Lateral Stability Control of Articulated Heavy Vehicles Based on Active Steering System

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***Abstract*—The main purpose of this paper is to design a controller for improving the lateral stability of long heavy vehicle combinations based on active steering system. An augmented optimal linear quadratic control system design is implemented. The controller is developed and evaluated with step and lane change maneuvers for a truck and trailer combination. The uncertainties masses of the truck and trailer are taken into account for analysis purpose. The nonlinear and linear model of the truck and trailer are presented. The simulation results show a decrease in yaw rate rearward amplification and sideslip angles significantly with successful desired yaw rate tracking for the trailer.**

***Index Terms*—active steering, linear quadratic control, yaw rate rearward amplification, sideslip angle, lateral stability**

I. Introduction

Recently, articulated heavy vehicles are so important part of transport of goods. They are generally very heavy and enormous volume vehicles travelling on the main roads, and the safety of drivers of these long-articulated vehicles and other passengers and other drivers on their vehicles is an essential concern. Therefore, to avoid accidents, the vehicle dynamic behavior must be stable and lateral performance must be as quite good as possible.

Two vital safety issues require improvement: poor lateral performance and rollover tendency at high velocities, which easily drive to accident. In order to improve lateral performance of the articulated heavy vehicles braking and steering based lateral control systems are proposed in the literature.

In a previous study, control of lateral motion of multiple articulated heavy vehicles had done by using electronic braking control system [1]. The lateral acceleration rearward amplification is used in this paper to evaluate the lateral performance. The yaw and lateral dynamics control problem is considered for a truck-dolly-semitrailer combination. A predictive control approach is presented to reduce the yaw rate rearward amplification and to prevent rollover of the rearmost unit, the acceleration lateral is limited, and simulation results are tested for a single lane change maneuver [2]. A feedback and feedforward active steering control system is designed for several types of heavy vehicle combinations such as truck-center axle trailer, tractor-semitrailer, truck-full trailer, Truck–B-double, tractor-semitrailer-center axle trailer, etc. Furthermore, different heavy vehicle combinations are made in both frequency and time domains [3]. The control system mentioned in [3] is applied experimentally to a truck-dolly-semitrailer combination in [4] with a feedback and feedforward active steering control system. Also, the simulation results are obtained for different heavy vehicle combinations using a sinusoidal lane change maneuver. They are compared for lateral acceleration rearward amplification, yaw rate rearward amplification, and off tracking. In [5], an active steering control system based on fuzzy logic is introduced with a novel desired articulation angle calculation method. The proposed control system is examined with low speed 90-degree turn and high-speed lane change. In [6], two PID type controllers are used to improving the lateral performance of a truck and trailer, step and lane change maneuver are applied in the simulation to truck and trailer combination. Uncertainties of masses are taken into account; the results show better performance in controlled case than the uncontrolled case. In [7], the stability of a semitrailer is investigated based on Neural Network Control. The results show that the neural network under the control of the center of mass of the train tractor-semitrailer improved the stability of the vehicle such as, side-slip angle, yaw rate, the longitudinal acceleration, etc. The robust Linear Quadratic Gaussian (LQG) and the μ synthesis control techniques are employed for designing the active trailer steering control system in [8], the control techniques are analyzed by using numerical simulations. The results showed that the μ synthesis control technique is achieved desired system performance in presence of parametric uncertainties and noise whereas LQG control technique effectively controls the system in the presence of noise. Two controllers are developed for a truck-dolly semitrailer combination and is evaluated in a sine with dwell maneuver, the purpose of designing controllers is to improve lateral stability of a truck-dolly-semitrailer combination, the first control is used to steer the semitrailer axles and the second one to steering both the dolly and the semitrailer, the results show that the improvement in lateral stability is higher for the second controller in [9]. Integral-plus-state feedback controller

Manuscript received February 13, 2022; revised June 23, 2022.

