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-semi-trailer 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

doi: 10.18178/ijmerr.11.8.575-582
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,

\[ F_{ij} = F_{ij} \cos(\delta_i) \]  

The equations of motion for the first unit are

\[ m_i (v_i + u_i r_i) = \sum_{j=1}^{n} F_{i1j} - F_{i1m} \]
\[ I_{j}\dot{r}_j = \sum_{j=1}^{n} F_{1j} I_{j} - F_{j\text{um}} e_i. \]  

Similarly, the equations of motion for the trailer unit are

\[ m_z (\dot{v}_z + u_z r_z) = \sum_{j=1}^{n} F_{zj} I_{zj} + F_{z\text{um}} \cos \theta. \]  

\[ I_{zj}\dot{r}_z = \sum_{j=1}^{n} F_{zj} I_{zj} + F_{z\text{um}} e_z \cos \theta. \]

where \( m_\text{z}, I_\text{z}, v_\text{z}, r_\text{z}, u_\text{z} \) 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 \( i\text{th} \) unit and the articulation point. \( \theta \) denotes the articulation angle. \( n \) and \( k \) are the numbers of the axles for the lead and the towed unit respectively. \( F_{j\text{um}} \) is the internal lateral force at the articulation point. \( I_j \) is the distance between the cog of the \( i\text{th} \) unit and the \( j\text{th} \) axle of \( i\text{th} \) unit.

Elimination of \( F_{j\text{um}} \) from these equations and combined (2), (4) and (3) and (4), (5) to obtain final motion equations:

\[ m_\text{z} (\dot{v}_z + u_z r_z) \cos \theta + m_z (\dot{v}_z + u_z r_z) \sin \theta = \sum_{j=1}^{n} F_{1j} I_{j} \cos \theta + \sum_{j=1}^{n} F_{1j} I_{zj} \cos \theta \]  

\[ I_{zj}\dot{r}_z - m_z e_z (\dot{v}_z + u_z r_z) = \sum_{j=1}^{n} F_{zj} I_{j} (I_j - e_i) \]  

\[ I_{zj}\dot{r}_z - m_z e_z (\dot{v}_z + u_z r_z) = \sum_{j=1}^{n} F_{zj} I_{zj} (I_{zj} - e_z). \]

Kinematic constraint equation is

\[ (v_1 + e_1 r_1) \cos \theta - u_1 \sin \theta = (v_2 + e_2 r_2). \]

Differentiating (9) results in

\[ (\dot{v}_1 + e_1 \dot{r}_1) \cos \theta - (v_1 + e_1 r_1)(r_2 - r_1) \sin \theta - u_1 (r_2 - r_1) \cos \theta = \dot{v}_2 + e_2 \dot{r}_2. \]

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 units' combination can be built by using (6) - (8) and (10), whose inputs are the steering angles \( (\delta_1, \delta_2) \) and the outputs are the yaw rates \( (\dot{r}_1, \dot{r}_2) \) and the lateral velocities \( (v_1, v_2) \) 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:

\[ F_{yj} = C_y \left( \delta_j - \frac{v_j + l_j r_j}{u} \right) \]

where \( C_y \) is the cornering stiffness of the \( j\text{th} \) axle of the \( i\text{th} \) unit.

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

\[ m_\text{z}(\dot{v}_z + u z r_z) + m_z(\dot{v}_z + u z r_z) = C_{11} \left( \delta_1 - \frac{v_1 + l_1 r_1}{u} \right) + C_{12} \left( \delta_2 - \frac{v_2 + l_2 r_2}{u} \right) \]

\[ + C_{21} \left( \delta_1 - \frac{v_1 + l_1 r_1}{u} \right) + C_{22} \left( \delta_2 - \frac{v_2 + l_2 r_2}{u} \right) \]

\[ I_{zj}\dot{r}_z - m_z e_z (\dot{v}_z + u z r_z) = C_{11} \left( \delta_1 - \frac{v_1 + l_1 r_1}{u} \right) (I_{zj} - e_1) \]  

\[ I_{zj}\dot{r}_z - m_z e_z (\dot{v}_z + u z r_z) = C_{12} \left( \delta_2 - \frac{v_2 + l_2 r_2}{u} \right) (I_{zj} - e_1) \]

\[ v_1 + e_1 r_1 - u_1 (r_2 - r_1) = \dot{v}_1 + e_1 \dot{r}_1, \]

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

\[ x = Ax + Bu = T^{-1} \dot{\tilde{A}} x + T^{-1} \dot{B} u \]

