LATERAL CONTROL OF AN A-DOUBLE COMBINATION VEHICLE CONSIDERING DIFFERENT MEASUREMENT SIGNALS AND DOLLY STEERING CONFIGURATIONS




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Abstract

The objective of this paper is to compare and study the performance of different steering configuration of an active dolly which is employed in an A-double combination vehicle (tractor-semitrailer-dolly-semitrailer) to enhance high speed stability and performance of the vehicle. Three dolly steering configurations are considered to be investigated; the front axle steering (FAS), rear axle steering (RAS) and double axle steering (DAS). A static output feedback control synthesis is used based on linear matrix inequality (LMI) and $\mathcal{H}_\infty$ control. The simulation results show significant effects of the active dolly steering on improving lateral stability and performance at high speeds. The effectiveness of approaches are demonstrated using a high-fidelity vehicle model. The simulation results indicate that the FAS and DAS configurations are preferable compared to the RAS configuration at high speeds.

Keywords: Static output feedback , LMI based $\mathcal{H}_\infty$ synthesis, Rearward amplification, lateral control, active dolly steering configurations.

1. Introduction

Nowadays, there is a growing interest in the use of longer and heavier combination vehicles (LHCVs) than current conventional heavy vehicles due to economic and environmental considerations. However, there is a major concern regarding their poor manoeuvrability at low speeds and instability at high speeds. Therefore, there is a crucial need to improve the performance of LHCVs in a way to ensure that they comply with different safety and maneuverability regulations. The topic of improving the low speed maneuverability and high speed stability of LHCVs has been intensively researched. The opportunity to improve the performance with employing active steering systems on the trailers has been investigated by many research communities worldwide (e.g. Cheng and Cebon, 2009; Kural et.al ,2017; Oreh, 2013; Kim et. al, 2016; Coleman, 2002). The main focus of this paper is on the high speed stability and performance of an A-double combination vehicle equipped with an active dolly with steerable axles as shown in figure 1.

In many practical control applications, all system state variables might not be available and easily measurable for feedback. In such cases, one would need to consider a synthesis based on an output feedback. In general, there are two types of output feedback, static and dynamic output feedback. The design of dynamic output feedback (DOFB) leads to high order controllers that might bring more complications to the system dynamics, especially in the case of uncertain systems. The use of the DOFB controllers might not be practical in industry due to complicated implementation and potential problems in maintenance. On the other hand, the static output feedback (SOFB) is conceptually the simplest output feedback controller since it concerns finding a static gain to achieve the desirable system characteristics. Furthermore, the SOFB controllers are very simple to implement in practice and more reliable because they do not need the state estimators and computer processors used to implement DOFB controllers. Therefore in this paper, a SOFB control synthesis is considered to enhance the lateral performance of the A-double combination.

The objective of the lateral controller is to suppress undesired yaw rate rearward amplification in the towed units by active steering of the dolly at high speeds, of course provided that it is possible to stabilize the system in this fashion. The synthesis of the controller is based on linear matrix inequality (LMI) conditions that ensure stability as well as desired performance objectives. The LMI method is a novel technique in solving optimization problems and is used to provide computationally efficient controller design techniques for a variety of linear control problems (Scherer and Weiland, 2005). In order to determine a suitable measurement signal for feedback, different set of measurements are assumed to be available, for instance one of the three articulation angles or a mixture of them.

In addition, three different dolly steering configurations are investigated, front axle steering (FAS), rear axle steering (RAS) and double-axle steering (DAS). In the case of DAS, both dolly axles are assumed to be steered with the same steering angle. The potential performance benefits of the selected steering configurations and controllers are then evaluated using a high fidelity vehicle model, Volvo Transport Model (VTM), developed and validated by Volvo Group Trucks Technology (Sundström and Laine, 2012).

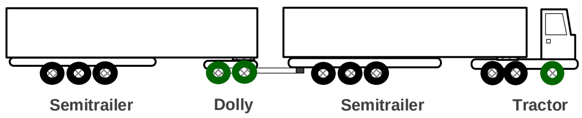


Figure 1 – A-double combination vehicle

2. Controller synthesis

This paper is concerned with the attenuation of yaw rate amplification in the last semi-trailer of the A-double combination. The controller is designed based on a linear vehicle model without accounting for load transfer and the roll dynamics. It is also assumed that steering and articulation angles are small. The total weight and length of the considered A-double combination are about 80 tones and 32 m, respectively. The dynamics of the A-double combination is described by the following state-space realization

$$\Sigma \begin{cases} \dot{x}(t) = A\ x(t) + H\ \delta_{driver}(t) + B\ \delta_{dolly}(t), \\ z(t) = C\ x(t) + G\ \delta_{driver}(t) + D\ \delta_{dolly}(t), \\ y(t) = S\ x(t) + R\ \delta_{driver}(t)\ , \end{cases} \tag{1}$$