doi: 10.18178/ijmerr.11.8.575-582

and a sliding mode controller are investigated to control of the linear dynamics of a truck and trailer combination in [10], the results demonstrate the superiority of the sliding model over the first controller by tracking of the truck and trailer to a target path. A fuzzy controller and a PID controller are designed for an automated steering articulated vehicle, both controllers work independently, the aim of controllers is to regulate the steering angle of all wheels, the results show that the off tracking in both tractor and trailer can be reduced even in very sharp curves in [11]. The effects of several lay-outs and parameters on the lateral dynamic behavior of heavy vehicles is presented in [12], as well the effect of multiple axles is investigated, the purpose of that research to get comprehensive understanding on the conventional vehicle combinations. In [13], improving maneuverability and stability of articulated vehicle are conducted, two issues are investigated, the poor maneuverability at low speeds and losing their stability in different manners at high speeds. The Linear Quadratic Regulator optimal control approach is presented to reduce tracking error at low-speed, and modulate Rearward Amplification ratio and roll at high-speed. A predictive control approach based on active steering has been presented to minimize the yaw rate rearward amplification in [14], lane change maneuver is applied in the simulation to truck-dolly-semitrailer combination. The lateral and yaw dynamics control problem is considered. In [15], the research presents an anti-jackknife apparatus for improving lateral stability of tractor semi-trailer, anti-jackknife apparatus was designed to avoid the jackknife during emergency braking, the simulation results show that the control method which used can improve the lateral stability of heavy vehicle under some conditions such as emergency braking at high speed. In [16], a comprehensive review study is conducted including classification and comparison of schemes for improving lateral stability of car-trailer combinations.

In this article, the optimal control system (LQI) based on active steering systems are proposed to control the truck and trailer yaw rates. The linearized model is used to obtain desired yaw rate value of the truck. The trailer desired yaw rate is calculated from the desired truck yaw rate. The control system is tested with step and sinusoidal lane change maneuvers by counting nominal and uncertain masses for the truck and trailer. The lateral performance of the overall system is evaluated by the yaw rate rearward amplification values. The main contribution in this study is that the tractor and trailer steering angles are actuated, whereas in previous researches only the trailer steering angles are actuated. Therefore, both unites, the truck and trailer can be controlled separately in this work.

The outline of the rest of the paper is as follows. The nonlinear truck and trailer model and linearized model are presented in Section II. In Section III, calculation of desired yaw rate for truck and trailer unit and active steering feedback control system are introduced. Simulation studies and evaluation of the results are given in Section IV. The paper ends with conclusion.

II. Vehicle Models

In vehicle dynamic studies, the truck and trailer combination illustrated in Fig. 1 is considered.

Details of nonlinear truck and trailer models are given and they are used for simulating the lateral dynamics in step and lane change maneuvers, while the linear model of the truck and trailer combinations are used for controller development phase.

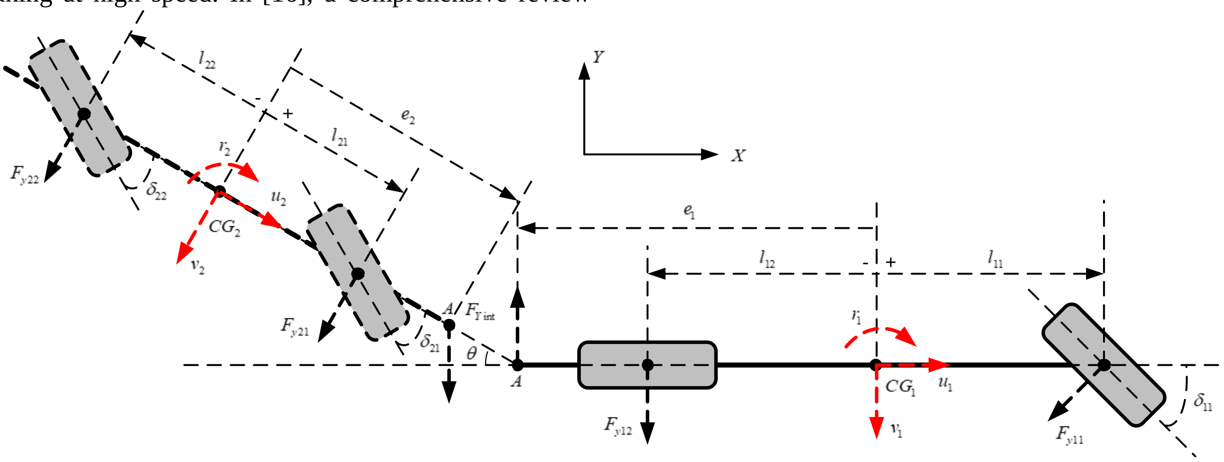


Figure 1. Truck and trailer model for active steering.

A. Nonlinear Truck Model

The vehicle combinations under this study are a two-units as a truck and a trailer. The nonlinear equations of motion of a two-units combinations are provided under some assumptions: (i) a constant and equal longitudinal velocity for both units are considered; (ii) rolling dynamics, suspension dynamics and the effect of longitudinal forces are neglected.