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

\[ A = \begin{bmatrix} e_1 m_1 & -I_{z1} & 0 & 0 \\ 0 & 0 & e_2 m_2 & -I_{z2} \\ -1 & -e_1 & 1 & e_z \end{bmatrix} \]

\[ B = \begin{bmatrix} C_{11} (e_1 - l_1) & 0 \\ 0 & C_{21} (e_2 - l_1) + C_{22} (e_2 - l_2) \\ C_{11} & C_{21} + C_{22} \end{bmatrix} \]

\[ \tilde{A} = \begin{bmatrix} a_{11} & a_{12} & 0 & 0 \\ 0 & a_{21} & 0 & 0 \\ a_{31} & a_{32} & a_{33} & a_{34} \\ 0 & u & 0 & -u \end{bmatrix}. \]
\[ a_{11} = -\frac{C_{11} + C_{12}}{u} \varepsilon_1 + \frac{C_{11}l_{11} + C_{12}l_{12}}{u} \varepsilon_2 \]
\[ a_{12} = -\frac{C_{11}l_{11} + C_{12}l_{12}}{u} \varepsilon_1 + \frac{C_{11}l_{11}^2 + C_{12}l_{12}^2}{u} \varepsilon_2 - \varepsilon_1 \varepsilon_2 m_\mu \]
\[ a_{23} = -\frac{C_{21} + C_{22}}{u} \varepsilon_2 - \frac{C_{21}l_{21} + C_{22}l_{22}}{u} \varepsilon_2 - \frac{C_{11}}{u} \varepsilon_2 - \frac{C_{12}}{u} \varepsilon_2 - \varepsilon_1 \varepsilon_2 m_\mu \]
\[ a_{s1} = -\frac{C_{11}}{u} \varepsilon_1, \quad a_{s2} = -\frac{C_{11}l_{11} + C_{12}l_{12}}{u} \varepsilon_2 - m_\mu \]
\[ a_{s3} = -\frac{C_{21} + C_{22}}{u} \varepsilon_2 - \frac{C_{11}l_{21} + C_{12}l_{22}}{u} \varepsilon_2 - m_\mu \]

Table I shows the truck and trailer parameters.

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Truck</th>
<th>Trailer</th>
</tr>
</thead>
<tbody>
<tr>
<td>( m )</td>
<td>Sprung mass of the ( i )-th unit [kg]</td>
<td>15000</td>
<td>25000</td>
</tr>
<tr>
<td>( k )</td>
<td>Radius of gyration, unit [m]</td>
<td>1.44</td>
<td>2.41</td>
</tr>
<tr>
<td>( l )</td>
<td>( j )-th axle distance from CG of the ( i )-th unit, unit [m]</td>
<td>[2.5, -2.5]</td>
<td>[0.68, -0.68]</td>
</tr>
<tr>
<td>( I )</td>
<td>Moment of inertia about z axis of the ( i )-th unit [kgm²]</td>
<td>21600</td>
<td>60250</td>
</tr>
<tr>
<td>( C )</td>
<td>( j )-th axle cornering stiffness of the ( i )-th unit [kN/rad]</td>
<td>[356,480]</td>
<td>[432,432]</td>
</tr>
<tr>
<td>( d )</td>
<td>Front coupling distance from CG of the ( i )-th unit [m]</td>
<td>NA</td>
<td>7</td>
</tr>
<tr>
<td>( d )</td>
<td>Rear coupling distance from CG of the ( i )-th unit [m]</td>
<td>-3</td>
<td>NA</td>
</tr>
</tbody>
</table>

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:

\[ r_{des} = G_{n|\delta_i} \delta_i \]  

where \( G_{n|\delta_i} \) is the transfer function between the yaw rate of the truck unit and steer angle of the same unit:

\[ G_{n|\delta_i} = \begin{bmatrix} B(1,1) \\ B(2,1) \\ B(3,1) \\ B(4,1) \end{bmatrix} (sI - A)^{-1} \begin{bmatrix} 1 \\ 0 \\ 0 \\ 0 \end{bmatrix} \]  

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 \( \tau \):

\[ r_{des} = r_{des} + \tau e^{-\tau s} \]  

where \( \tau \) 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 \( u \):

\[ u = -kx + k_i x_i \]  

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

\[ J(t) = \int_{0}^{t} (x(t)^T Q x(t) + u(t)^T R u(t)) dt \]  

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:
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:

\[
\text{yaw rate } RWA = \frac{\max |F_{\text{trailer}}|}{\max |F_{\text{truck}}|}
\]  

(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.

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.

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.
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.

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.

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 sinusoidal steering input and calculated control signals for active steering in the case of uncertain masses.
Fig. 12 shows that with considering mass uncertainties, the yaw rates of controlled truck and trailer are following the desired values successfully. 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>
<thead>
<tr>
<th>TABLE II. YAW RATE RWA VALUES FOR DIFFERENT CASES</th>
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<tbody>
<tr>
<td>Yaw rate RWA</td>
</tr>
<tr>
<td>----------------</td>
</tr>
<tr>
<td>Lane change maneuvers RWA</td>
</tr>
<tr>
<td>Unit step maneuvers RWA</td>
</tr>
</tbody>
</table>

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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