

An Integrated Design Method for Articulated Heavy Vehicles with Active Trailer Steering Systems

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ABSTRACT

This paper presents an integrated design method for active trailer steering (ATS) systems of articulated heavy vehicles (AHVs). Of all contradictory design goals of AHVs, two of them, i.e. path-following at low speeds and lateral stability at high speeds, may be the most fundamental and important, which have been bothering vehicle designers and researchers. To tackle this problem, a new design synthesis approach is proposed: with design optimization techniques, the active design variables of ATS systems and passive design variables of trailers can be optimized simultaneously; the ATS controller derived from this approach has two operational modes, one for improving lateral stability at high speeds and the other for enhancing path-following at low speeds. To demonstrate the effectiveness of the proposed approach, it is applied to the design of an ATS system for an AHV with a tractor and a full trailer. Simulation results illustrate that compared with the baseline vehicle, the one derived from the design synthesis approach decreases low-speed off-tracking by 35.2% and reduces high-speed rearward amplification ratio by 30.0%. The proposed approach may be used for identifying desired design variables and predicting performance envelopes in the early design stages of AHVs with ATS systems.

INTRODUCTION

Articulated heavy vehicles (AHVs) exhibit unstable motion modes at high speeds, including jack-knifing, trailer swing and roll-over [1]. These unstable motion modes may lead to fatal accidents. An important dynamic performance measure for high-speed unstable motion modes is rearward amplification (RWA) ratio, which is defined as the ratio of

the peak lateral acceleration at the rearmost trailer's centre of gravity (CG) to that of the vehicle's towing unit in an obstacle avoidance lane-change maneuver [2, 3]. AHVs' high-speed lateral stability and, in particular, roll-over is highly dependent on the RWA ratio [4].

Due to AHVs' complex configurations and large sizes, these vehicles have poor path-following while traveling at low speeds on local roads and city streets [5]. AHVs' poor path-following causes the tires to scrub against the road during tight cornering, damaging both the tires and road surface [6]. Moreover, poor path-following of AHVs raises safety concerns for the surrounding traffic and damage of road infrastructure [7]. The commonly used performance measure for low-speed path-following is off-tracking [2]. Off-tracking is defined as the maximum radial offset between the path of the tractor's front axle centre and that of the trailer's rear axle centre during a specific maneuver [8].

In order to improve low-speed path-following of AHVs, several passive trailer steering systems, including self-steering, command steering, and pivotal bogie mechanisms, have been developed. A number of active trailer steering (ATS) systems have also been proposed for reducing path-following off-tracking. It is found that these passive and active trailer steering systems can improve path-following ability at low speeds. However, the high-speed lateral stability of the corresponding AHVs is inherently poor [2, 6]. This phenomenon matches a rule-of-thumb observation pertaining to the trade-off relationship between AHVs' low-speed path-following and high-speed lateral stability: "what one does to improve low-speed performance is likely to degrade high-speed performance and vice versa" [2]. The reason of this conflict lies in the fact that wheelbases and

number and locations of articulation points influence low-speed path-following and high-speed stability in opposite ways. Longer wheelbases and fewer articulation points will decrease high-speed RWA ratio, but they also increase low-speed off-tracking [2].

The last two decades have witnessed the advances in trying to find solutions to the contradictory design goals of path-following at low speeds and lateral stability at high speeds. Aurell and Edlund examined the influence of the location of passive steered axles on the dynamic stability of a tractor/semi-trailer system [9]. Jujnovich and Cebon compared the performance of various passive trailer steering systems [10]. Results derived from these studies imply that identifying optimal design variables, such as the trailer length, axle group location and weight distribution, can lead to the improvement of both high and low speed performance. Wu and Lin's research shows that an active multi-axle steering system for a tractor/full-trailer combination can improve the high-speed lateral stability [11]. El-Gindy et al used RWA ratio as a control criterion in the design of active yaw controllers for a tractor/full-trailer combination [3]. It is demonstrated that compared with the baseline vehicle, the one with the active trailer yaw controller can reduce the RWA level without significant change of the baseline vehicle trajectory. Recently, Rangavajhula and Tsao also used RWA ratio as the control criterion in the design of optimal trailer steering controllers based on the linear quadratic regulator (LQR) technique [7, 8]. Numerical simulations show that active trailer steering systems can achieve acceptable levels of RWA and off-tracking.