where $x \in \mathbb{R}^{n_x}$ represents the state vector, $\delta_{driver} \in \mathbb{R}^{n_d}$ is the external and disturbance inputs acting on the system and here representing the driver steering input, and $y \in \mathbb{R}^{n_y}$ is the measured output vector. The control input of the system is the signal $\delta_{dolly} \in \mathbb{R}^{n_u}$ which is the dolly steering angle to be designed. The steering input δ_{dolly} can be δ_{31} (steering angle on the front axle of the dolly), δ_{32} (steering angle on the rear axle of the dolly) or both depending on the dolly steering configuration. The performance of the system will be assessed based on the performance output $z \in \mathbb{R}^{n_z}$ obtained in response to δ_{driver}. In this linear model, the state vector is given as

$$x = [\theta_1\ \theta_2\ \theta_3\ v_{y1}\ \omega_{z1}\ \dot{\theta}_1\ \dot{\theta}_2\ \dot{\theta}_3]^T, \tag{2}$$

which has eight components identified according to the following. θ_1 is the articulation angle between the tractor and the first semitrailer, θ_2 is the articulation angle between the first semitrailer and the dolly and θ_3 is the articulation angle between the dolly and the last semitrailer. The signal v_{y1} is the lateral velocity of the tractor and ω_{z1} is the yaw rate of the tractor. The vehicle parameters used in the linear vehicle model and the derivations of the system matrices A, H and B can be found in (Nilsson and Tagesson, 2013).

The aim of the considered SOFB control synthesis is to stabilize the system and satisfy desired performance requirements by finding a gain matrix $K_{fb} \in \mathbb{R}^{n_u \times n_y}$ with which the control input δ_{dolly} is generated as follows:

$$\delta_{dolly}(t) = K_{fb} y(t). \tag{3}$$

The controller gain K_{fb} is found based an $\mathcal{H}_\infty$ synthesis to ensure bounds on the energy gain from the external input to the performance output of the system. The SOFB design based on the LMI-based $\mathcal{H}_\infty$ technique is adapted from (Köroğlu and Falcone, 2014). To formulate the $\mathcal{H}_\infty$ synthesis problem for the system with the state-space description in (1), one needs to find a gain vector K_{fb} such that the closed-loop system is stable and the performance output z is ensured to satisfy the following condition for the external signal $\delta_{driver}(.)$ with $0 < \|\delta_{driver}(t)\|_2 \triangleq \sqrt{\int_0^\infty \delta_{driver}(t)^T \delta_{driver}(t) dt} < \infty$ when $x(0) = 0$:

$$\|z(t)\|_2 < \gamma \|\delta_{driver}(t)\|_2, \tag{4}$$

where γ is the $\mathcal{H}_\infty$-gain performance level that is desired to be minimized. The LMI condition $\mathcal{N}$ is then defined in terms of the matrix variables $Y = Y^T \in \mathbb{R}^{n_x \times n_x} \succ 0$, $W \in \mathbb{R}^{n_y \times n_y}$, $N \in \mathbb{R}^{n_u \times n_y}$ as

$$\mathcal{N}(\phi) = \mathsf{He}\begin{bmatrix} -\phi W & \phi(SY - WS) & \phi R & 0 \\ BN & AY + BNS & H & 0 \\ 0 & 0 & -\frac{\gamma}{2}I & 0 \\ DN & CY + DNS & G & -\frac{\gamma}{2}I \end{bmatrix} \prec 0, \tag{5}$$

where $\mathsf{He}\,\mathcal{N} \triangleq \mathcal{N} + \mathcal{N}^T$ and ϕ is an arbitrary (yet fixed) positive scalar considered by the designer. The matrix variable W is assumed to be non-singular. To summarize, the static output feedback gain is calculated by solving the following optimization problem

$$\begin{gathered} \text{minimize } \gamma \\ \text{subject to } \mathcal{N} \prec 0,\ \phi \succ 0,\ Y = Y^T \succ 0,\ (N,W) \in \mathcal{M}, \\ \text{given } (A, H, B, C, G, D, S, R) \end{gathered} \tag{6}$$

where $\mathcal{M}$ is a set that contains all the pairs of the matrix variables (N,W) for which there exists a matrix gain K_{fb} associated with γ_{min} such that

$$K_{fb} = NW^{-1}. \tag{7}$$

In the next section, the synthesis procedure is applied to the A-double combination and thereafter the simulation results are provided to validate the controller.

3. Synthesis Results and Simulations

In this section, the lateral control of the A-double combination is considered at high speeds, with the intention to apply the synthesis method developed in the previous section. In order to have a better comparison between different configurations, a simple performance objective is chosen in which the only concern will be the minimization of the yaw rate of the last semitrailer. Thus the $\mathcal{H}_\infty$ constraint of (4) is imposed on the performance output

$$z(t) = \omega_{z4}(t), \tag{8}$$

where $\omega_{z4} = \omega_{z1} + \dot{\theta}_1 + \dot{\theta}_2 + \dot{\theta}_3$. In this fashion, the yaw rate of the last semitrailer is suppressed to attenuate undesired yaw rate amplifications of this unit caused by evasive maneuvers. With this choice of the performance output, the matrices C, G and D in (1) are hence identified as follows: $C = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 & 1 & 1 & 1 \end{bmatrix}$, $G = 0$ and $D = 0$.

Otherwise, one can also include some more signals of interest in the performance objective to be minimized such as the yaw rate of the dolly or the steering angle of the dolly in order to limit the utilized steering angle on the dolly axles (Kati et. al, 2016).