The lateral tire forces in the tire coordinate system,

$$F_{Yij} = F_{yij}\cos\left(\delta_{ij}\right) \tag{1}$$

The equations of motion for the first unit are

$$m_1\left(\dot{v}_1 + u_1 r_1\right) = \sum_{j=1}^{n} F_{Y1j} - F_{Y\text{int}} \tag{2}$$

$$I_{z1}\dot{r}_1 = \sum_{j=1}^{n} F_{Y1j} l_{1j} - F_{Y\,\mathrm{int}} e_1 \,. \tag{3}$$

Similarly, the equations of motion for the trailer unit are

$$m_2\left(\dot{v}_2 + u_2 r_2\right) = \sum_{j=1}^{k} F_{Y2j} + F_{Y\,\mathrm{int}} \cos\theta \tag{4}$$

$$I_{z2}\dot{r}_2 = \sum_{j=1}^{k} F_{Y2j} l_{2j} + F_{Y\,\mathrm{int}} e_2 \cos\theta \,. \tag{5}$$

where m_i , I_{zi} , v_i , r_i , u_i are the mass, the moment of inertia, lateral velocity, yaw rate of the unit and longitudinal velocity respectively. e_i is the distance between the center of gravity of the ith unit and the articulation point. θ denotes the articulation angle. n and k are the numbers of the axles for the lead and the towed unit respectively. $F_{Y\,\mathrm{int}}$ is the internal lateral force at the articulation point. l_{ij} is the distance between the cog of the ith unit and the jth axle of ith unit.

Elimination of $F_{Y\,\mathrm{int}}$ from these equations and combined (2), (4) and (2), (3) and (4), (5) to obtain final motion equations:

$$\begin{aligned} & m_1\left(\dot{v}_1 + u_1 r_1\right)\cos\theta + m_2\left(\dot{v}_2 + u_2 r_2\right) \\ & = \sum_{j=1}^{n} F_{Y1j} \cos\theta + \sum_{j=1}^{k} F_{Y2j} \end{aligned} \tag{6}$$

$$I_{z1}\dot{r}_1 - m_1 e_1\left(\dot{v}_1 + u_1 r_1\right) = \sum_{j=1}^{n} F_{Y1j}\left(l_{1j} - e_1\right) \tag{7}$$

$$I_{z2}\dot{r}_2 - m_2 e_2\left(\dot{v}_2 + u_2 r_2\right) = \sum_{j=1}^{k} F_{Y2j}\left(l_{2j} - e_2\right) \tag{8}$$

Kinematic constraint equation is

$$\left(v_1 + e_1 r_1\right)\cos\theta - u_1 \sin\theta = v_2 + e_2 r_2 \,. \tag{9}$$

Differentiating (9) results in

$$\begin{aligned} & \left(\dot{v}_1 + e_1\dot{r}_1\right)\cos\theta - \left(v_1 + e_1 r_1\right)\left(r_2 - r_1\right)\sin\theta - \dot{u}_1 \sin\theta \\ & - u_1\left(r_2 - r_1\right)\cos\theta = \dot{v}_2 + e_2\dot{r}_2 \end{aligned} \tag{10}$$

Assuming constant and equal longitudinal velocity for both truck and trailer units, instead of u_1 and u_2 , u can be used in the equations. Also, $\dot{u}$ can be taken as zero in (10).

A nonlinear MATLAB/Simulink model of two unit's combination can be built by using (6) - (8) and (10), whose inputs are the steering angles $\left(\delta_1, \delta_2\right)$ and the outputs are the yaw rates $\left(r_1, r_2\right)$ and the lateral velocities $\left(v_1, v_2\right)$ of the units. The vehicle parameters can be found for different combinations in [8].

B. Linear Truck Model

In vehicle dynamic investigation, the linear model is prominently used for the design phase of control systems. For linearization truck and trailer, some assumptions are used: (i) steering and articulation angles are small; (ii) constant linear tires and longitudinal velocities and equal for both units.

The lateral forces can be written as:

$$F_{Yij} = C_{ij}\left(\delta_{ij} - \frac{v_i + l_{ij} r_i}{u}\right) \tag{11}$$

where C_{ij} is the cornering stiffness of the jth axle of the ith unit.