All the above studies focused on identifying the effects of either passive or active trailer steering systems on the contradictory design goals based on dynamic simulation and analysis. However, these studies have not addressed the trade-off relationship between path-following at low speeds and later stability at high speeds using design synthesis approaches. In these dynamic analyses of active trailer steering (ATS) systems of AHVs, it was assumed that the passive vehicle system was designed first and then the ATS systems developed were added onto the vehicle originally designed from a purely mechanical viewpoint. This sequential design approach may not achieve optimal trade-off solutions between the contradictory performance requirements of AHVs [12].

In the current research, a new design synthesis approach is proposed: with design optimization techniques, the active design variables of ATS systems and passive design variables of trailers can be optimized simultaneously; the ATS controller derived has two operational modes, one for improving lateral stability at high speeds and the other for enhancing path-following at low speeds. To demonstrate the effectiveness of the proposed approach, it is applied to the design of an ATS system for a tractor/full-trailer combination

based on a 3 degree-of-freedom (DOF) linear yaw plane model. It is expected that the proposed approach can be used for identifying desired design variables and predicting performance envelopes in the early design stages of AHVs with ATS systems.

The rest of this paper is organized as follows. Section 2 describes the 3 DOF linear yaw plane model used in the design synthesis of active trailer steering systems of articulated heavy vehicles. To evaluate the AHV performance in the lateral stability at high speeds and path-following at low speeds, the maneuvers emulated are also presented in Section 2. The design of the LQR controller for ATS systems is introduced in Section 3. The proposed design synthesis approach is presented in Section 4. Section 5 compares the design results derived from the proposed approach against those based on the baseline vehicle system without ATS systems. Finally, conclusions are drawn in Section 6.

VEHICLE SYSTEM MODELS

In order to evaluate the high-speed lateral stability and low-speed path-following of the tractor/full-trailer combination, the 3 DOF linear yaw plane model [11, 13] and the maneuvers emulated [2, 4, 10] are introduced in this section.

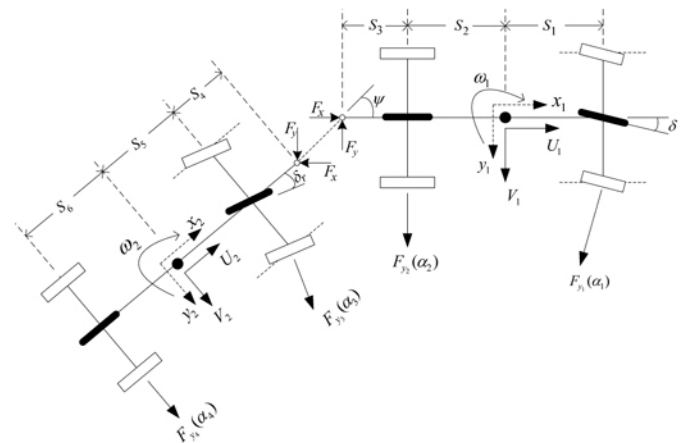


Fig. 1. Schematic diagram showing degrees of freedom and system parameters of vehicle model

VEHICLE MODEL

The articulated vehicle consists of a tractor and a full-trailer, which are connected by a hinge pin. As shown in Fig. 1, the system is telescoped laterally and each axle set is represented by one wheel. Based on the body fixed coordinate systems $x_1 - y_1$ and $x_2 - y_2$ for the tractor and trailer, respectively, the governing equations of the system can be derived. The motions considered are tractor lateral velocity V_1 , tractor yaw rate ω_1 and articulation angle ψ between the tractor and trailer. In this model, aerodynamic forces, rolling and

pitching motions, and longitudinal forces between tire and road are ignored. From Newton's laws of dynamics, the equations of motion for the tractor are written as

$$\begin{aligned} m_1(\dot{U}_1 - V_1\omega_1) &= F_x - F_{y1} \sin \delta \\ m_1(\dot{V}_1 + U_1\omega_1) &= -F_y + F_{y1}(\alpha_1) \cos \delta + F_{y2}(\alpha_2) \\ I_1\dot{\omega}_1 &= F_y d + F_{y1}(\alpha_1)S_1 \cos \delta - F_{y2}(\alpha_2)S_2 \end{aligned} \quad (1)$$