In this study, different alternatives of the measurement output y are considered. The matrices S and R in (1) are determined based on the available measurements. For instance in the case of the second articulation angle available (i.e. $y = \theta_2$), the matrices S and R are chosen as $S = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$ and $R = 0$.

In order to solve the optimization problem and find the minimum achievable value of γ in (4), one needs to perform the LMI condition of (5) for each fixed value of ϕ over a chosen grid. The optimizations are performed in MATLAB by using Yalmip (Löfberg, 2004) together with SeDuMi solver (Strum, 1998). The required dolly steering angle is then simply calculated by multiplying the obtained control gain of (7) with the measured signals y as in (3).

The minimum γ levels, γ_{min} , obtained for various control designs are listed in Table 1. Here the γ_{min} is the minimal achievable $\mathcal{H}_\infty$ norm of the transfer function $\mathcal{T}_{zd}$ from δ_{driver} to ω_{z4}. In fact, γ_{min} is a uniform bound over all frequencies on the transfer function $\mathcal{T}_{zd}$. As can be seen, the γ values are decreased in all controlled cases, if compared to the passive vehicle with $\gamma = 5.7621$. It is observed that the syntheses based on the DAS and FAS configurations result in the smaller values of γ in the case of the first and second articulation angles available compared to the RAS configuration. While in the case of available measurement of all three articulation angles, the RAS configuration results in the lowest achievable γ level. It is also interesting to see that with increased number of the measurement outputs the γ level is pushed down more in all three configurations. Table 2 summarizes the obtained controller gains for various design cases.

Table 1 – Minimum value of γ (γ_{min}) for different steering configurations and controllers ($\gamma = 5.7621$ for the passive vehicle)

Measurements	FAS	RAS	DAS
$y = \theta_2$	3.9140	4.5088	3.5987
$y = [\theta_1,\theta_2]^T$	3.9088	4.1226	3.6005
$y = [\theta_1,\theta_2,\theta_3]^T$	3.8609	3.4017	3.5359

Table 2 – Control gain vectors for different steering configurations and controllers

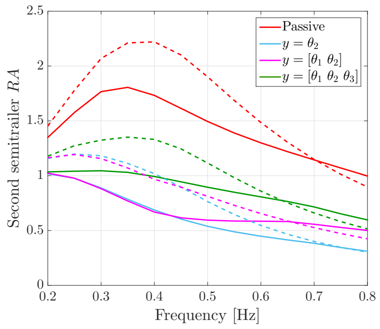
Measurements	FAS	RAS	DAS
$y = \theta_2$	-0.5604	-0.8604	-0.6692
$y = [\theta_1,\theta_2]^T$	[-0.0260 -0.9301]	[-0.0260 -0.9301]	[-0.1003 -0.6296]
$y = [\theta_1,\theta_2,\theta_3]^T$	[-1.3340 -2.3375 -2.2806]	[-1.3340 -2.3375 -2.2806]	[-0.7246 -0.8686 -0.6559]

To choose a suitable controller and dolly steering configuration from the practical point of view, the following objectives should be considered. First of all based on the chosen performance output, the controller should suppress the maximum yaw rate of the last semitrailer without imposing any rollover risk. Furthermore, it is desirable and more cost effective to achieve the desired performance with a minimum required control effort which is the amount of steering on the dolly axles. In the end, the candidate controller should allow easy implementation with a minimum number of required sensors and actuators.

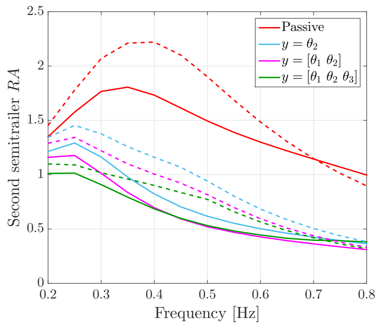
In the next section, different lateral performance measures have been considered to evaluate the effectiveness of the controllers; one time-domain and two frequency-domain approaches. The time-domain approach is based on a single sine-wave steering input of the driver (Aurell and Winkler, 1995). The first frequency-domain approach is based on a random steering input (Aurell and Winkler, 1995) and the other one is based on a chirp steering input (Zhu and He, 2015).

3.1 Time domain analysis

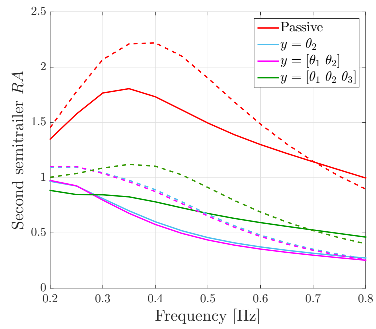
In order to evaluate the effectiveness of the controllers in the time domain, several single sine-wave steering inputs with specific amplitudes and frequencies are applied to the VTM vehicle model at a velocity of 80 km/h. To have a better comparison, a peak lateral acceleration of 1.5 m/s^{-2} on the first axle of the tractor is prescribed for all single sine-wave manoeuvres, below called SLC, because these are close to a single lane change maneuver. To achieve the required level of the lateral acceleration in the first axle, the amplitude of the driver steering input is adjusted accordingly. The reason behind the choice of this level of peak lateral acceleration is that to ensure the tire lateral forces remain in the linear region of the tire characteristics.