The nonlinear equations of motion (6) – (8) and (10) can be linearized to

$$\begin{aligned} & m_1\left(\dot{v}_1 + u r_1\right) + m_2\left(\dot{v}_2 + u r_2\right) = C_{11}\left(\delta_{11} - \frac{v_1 + l_{11} r_1}{u}\right) + C_{12}\left(\delta_{12} - \frac{v_1 + l_{12} r_1}{u}\right) \\ & + C_{21}\left(\delta_{21} - \frac{v_2 + l_{21} r_2}{u}\right) + C_{22}\left(\delta_{22} - \frac{v_2 + l_{22} r_2}{u}\right) \end{aligned} \tag{12}$$

$$\begin{aligned} & I_{z1}\dot{r}_1 - m_1 e_1\left(\dot{v}_1 + u r_1\right) = C_{11}\left(\delta_{11} - \frac{v_1 + l_{11} r_1}{u}\right)\left(l_{11} - e_1\right) \\ & + C_{12}\left(\delta_{12} - \frac{v_1 + l_{12} r_1}{u}\right)\left(l_{12} - e_1\right) \end{aligned} \tag{13}$$

$$\begin{aligned} & I_{z2}\dot{r}_2 - m_2 e_2\left(\dot{v}_2 + u r_2\right) = C_{21}\left(\delta_{21} - \frac{v_2 + l_{21} r_2}{u}\right)\left(l_{21} - e_2\right) \\ & + C_{22}\left(\delta_{22} - \frac{v_2 + l_{22} r_2}{u}\right)\left(l_{22} - e_2\right) \end{aligned} \tag{14}$$

$$\dot{v}_1 + e_1\dot{r}_1 - u_1\left(r_2 - r_1\right) = \dot{v}_2 + e_2\dot{r}_2 \tag{15}$$

Finally, the linear truck and trailer model is transferred to state-space form as

$$\dot{x} = Ax + Bu = T^{-1}\bar{A}x + T^{-1}\bar{B}u \tag{16}$$

where the states are $x = \left[v_1 \quad r_1 \quad v_2 \quad r_2\right]^T$,the inputs are $u = \left[\delta_1 \quad \delta_2\right]^T$, and the remaining matrices can be calculated as follows:

$$T = \begin{bmatrix} e_1 m_1 & -I_{z1} & 0 & 0 \\ 0 & 0 & e_2 m_2 & -I_{z2} \\ m_1 & 0 & m_2 & 0 \\ -1 & -e_1 & 1 & e_2 \end{bmatrix},$$

$$B = \begin{bmatrix} C_{11}\left(e_1 - l_{11}\right) & 0 \\ 0 & C_{21}\left(e_2 - l_{21}\right) + C_{22}\left(e_2 - l_{22}\right) \\ C_{11} & C_{21} + C_{22} \\ 0 & 0 \end{bmatrix},$$

$$\bar{A} = \begin{bmatrix} a_{11} & a_{12} & 0 & 0 \\ 0 & 0 & a_{23} & a_{24} \\ a_{31} & a_{32} & a_{33} & a_{34} \\ 0 & u & 0 & -u \end{bmatrix}. \tag{17}$$

where the coefficients are

$$a_{11} = -\frac{C_{11}+C_{12}}{u}e_1 - \frac{C_{11}l_{11}+C_{12}l_{12}}{u}$$

$$a_{12} = -\frac{C_{11}l_{11}+C_{12}l_{12}}{u}e_1 + \frac{C_{11}l_{11}^2+C_{12}l_{12}^2}{u} - e_1 m_1 u$$

$$a_{23} = -\frac{C_{21}+C_{22}}{u}e_2 - \frac{C_{21}l_{21}+C_{22}l_{22}}{u}$$

$$a_{24} = -\frac{C_{21}l_{21}+C_{22}l_{22}}{u}e_2 + \frac{C_{21}l_{21}^2+C_{22}l_{22}^2}{u} - e_2 m_2 u$$

$$a_{31} = -\frac{C_{11}+C_{12}}{u},\ a_{32} = -\frac{C_{11}l_{11}+C_{12}l_{12}}{u} - m_1 u$$

$$a_{33} = -\frac{C_{21}+C_{22}}{u},\ a_{34} = -\frac{C_{21}l_{21}+C_{22}l_{22}}{u} - m_2 u \cdot$$

Table I shows the truck and trailer parameters.

TABLE I. TRUCK AND TRAILER PARAMETERS

Symbol	description	truck	trailer
m_i	Sprung mass of the *i*-th unit[kg]	15000	25000
k_i	, radius of gyration, unit [m]	1.44	2.41
l_{ij}	j-th axle distance from CG of the i-th, unit [m]	[2.5, -2.5]	[0.68, -0.68]
I_{zi}	Moment of inertia about z axis of the *i*-th unit [kgm2]	21600	60250
C_{ij}	*j*-th axle cornering stiffness of the i-th unit [kN/rad]	[356,480]	[432,432]
d_{fi}	Front coupling distance from CG of the *i*-th unit [m]	NA	7
d_{ri}	Rear coupling distance from CG of the *i*-th unit [m]	-3	NA

III. DESIRED VALUE CALCULATIONS AND CONTROL METHOD

The aim of this section is to present some principles and explanation about calculating desired yaw rate values of the truck, trailer, and also introduce control method that used for active steering control. The control system structure and the signal flow are presented as below in Fig. 2.