and the equations of motion for the trailer are cast as

$$\begin{aligned} m_2(\dot{U}_2 - V_2\omega_2) &= -F_{y3}(\alpha_3) \sin \delta_r - F_x \cos \psi - F_y \sin \psi \\ m_2(\dot{V}_2 + U_2\omega_2) &= F_{y4}(\alpha_4) + F_{y3}(\alpha_3) \cos \delta_r - F_x \sin \psi + F_y \cos \psi \\ I_2\dot{\omega}_2 &= -F_{y4}(\alpha_4)S_6 + F_{y3}(\alpha_3)S_5 \cos \delta_r + F_y e \cos \psi - F_x e \sin \psi \end{aligned} \quad (2)$$

where the notation is given in [Appendix A](#).

To derive the simplified vehicle model, the following assumptions have been made: 1) the forward speed U_1 remains constant; 2) the tractor steering angle δ is small; 3) the articulation angle ψ is small; 4) all products of variables are ignored; 5) the lateral tire force $F_{yi}(\alpha_i)$ is linear functions of sideslip angle α_i , $i = 1, \dots, 4$. The velocities of the pin using either of the coordinate systems should be compatible. Eliminating the coupling reaction forces F_x and F_y from [equations \(1\) and \(2\)](#) leads to the following 3 DOF linear yaw plane model expressed in the state space form as

$$\dot{\mathbf{x}} = \mathbf{Ax} + \mathbf{Bu} + \mathbf{C}\delta \quad (3)$$

where the matrices $\mathbf{A}$, $\mathbf{B}$, and $\mathbf{C}$ are provided in [Appendix B](#). The state vector $\mathbf{x}$ and control vector $\mathbf{u}$ are expressed as

$$\mathbf{x} = [V_1 \quad \omega_1 \quad \psi \quad \psi]^T \quad (4)$$

and

$$\mathbf{u} = [\delta_r] \quad (5)$$

Note that $\mathbf{u}$ is the control vector related to the trailer front axle tire steering angle.

MANEUVERS EMULATED

To evaluate AHVs' performance levels in high-speed RWA ratio and low-speed path-following off-tracking, two types of simulations have been used. The first type is based on an open-loop control approach with specified steer inputs,

including pulse and step steer inputs. These simulations require the precise application of a predetermined steer sequence and the resultant vehicle responses are observed. To assess the RWA and path-following levels, respectively, one-period sinusoidal steer inputs [7, 8, 9, 11] and step steer inputs [7, 8] have been used accordingly. The second type is essentially a closed-loop steer control process. In these simulations, a driver model is introduced and the vehicle model follows a well-defined path under the control of the driver model. During the process, the combined driver/vehicle models' responses are observed. To determine the RWA ratio of AHVs with semi-trailer steering systems based on the lane change maneuver recommended by SAE [14], a driver model has been used [10].

The proposed design synthesis approach, to be described in section 4, is based on a genetic algorithm, which requires extensive fitness function evaluations. This may lead to poor computation efficiency of the design approach. In the simulations based on the closed-loop steer control, the introduction of driver model will greatly increase computation time. If the driver model is introduced in the design approach, it will further degrade the computation efficiency. Thus, in the current research, the simulations based on the open-loop steer control are utilized to identify the high-speed and low-speed performance measures.

A single lane change maneuver is simulated for determining the high-speed RWA ratio. In the simulation, the vehicle is traveling at the speed of 88 km/h along a straight path and a sudden lane change is conducted. The lateral displacement of the vehicle in the lane change is 3.7 m. The steering input, in radians, for this maneuver takes a single sinusoidal wave as

$$\delta(t) = A \sin\left(\frac{2\pi}{T}t\right) \quad (6)$$

where the period T is 2.5 seconds and the value of amplitude A is selected in such a way that the vehicle is able to complete the single lane change.