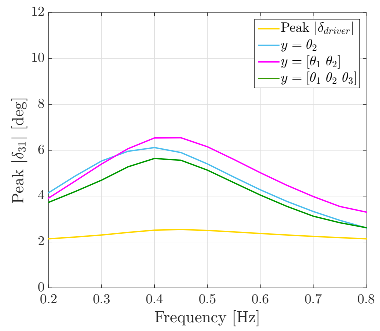
(a) $RA_{\omega z_4}$, FAS



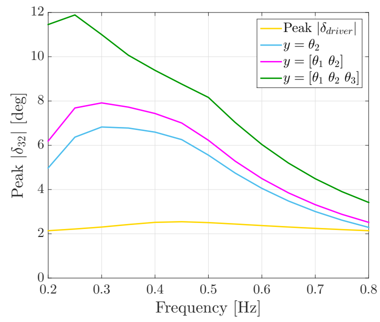
(b) $RA_{\omega z_4}$, RAS



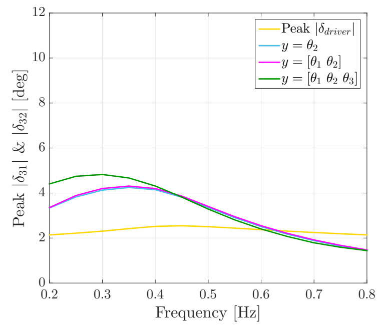
(c) $RA_{\omega z_4}$, DAS



(d) Peak of $|\delta_{31}|$, FAS



(e) Peak of $|\delta_{32}|$, RAS



(f) Peak of $|\delta_{31}|$ and $|\delta_{32}|$, DAS

Figure 2 – Second semitrailer RA and corresponding peak value of dolly steering angles applied to the VTM vehicle model ($v_x = 80\ km/h$, $ay_{11} = 1.5\ m/s^{-2}$); yaw rate RA (solid curves), lateral acceleration RA (dashed curves)

The most common performance measure to assess the high speed lateral performance is rearward amplification (RA). The performance measure RA is defined as the ratio of the maximum value of a motion variable of interest (yaw rate or lateral acceleration) for the worst excited following vehicle unit to that of the lead unit in an obstacle avoidance maneuver. In the time domain, the RA is mathematically defined as

$$RA_{\omega_{zi}} \triangleq \frac{||\omega_{zi}||_\infty}{||\omega_{z1}||_\infty} \quad \text{and} \quad RA_{a_{yi}} \triangleq \frac{||a_{yi}||_\infty}{||a_{y11}||_\infty},\ i = 2,3,4 \tag{9}$$

where $\|\cdot\|_\infty$ represents the $\mathcal{L}_\infty$-norm: $\|\omega_z\|_\infty \triangleq \sup_{t\geq 0} |\omega_z(t)|$. The signal ω_{z1} denotes the yaw rate of the tractor and the signal a_{y11} represents the lateral acceleration at the front axle of the tractor. The signals ω_{zi} and a_{yi} represent the yaw rates and lateral accelerations of the ith vehicle unit, respectively. The lateral acceleration of the towed units are measured at the center of gravity of each vehicle unit.

The obtained results for the yaw rate and lateral acceleration RA of the last semitrailer and the utilized dolly steering angle in the controlled vehicle versus the input frequency for different design cases are illustrated in Figure 2. The results show significant performance improvements are achieved in the controlled vehicle in all cases by the reduction of the yaw rate RA of the second semitrailer. As can be seen, in the FAS and DAS configurations, the yaw rate RA is suppressed by utilizing a lower maximum amplitude of the steering angle on the dolly axles compared to the RAS configuration. On the other hand, in the RAS configuration, the design based on the measurements of three articulation angles lead to a better performance with respect to suppression of the RA values, but in the cost of large steering angle on the rear axle.

It is observed that the control designs based on the information of the second articulation angle has shown an acceptable performance in terms of reduction of RA with an adequate level of steering effort for all three steering configurations. Since the second articulation angle can simply be measured reliably, thus the controller designed based on the measurement of this angle is chosen for the rest of the paper.

