vehicle performance envelope is also utilized to make sure that at every moment there is always some reserved braking capability for the MTAHV while the vehicle performance capabilities are exploited efficiently

vi) Since the proposed automated controller uses both previewed and real-time MTAHV states and the main goal is to limit the lateral acceleration within a desirable range by adjusting the vehicle speed, it can operate as a reactive safety system (RSS) to improve the lateral stability

vii) Using this MTAHV speed control strategy may reduce the trip time and increase the safety for long journeys

For the future work it is desirable that the wheels' dynamics will be integrated in the equations of motion. Besides, it may be beneficial if the longitudinal vehicle dynamics is also embedded in the speed controller design to make the model more precise. Finally, it is suggested that a more accurate driver model which is characterized for MTAHVs will be used instead of the current one which considers just the tractor's motion cues for the purpose of steering.

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