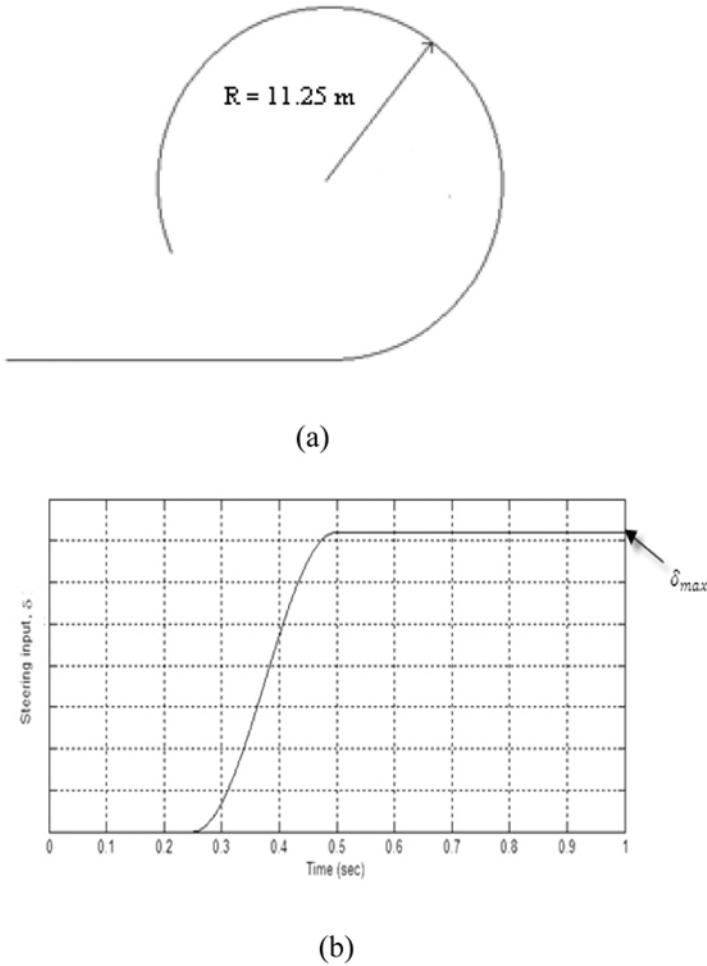


Fig. 2. (a) layout of circle path-following test course; (b) step steer input for circle path-following [4]

The maneuver emulated for measuring low-speed path-following off-tracking is based on a step steer simulation, which is similar to the 360° roundabout defined in the United Kingdom's Road Vehicles Regulation [10]. In this near zero-speed path-following task, once the transients have settled after the initial application of steer, the vehicle will enter a steady turn and follow a circular path on a constant radius. As shown in Fig. 2 (a), in the simulation, the centre of tractor front axles is required to travel along a section of straight path first and then approach to a circular arc with a radius of 11.25 m. The vehicle forward speed is less than 4 km/h. The step steer input follows the schedule illustrated in Fig. 2(b).

The maximum steer angle $\delta_{\max}$ is determined in such a way that the centre of tractor front axle follows a circular path with its radius approaching 11.25m.

OPTIMAL CONTROLLER DESIGN

As shown in Fig. 1, the front axle of the full trailer is steerable and the steering angle δ_r is determined by the

optimal controller based on linear quadratic regulator (LQR) theory [15].

The design criterion of the controller is to minimize the tractor/full-trailer system's RWA ratio at high speeds and path-following off-tracking at low speeds. The LQR controller design is an optimization problem: minimize the performance index

$$J = \int_0^{\infty} [q_1 (\dot{V}_1 + U_1 \omega_1)^2 + q_2 (\dot{V}_2 + U_2 \omega_2)^2 + q_3 \delta_r^2] dt \quad (7)$$

subject to equation (3). By solving the algebraic Riccati equation, the solution of the optimization problem is the control vector of the form

$$u = -Kx \quad (8)$$

where K is the control gain matrix with a dimension of 1×4 , x and u are the state and control variable vectors defined by equations (4) and (5), respectively. In equation (7), q_1 , q_2 and q_3 are weighting factors that impose penalties upon the magnitude and duration of the lateral acceleration of the tractor at the centre of gravity (CG), $\dot{V}_1 + U_1 \omega_1$, the lateral acceleration of the trailer at CG, $\dot{V}_2 + U_2 \omega_2$, and the active steering angle, δ_r , respectively. Note that the third term on the right side of equation (7) represents the energy consumption of the active trailer steering system.

PROPOSED INTEGRATED DESIGN METHOD

In this section, an optimization method for determining the LQR controller parameters is introduced. Then, the proposed design synthesis approach is described.