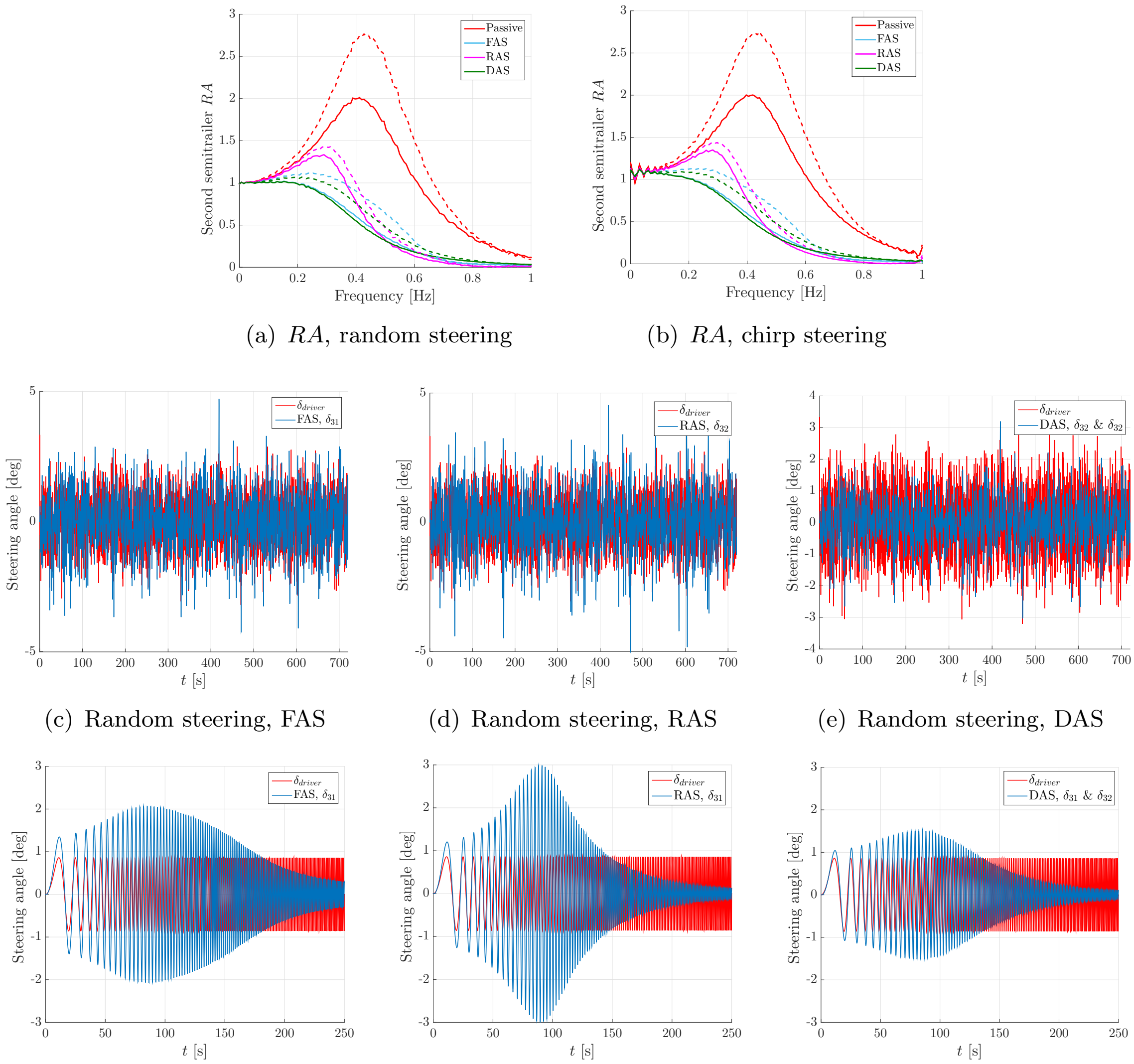
It should be emphasised that in the control synthesis, the upper bound of the transfer function from the driver steering input to the yaw rate of the last semitrailer is minimized while the yaw rate RA measure in the last semitrailer is the ratio between the yaw rates of the tractor and the last semitrailer. It is hence important to stress that by applying the $\mathcal{H}_\infty$ controller, the yaw rate RA is indirectly reduced. Therefore, the results obtained in this section might not be highly consistent with the results obtained in the previous section.

3.2 Frequency domain analysis

In the frequency domain approaches, the driver is simply applying a random steering input to the vehicle. The intent of the random driver steering is to excite the lateral dynamics of the vehicle in a wide range of steering frequencies. The frequency-domain approaches are obtained by using the time-domain simulations in the VTM vehicle model. In the first approach, the RA measures are achieved by using a random steering input in which both the amplitude and frequency of the driver steering input are varied randomly and continuously for at least 12 min (Fancher and Winkler, 1992; Aurell and Winkler, 1995). In the second approach, the RA values are measured when a swept-sine (chirp) steering input as in (Zhu and He, 2015) is applied as the driver steering input. The amplitude of the chirp steering input is chosen as 0.86° that sweeps between the frequencies of 0 Hz and 1 Hz for 250 s. The values of the RA for the second semitrailer are then obtained by calculating Fast Fourier transform (FFT) of the ratio between the yaw rates or lateral acceleration of the tractor and the second semitrailer.

The frequency-domain calculations of the yaw rate and lateral acceleration RA for the second semitrailer and also applied steering angels are shown in Figure 3. It is interesting to observe that the RA curves in both frequency domain approaches are very similar indicating that there is a great agreement between the random steering and chirp steering approaches. As shown the effectiveness of the controllers are verified on various design cases, which show significant reduction in the yaw rate RA and also reduction in lateral acceleration RA, as a byproduct. Consequently, the rollover risk due to high lateral acceleration is decreased by the controllers. As can be seen, the DAS configuration has shown the best performance and the RAS configuration has shown the worst performance in terms of the RA reduction with a minimum amount of utilized dolly steering angle.

A comparison between the results illustrated in Figure 2 and Figure 3 reveals that the RA values obtained in the time-domain approach are smaller than the ones obtained in the frequency-domain approaches. This observation implies that the obtained results from the time- and frequency-domain approaches might not be identical but both approaches are a function of the input frequency. It should also be noted that the time-domain and frequency-domain approaches are highly correlated and comparable but there is not a mathematical relation between these approaches (Luijten et. al, 2012).