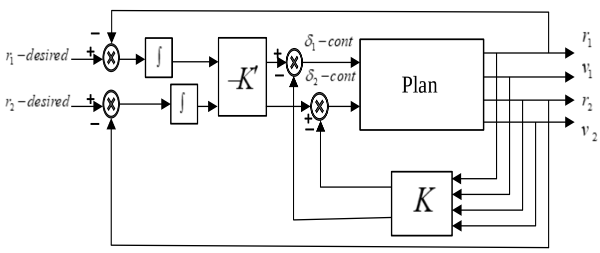


Figure 2. Control system structure.

A. Desired Yaw Rate Calculation for Truck and Trailer

The desired yaw rate values of the truck and trailer are required for proposed control system structure. The desired truck yaw rate is calculated by using the linearized model as follows:

$$r_{1des} = G_{r_1/\delta_1}\delta_1 \tag{18}$$

where G_{r_1/δ_1} is the transfer function between the yaw rate of the truck unit and steer angle of the same unit:

$$G_{r_1/\delta_1} = \begin{bmatrix}0 & 1 & 0 & 0\end{bmatrix}(sI-A)^{-1}\begin{bmatrix}B(1,1)\\B(2,1)\\B(3,1)\\B(4,1)\end{bmatrix}. \tag{19}$$

The desired trailer yaw rate of the trailer is considered as a time delayed version of the truck yaw rate in order to follow the same desired yaw rate values with a pure delay τ:

$$r_{2des} = r_{1des}e^{-\tau s} \tag{20}$$

where τ is taken as 1/*v* [9].

B. LQI Control Method

LQR control is an optimal control method, which is used for the state feedback design. Whereas, LQI control besides the simple state feedback used in LQR, has the output feedback with integral action. The purpose of placed integrator into the controller to eliminate the steady state error between the controlled variable and the control reference [17]-[19]. The block diagram of the LQI servo system where the plant has no integrator is shown in Fig. 3.

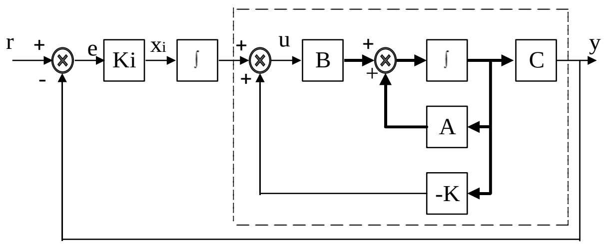


Figure 3. LQI control system .

Since the LQI control increases the number of states that will define the control input u:

$$u = -kx + k_i x_i \tag{21}$$

The conventional LQI problem is to obtain the control input u which minimizes the following cost function:

$$J(t) = \int_0^\infty \left(x^T(t)Qx(t) + u^T(t)Ru(t)\right)dt \tag{22}$$

The LQI control system design has been obtained by using the lqi MATLAB command. The values of Q and R weighting matrices are randomly chosen and can be varied until getting the desired values.

By choosing the appropriate values of Q and R matrices in MATLAB, the control gain matrix can be calculated as follows:

$$k_{LQI} = \begin{bmatrix} k_{11} & k_{12} & k_{13} & k_{14} & k_{15} & k_{16} \\ k_{21} & k_{22} & k_{23} & k_{24} & k_{25} & k_{26} \end{bmatrix} \quad (23)$$

IV. Simulation Studies

In this paper, LQI controller is used as active steering control system. In order to test the proposed controller system, four different simulation studies are conducted. Since the rearward amplification is used as a performance indicator to present increased risk for a rollover or swing out of the last unit compared to the lead unit, it is used to evaluate the controller. Also, the rearward amplification of lateral motions causes path deviation and large side slip. For all of that the yaw rate rearward amplification is used to evaluate the controller and should be reduced to 1 [4].

Rearward Amplification is defined as the ratio of the maximum absolute value of the motion variable of interest of a following vehicle unit to that of the lead unit during a specified maneuver [9], [20]. It can be defined as follows;

$$\text{yaw rate RWA} = \frac{\max |r_{trailer}|}{\max |r_{truck}|} \quad (24)$$

The first simulation is step steering angle with 0.5 sec, the amplitude of the steering angle is 5 degrees, and the truck velocity is taken as 80 km/h. The driver steering input and control signals for active steering are given in Fig. 4.

The step responses of the uncontrolled and controlled truck and trailer combinations can be seen from Fig. 5.