METHOD FOR DETERMINING LQR CONTROLLER WEIGHT FACTORS

With the LQR controller structure designed in section 3, the next step is to determine the controller parameters, i.e. the

weighting factor vector $[q_1 \ q_2 \ q_3]$ as shown in equation (7). After this vector is determined, the control gain matrix K indicated in equation (8) can be achieved following the procedure described in the previous section. To identify the weighting factor vector, trial and error approaches are commonly used and this is a time-consuming and tedious process [12]. To facilitate the identification of the controller parameters, an optimization method is proposed in the current research. In this subsection, this method is explained through

the determination of the weight factors for the controller of the active trailer steering system.

The LQR controller has two operational control modes, i.e. high-speed RWA control mode and low-speed Off-tracking control mode. The design objective of the RWA control mode is to minimize the RWA ratio in the single lane change maneuver described in subsection MANEUVERS EMULATED. The design criterion of the Off-tracking control mode is to minimize the off-tracking in the circle path-following maneuver.

For the RWA control mode design, the performance index corresponding to that indicated in [equation \(7\)](#) takes the following form

$$J_{RWA} = \int_0^{\infty} [q_{RWA1}(\dot{V}_1 + U_1\omega_1)^2 + q_{RWA2}(\dot{V}_2 + U_2\omega_2)^2 + q_{RWA3}\delta_r^2] dt \quad (9)$$

[Fig. 2](#) shows the procedure to determine the vector $[q_{RWA1} \quad q_{RWA2} \quad q_{RWA3}]$. The genetic algorithm (GA) [17] sends the randomly selected weighting factor vector to the 3 DOF vehicle model and LQR controller constructed in Matlab. Based on the given weighting factor vector, the controller is updated and corresponding simulation is performed. Then a data processor calculates the performance measure of RWA for the single lane change maneuver based on the simulation results. This set of calculated RWA ratio is used as fitness function values. At this point, if the convergence criteria are satisfied, the optimization terminates. Otherwise, the fitness function values are sent back to the GA. Based on the returned fitness values, the GA produces the next generation of design variable sets using genetic operators, e.g. selection, crossover, and mutation. This procedure repeats until the optimized weighting factor vector is found.

Following the same procedure shown in [Fig. 2](#), the weighting factor vector for the low-speed Off-tracking mode, $[q_{OFF1} \quad q_{OFF2} \quad q_{OFF3}]$, can also be determined.

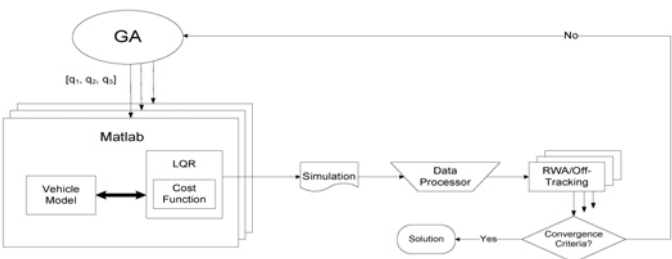


Fig. 2. The optimization method for determining LQR controller weighting factors

PROPOSED INTEGRATED DESIGN METHOD

For the purpose of simplicity, the proposed design method is described through the design optimization of the active trailer steering system for the tractor/full-trailer combination.

As introduced in the above subsection, the RWA and Off-tracking modes of the controller are designed independently and the vehicle system parameters (i.e. the passive design variables including vehicle geometry and inertia parameters) take their nominal values. In order to improve the overall performance of the vehicle system, in the following design synthesis, the two control modes are integrated and the passive design variables $\mathbf{X}$ are optimized. Since the control gain matrix $\mathbf{K}$ (i.e. the active design variables) for each of the two control modes is dependent on $\mathbf{X}$, through the design synthesis, the passive and active design variables are optimized simultaneously.

The following utility function is used as the objective function for the design optimization.

$$obj = \frac{1}{\rho_{rwa}} \times F_{rwa}(\mathbf{X}) + \frac{1}{\rho_{off}} \times F_{off}(\mathbf{X}) \quad (10)$$

where ρ_{rwa} and ρ_{off} are the performance measures when $\mathbf{X}$ takes its nominal value. These performance measures are achieved using the numerical simulations based on the high-speed single lane change maneuver and the low-speed circle path-following maneuver, respectively. $F_{rwa}(\mathbf{X})$ and $F_{off}(\mathbf{X})$ are the corresponding performance measures with the current design variable set $\mathbf{X}$.