(a) RA, random steering

(b) RA, chirp steering

(c) Random steering, FAS

(d) Random steering, RAS

(e) Random steering, DAS

(f) Chirp steering, FAS

(g) Chirp steering, RAS

(h) Chirp steering, DAS

Figure 3 – Frequency domain analysis with corresponding steering angles applied to the VTM vehicle model; yaw rate RA (solid curves), lateral acceleration RA (dashed curves)

3.3 Time domain simulations

In this section, the time responses of the controlled and uncontrolled vehicles are illustrated in a SLC maneuver. The SLC maneuver is simulated using the VTM vehicle model at a longitudinal velocity of 80 km/h. The frequency of the driver steering is chosen as 0.35 Hz in which the largest peak of the yaw rate RA occurred in the passive vehicle as shown in Figure 2. The amplitude of the sine-wave steering input is adjusted in a way that a lateral acceleration of 1.5 m/s^{-2} occurs in the front axle of the tractor. The high speed transient offtracking (HSTO) is also considered as another high-speed performance measure and is defined as the maximum lateral deviation between the path of the front axle of the tractor and the path of the rearmost axle of the towed units.

The yaw rates and lateral accelerations of the vehicle units during the performed SLC maneuver are illustrated in Figure 4 and 5 for the passive vehicle and different controlled vehicles. As can be seen in the passive vehicle, the yaw and lateral motions get amplified at the towed units which causes large values of the yaw rate and lateral acceleration RA.

As a result the side slip angles of the towed units increase which in return lead to an increased HSTO as can be seen in the passive vehicle in Figure 6. The steering angles applied on the dolly axles are also shown in Figure 7.

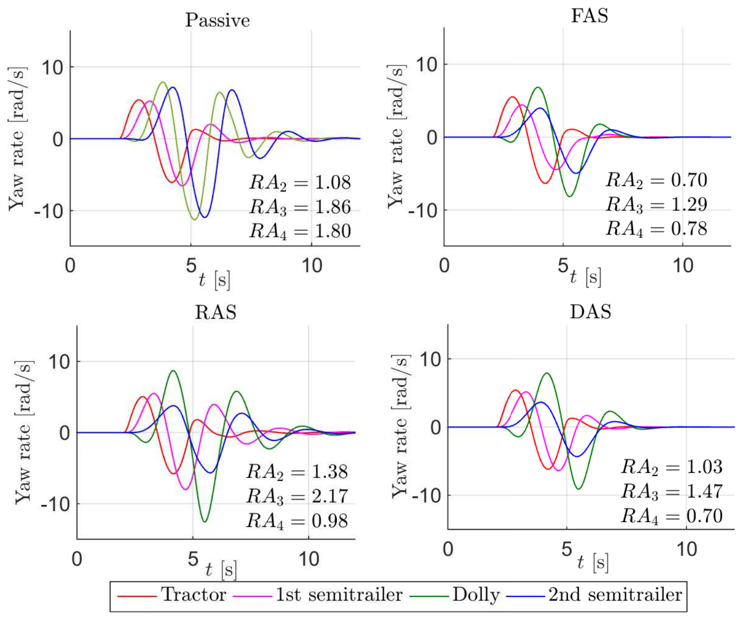


Figure 4 – Yaw rates of the A-double units for SLC (v_x = 80 km/h, a_{y11} = 1.5 m/s^{-2}, f =0.35 Hz)

Considering the dynamic response of the last semitrailer in the passive vehicle, the yaw rate RA is decreased from 1.8 to 0.70, 0.78 and 0.98 in the DAS, FAS and RAS configurations, respectively. On the other hand, the yaw rate of the dolly in the passive vehicle is changed from 1.86 to 1.47, 1.29 and 2.17 in the DAS, FAS and RAS configurations, respectively. This indicates that a larger corrective yaw motion in the dolly is required in the RAS configuration compared to two other configurations. It is also observed that the FAS and RAS configurations influence the dynamic response of the first semitrailer which implies that a force transmission is occurred between the dolly and the first semitrailer via drawbar. In the FAS configuration, the transferred force results in the reduction of RA values of the first semitrailer while in the RAS configuration the RA values are increased significantly. As a result of this, the first semitrailer axles experience less offtracking in the FAS configuration and more offtracking in the RAS configuration as shown in Figure 6. The DAS configuration has shown to have the least impact on the first semitrailer.

It is also observed that the achieved improvement in the lateral acceleration of the dolly and the last semitrailer is larger in the DAS configuration in comparison with two other configurations. Although, both the FAS and DAS configurations have shown a reasonable lateral performance improvement if compared to the RAS configuration. In general, the obtained results indicate that the DAS configuration has lead to a better lateral performance considering the following objectives and measures; the yaw rate and lateral acceleration RA, the HSTO, the minimum use of the dolly steering angle and less influence on the leading units.