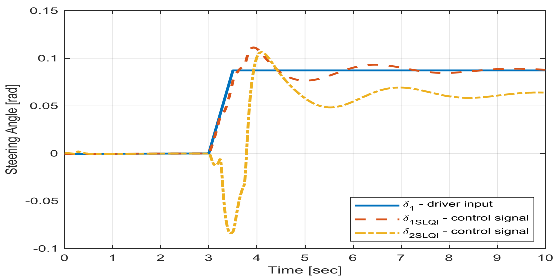
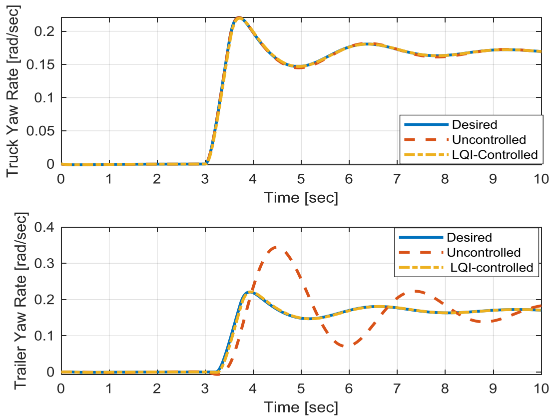


Figure 4. Simulation 1: The step steering input and the calculated control signals.



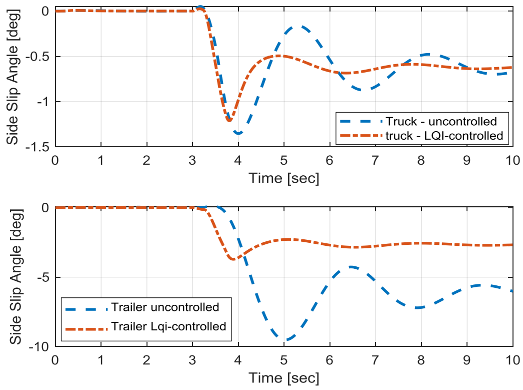


Figure 5. Simulation 1: The step responses of the uncontrolled and controlled truck and trailer combinations.

It is clear that the uncontrolled truck and the controlled case yaw rate responses are following desired yaw rate successfully. It is expected to have similar results for truck in linear region, because the desired yaw rate calculation model is the linearized model of the nonlinear vehicle model. On the other hand, the uncontrolled trailer shows quite oscillatory motion and could not follow the desired trailer yaw rate values. In contrast with the uncontrolled case, the controlled trailer follows the desired yaw rate values successfully. The sideslip angle of the trailer unit is decreased significantly. The yaw rate rearward amplification value is decreased from 1.5595 to 1.0064. It is consistent with the objective of being close to 1. It is obvious that, the LQI controller gives better performance in this term.

To examining the controller in a more realistic situation, uncertainties masses of truck and trailer are taken into account. By increasing the mass of truck 25% and the trailer 40%, a second simulation is conducted. The driver steering input and control signals for active steering are calculated as shown in Fig. 6.

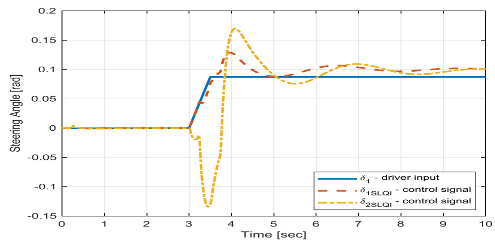
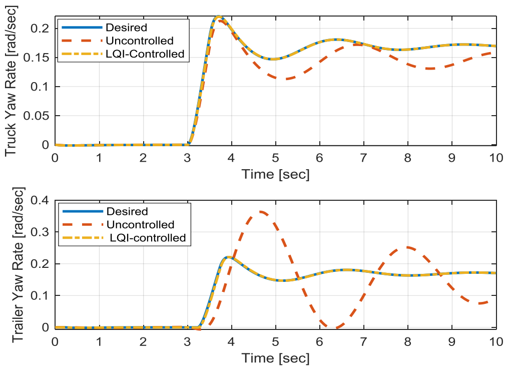


Figure 6. Simulation 2: The step steering input and the calculated control signals



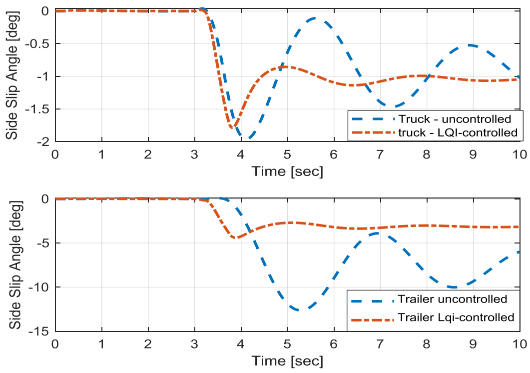


Figure 7. Simulation 2: The step responses of the uncontrolled and controlled truck and trailer combinations considering uncertainties.