This design synthesis is implemented as shown in [Fig. 3](#). First, a population of n sets of design variables is randomly selected by the GA; the corresponding sets of design variables are sent forward in parallel to the Matlab platform, where with a given set of design variables $\mathbf{X}$, the vehicle system model and LQR controller are updated. Then, the vehicle model simulates to travel in both the high-speed single lane change maneuver and the low-speed circle path-following maneuver, respectively. In these two maneuvers, the LQR controller operates in its high-speed RWA control mode and low-speed Off-tracking control mode, accordingly. The corresponding data processor determines the resulting performance measure, RWA representing $F_{rwa}(\mathbf{X})$ and Off-tracking denoting $F_{off}(\mathbf{X})$. These performance measures are sent to the utility function shown in [equation \(10\)](#). With this utility function, the resulting fitness value

vector $[obj_1 \quad obj_2 \quad \dots \quad obj_n]$ can be achieved. At this point, if the convergence criteria are satisfied, the calculation stops; otherwise, this vector is sent back to the GA. Based on the returned fitness values corresponding to the given sets of design variables, the GA produces the next generation of design variable sets using selection, crossover, and mutation operators. This procedure repeats until the optimized variable set is found.

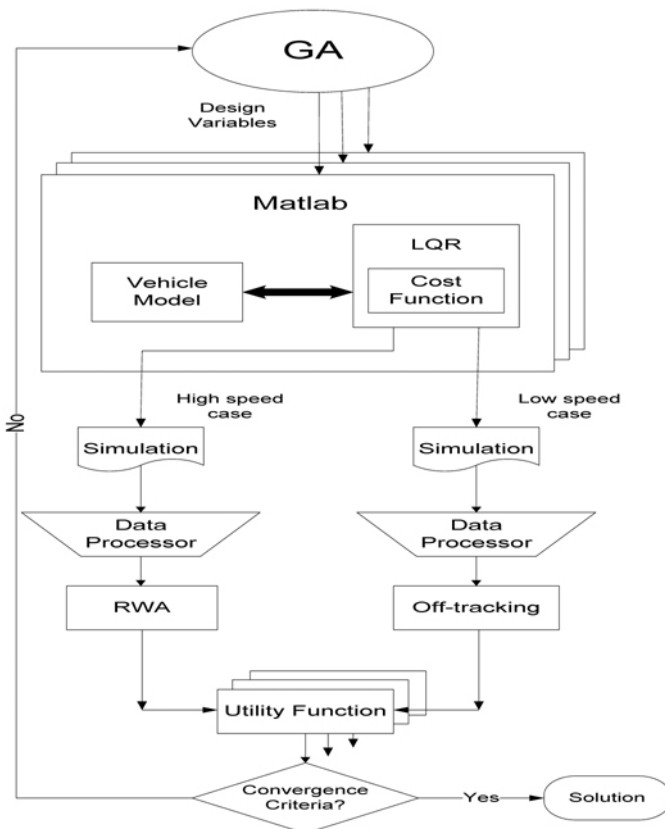


Fig. 3. Schematic representation of the proposed design synthesis approach

RESULTS AND DISCUSSION

In this section, the simulation results derived from the following cases are presented and discussed: 1) baseline case, where the design variables $\mathbf{X}$ take the nominal values and the vehicle has no active trailer steering system; 2) control case, where the design variables $\mathbf{X}$ and control gain matrices $\mathbf{K}$ take their nominal values; 3) optimal case, where the design variables $\mathbf{X}$ and control gain matrices $\mathbf{K}$ take their optimized values.

SIMULATION RESULTS OF BASELING CASE

In the case of numerical simulations for the baseline vehicle with non-steerable trailer axles, the vehicle system parameters take their nominal values listed in Table C1 of Appendix C. The simulation results for the lane change maneuver are shown in Fig. 4. As shown in equation (6), the amplitude of sinusoid steer input, A , takes the value of 1.1×10^{-2} . From Fig. 4(a), it can be seen that compared with the 2.5 second time period of the sinusoid steer input, the vehicle takes a longer time (around 7 seconds) to reach a steady state. This indicates that the vehicle system is stable, but external perturbations will take a long time to decay. The RWA ratio is 1.2. Fig. 4(b) shows the trajectory of tractor front axle centre and that of trailer rear axle centre. It is observed that before the trailer reaches the steady state, there is a small overshoot.