Note that the previous results are based on the simulated maneuvers on the high friction surface with a friction coefficient of $\mu = 0.8$. A summary of the previously presented results and also new results associated to the SLC maneuver performed on the low friction surface with $\mu = 0.3$ are provided in Table 3. As expected for the passive vehicle the situation

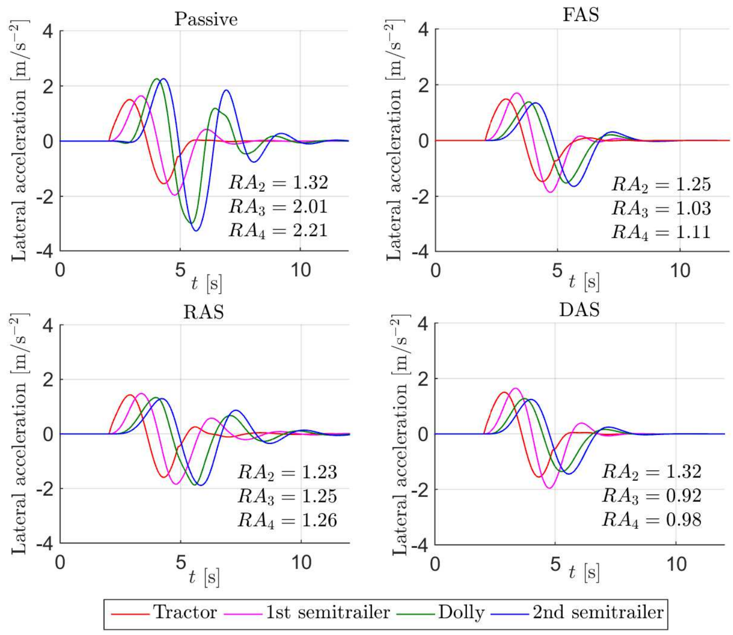


Figure 5 – Lateral accelerations of the A-double units, for SLC (v_x = 80 km/h, a_{y11} = 1.5 m/s^{-2}, f =0.35 Hz)

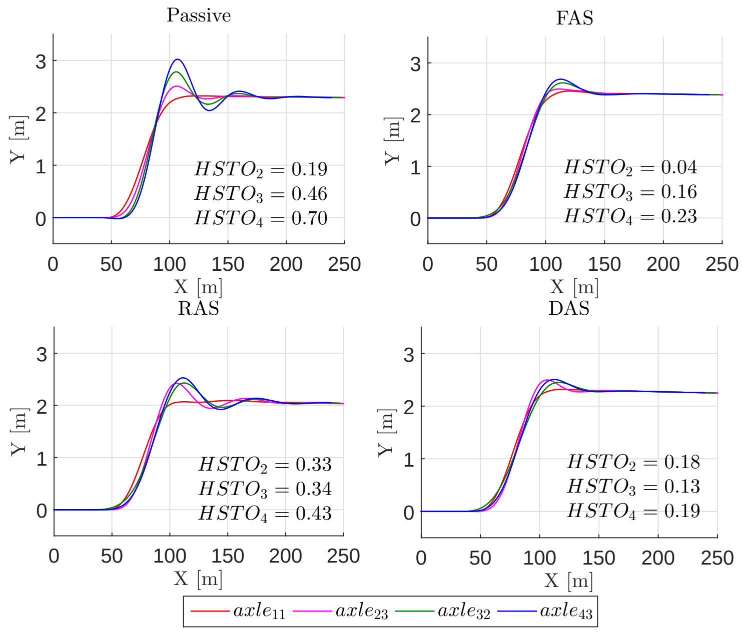


Figure 6 – Trajectories of the first axle of tractor ($axle_{11}$) and the last axle of the towed units ($axle_{23}$, $axle_{32}$, $axle_{43}$)

is more critical on the low surface friction, for which both yaw rate RA and HSTO are larger. In fact, there is a significant correlation between all three lateral performance measures but the yaw rate RA and HSTO have a stronger relationship especially in low friction.

As a matter of fact, there are some cases that the use of the lateral acceleration RA as a stability criterion is very misleading, for instance when the friction between the tires and the road are reduced. In such case, the yaw rate RA will increases significantly while this increase cannot be seen in the lateral acceleration RA. As shown in Table 3, the lateral acceleration RA of the passive vehicle on the low friction surface is decreased while the yaw rate RA is increased. This is largely due to the fact that on the low friction surface, the generation of the tire lateral forces in the towed units is reduced because of the limited friction availability.

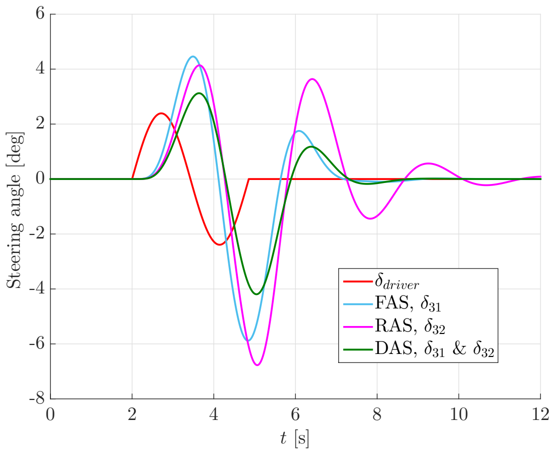


Figure 7 – Applied driver and dolly steering angles

Table 3 – Performance results in SLC ($v_x = 80\ km/h$, $a_{y11} = 1.5\ m/s^{-2}$, f =0.35 Hz)