Fig. 7 shows the step response of uncontrolled and controlled truck and trailer combination considering uncertainties. The results are consistent that of the first simulation. Uncontrolled case deviations from the desired values are increased due to the uncertainties in the truck and trailer masses.

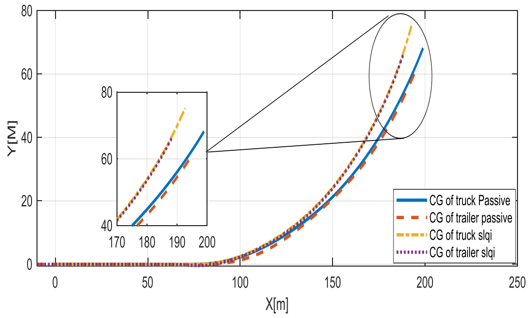


Figure 8. Simulation 2: Trajectories of the uncontrolled and controlled truck and trailer combinations for step input.

Fig. 8 shows the trajectories of the uncontrolled and controlled case. It is clear that when mass uncertainties are taken into account, uncontrolled trailer cannot follow its truck in a good way and swings out. In controlled case, it is obvious that the trailer follows its truck within the same lane.

The third simulation includes a sinusoidal steering input. It is a lane change maneuver. The steering input is selected as 3 degrees (0.0524 rad) amplitude, 0.4 Hz sinusoidal input. The truck velocity is selected as 80 km/h. Fig. 9 shows the calculated control signals for active steering and sinusoidal steering input for the lane change maneuver.

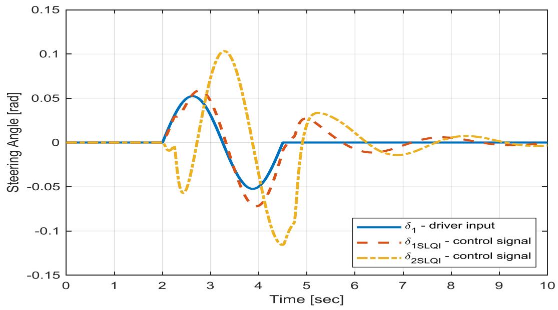


Figure 9. Simulation 3: The sinusoidal steering input and the calculated control signals.

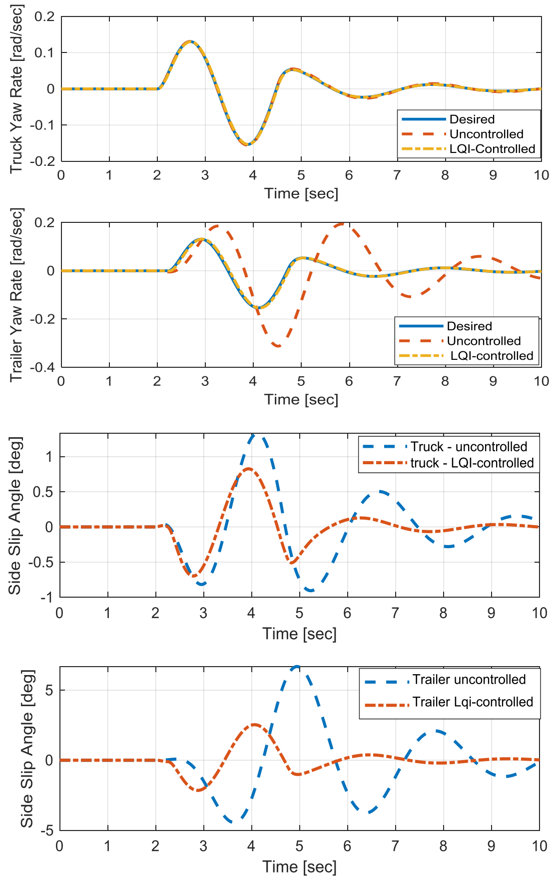


Figure 10. Simulation 3: The lane change maneuver responses of uncontrolled and controlled truck and trailer combinations.

Fig. 10 shows that the controlled case's truck and trailer yaw rate outputs are tracking the desired yaw rate values successfully. The designed LQI controller reduces the yaw rate RWA from 2.0086 to 1.0071. Moreover, it can be seen that sideslip angles for combination vehicles are decreased significantly particularly for towed unit, which means the trailer swing is eliminated.