SLC on a high friction surface (i.e. $\mu = 0.8$)									
	$RA_{\omega z_2}$	$RA_{\omega z_3}$	$RA_{\omega z_4}$	RA_{ay_2}	RA_{ay_3}	RA_{ay_4}	$HSTO_4$ [m]	max $\lvert\delta_{31}\rvert$ [deg]	max $\lvert\delta_{32}\rvert$ [deg]
Passive	1.08	1.86	1.80	1.32	2.01	2.21	0.70	–	–
FAS	0.70	1.29	0.78	1.25	1.03	1.11	0.23	5.88	–
RAS	1.38	2.17	0.98	1.23	1.25	1.26	0.43	–	6.77
DAS	1.03	1.47	0.70	1.32	0.92	0.98	0.19	4.19	4.19
SLC on a low friction surface with (i.e. $\mu = 0.3$)									
Passive	1.14	2.56	2.60	1.31	1.97	1.85	1.26	–	–
FAS	0.77	1.64	1.05	1.30	1.29	1.34	0.48	7.37	–
RAS	1.48	3.45	1.20	1.17	1.70	1.31	0.73	–	8.91
DAS	1.08	1.69	0.72	1.33	0.95	1.00	0.24	4.74	4.74

4. Conclusions

In this paper, a static output feedback synthesis is proposed for the lateral control of an A-double combination vehicle at high speeds. The proposed control strategy is easy-to-implement and does not require any observers to estimate the vehicle states. Several controllers are investigated based on a selected set of available measurements for feedback. The controller constructed based on only the information of the second articulation angle has shown acceptable performance improvement. From the simulations results, it could be concluded that the DAS and FAS configurations can improve the performance and stability of the A-double combination better in terms of reduced transient offtracking and rearward amplification, if compared to the RAS configuration. The results from the DAS configuration has also shown that the performance improvement is achieved with less dolly steering effort and almost no influence on the first semitrailer in comparison with two other configurations. However, this configuration may be more expensive than single steered axle configurations. It would be interesting to compare the performance of three dolly steering configurations at low speeds.

5. References

- P. Sundström, L. Laine (2012)," Validation of VTM model of tractor 4×2 with semitrailer using winter test results from Arjeplog 2011w11 P2685", Engineering report ER-624557.
- J. Aurell and C.B. Winkler (1995),"Standard Test Procedures for the Lateral Stability of Heavy Vehicle Combinations", Road Transport Technology 4, UMTRI, Ann Arbor.
- S. Zhu and Y. He (2015),"Articulated heavy vehicle lateral dynamic analysis using an automated frequency response measuring technique", International journal of vehicle performance, vol. 2, no. 1, pp. 30-57.
- C. Scherer, S. Weiland (2005)," Linear matrix inequalities in control", Lecture Notes for a course of the Dutch Institute of Systems and Control.
- H. Köroğlu, and P. Falcone (2014), "New LMI conditions for static output feedback synthesis with multiple performance objectives", In Proc. 53rd IEEE Conference on Decision and Control, Los Angeles, CA.
- P. Nilsson and K. Tagesson (2013),"Single-track models of an A-double heavy vehicle combination", Report 2013:08, Chalmers University of Technology.
- M. F. J. Luijten, I. J. M. Besselink, R. M. A. F. Verschuren and H. Nijmeijer (2012),"Analysis of the lateral dynamic behaviour of articulated commercial vehicles", Vehicle System Dynamics, 50:sup1, p.169-189.
- B. Coleman (2002),"Productivity opportunities with steerable axles", Delft, The Netherlands, 7th International Symposium on Heavy Vehicle Weights and Dimensions.
- C. Cheng and D. Cebon (2009),"Improving roll stability of articulated heavy vehicles using active semi-trailer steering", Vehicle System Dynamics, 46:sup1, 373-388.
- K. Kim, H. Guan, B.Wang, R. Guo, and F. Liang (2016),"Active steering control strategy for articulated vehicles", Frontiers of Information Technology and Electronic Engineering, vol. 17, no. 6, pp. 576-586.
- S. H. Oreh, R. Kazemi, and S. Azadi (2013),"Directional control of articulated heavy vehicles", SAE International Journal of Commercial Vehicles, vol. 6, no. 1, pp. 143-149.
- J. Löfberg (2004)," YALMIP : a toolbox for modeling and optimization in MATLAB", in 2004 IEEE International Conference on Robotics and Automation (IEEE Cat. No.04CH37508), pp. 284-289.
- J. F. Sturm (1998). Using SeDuMi 1.02,"A MATLAB toolbox for optimization over symmetric cones", vol. 11.
- M. S. Kati, H. Köroğlu, and J. Fredriksson (2016),"Robust lateral control of an A-double combination via $\mathcal{H}_\infty$ and generalized $H2$ static output feedback", 8th IFAC Symposium on Advances in Automotive Control AAC, vol. 49, no. 11, pp. 305-311.
- K. Kural and P. Hatzidimitris and N. Wouw and I. Besselink and H. Nijmeijer (2017),"Active Trailer Steering Control for High-Capacity Vehicle Combinations", IEEE Transactions on Intelligent Vehicles, vol. 2, pp 251-265.