The last simulation is a lane change maneuver with uncertain truck and trailer masses. 3 degrees and 0.4 Hz sinusoidal input are selected as the steering input. The uncertainties in the truck and trailer masses are taken as in the second simulation. In addition, the truck velocity is selected as 80 km/h. Fig. 11 shows the driver's sinusodial steering input and calculated control signals for active steering in the case of uncertain masses.

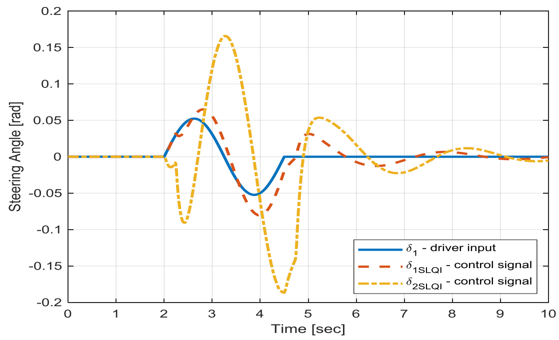


Figure 11. Simulation 4: The sinusoidal steering input and the calculated control signals.

Fig. 12 shows that with considering mass uncertainties, the yaw rates of controlled truck and trailer are following the desired values succesfully. The yaw rate rearward amplification is reduced from 2.0015 to 1.0043 by using the LQI controller. Moreover, it can be seen that the sideslip angles are decreased obviously.

According to the results shown in Table II, it can be understood that the proposed control system decreased the yaw rate RWA values and it increased the lateral performance of the truck and trailer vehicle combination.

TABLE II. YAW RATE RWA VALUES FOR DIFFERENT CASES

Yaw rate RWA	Uncontrolled truck and trailer	controlled truck and trailer	Uncontrolled truck and trailer with uncertainty	controlled truck and trailer with uncertainty
Lane change maneuvers RWA	2.0086	1.0071	2.0015	1.0043
Unit step maneuvers RWA	1.5595	1.0064	1.7081	1.0037

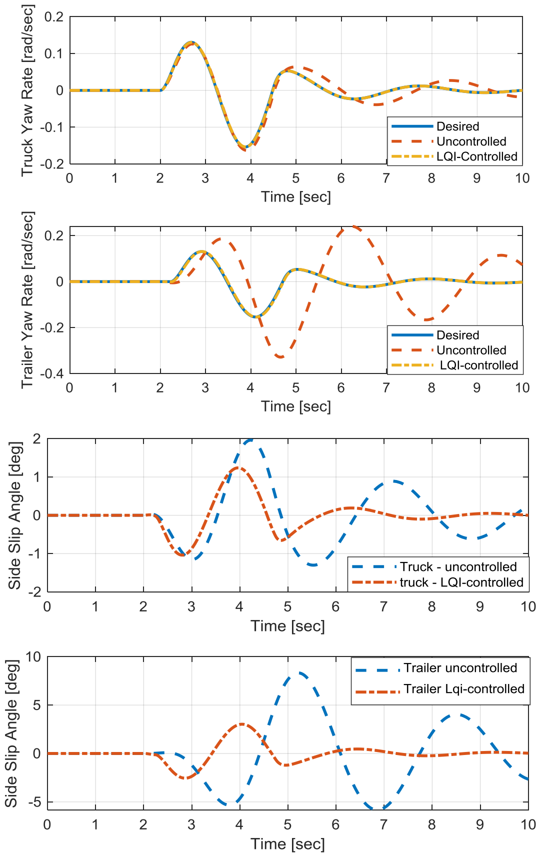


Figure 12. Simulation 4: The lane change maneuver responses of the uncontrolled and controlled truck and trailer combinations considering uncertainties.

V. CONCLUSION

LQI controller based on active steering control system for articulated heavy vehicle was proposed. Four different simulations were conducted to test the proposed control system by using step steering and sinusoidal steering inputs (lane change maneuver). The uncertainties of truck and trailer masses were taken into account in the simulation studies. Yaw rate RWA was used as a lateral performance indicator. The simulation results indicated that LQI controller improves the vehicle lateral performance. Furthermore, the LQI optimal controller reduces the yaw rate rearward amplification significantly which means that LQI controller succeeds to prevent large path deviation, trailer swing and large side slip.

CONFLICT OF INTEREST

The authors declare no conflicts of interest.

AUTHOR CONTRIBUTIONS

All authors contributed to design, implementation and analysis of the research. Mustafa A. Emheisen and Mümin Tolga Emirler were involved in the drafting of the final manuscript. Mustafa A. Emheisen completed the research under the supervision of Mümin Tolga Emirler and Basar Ozkan.

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