# Countering the Destabilizing Effects of Shifted Loads through Pneumatic Suspension Design

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## Abstract

This paper proposes a novel approach to reduce the destabilizing impacts of the shifted loads of heavy trucks (due to improper loading or liquid slosh) by pneumatic suspension design. In this regard, the pneumatically balanced suspension with dual leveling valves is introduced, and its potential for the improvement of the body imbalance due to the shifted load is determined. The analysis is based on a multi-domain model that couples the suspension fluid dynamics, shifted-load impacts, and tractor-semitrailer dynamics. Truck dynamics is simulated using TruckSim, which is integrated with the pneumatic suspension model developed in AMESim. This yields a reasonable prediction of the effect of the suspension airflow dynamics on vehicle dynamics. Moreover, the ability of the pneumatic suspension to counteract the effects of two general shifted loads - static (rigid cargo) and dynamic (liquid) - is studied. The simulation results indicate that the dual-leveling-valve suspension results in a reduction in roll angle and roll rate of the vehicle body for both static and dynamic load shifting cases, as compared to the conventional single-leveling-valve suspension. A suppression to the liquid sloshing behavior is obtained by the truck with the dual-leveling-valve suspension. Furthermore, the co-simulation platform established in the study is useful for efficient and accurate analyses of the coupled shifted load-pneumatic suspension-vehicle system dynamics.

**Keywords**: shifted load; pneumatic suspension; heavy truck; TruckSim; roll stability; multi-domain model; liquid slosh;

## 1. Introduction

A load shift refers to an unintended change in the position of the commercial truck's cargo load during road transportation. This could be a sideway shift of solid or liquid cargo on ramps, curves, or sudden swerves, which is undesired. The unwanted load or goods shifting may result in serious mechanical problems and rollover accidents, which must be avoided. Some examples of load shifting during transportation are illustrated in Figure 1a-c. Different cargo materials cause different load shifting that exerts an uneven weight distribution and hence increases the stress on both axle and suspension components. This stress potentially increases the chances of break-down and functional failures of the components, which in turn leads to several other maintenance and safety issues [1]. Moreover, during a turn, the shifted load incurs higher load transfer as compared to an in-center load, resulting in the truck losing control with subsequent rollover [2, 3].

Figure 1. (a) Free rolling pipes [4]; (b) slippery bags shifting [4]; (c) liquid slosh occurrence

As compared with the solid cargo, the liquid is more difficult to handle because it cannot be tied down or locked and continues moving on the road. Liquid slosh elicited by a steering maneuver makes the vehicle body tilt more, thereby diminishing the roll stability. A number of past studies have been associated with the tank design to resist or prevent slosh. Baffles have been found to be effective in controlling and minimizing slosh formation [5]. A study conducted by Jung et al. [6] indicated the significance of baffle height in suppressing the sloshing, while Ibrahim et al. [7] found that the baffle with circular grooves is more favorable than that with rectangular slots. Similarly, some studies revealed the significance of tank cross-section in providing additional stability [8-10]. However, these efforts cannot completely reduce the load shifting, which is in fact inevitable. There are also a number of other reasons that influence and cause the load shifting, such as the breakdown and failure of load restraint devices during transportation. Therefore, it is important to consider the shifted loading in the suspension design [11, 12], but in a quite limited number of studies examining the performance of the suspension in transportation of shifted loads, especially for heavy trucks. This paper is aimed at introducing a novel design of a heavy truck pneumatic suspension capable of resisting static and dynamic load shifts to keep the vehicle balanced and to reduce the stability problems.

The rest of the paper is structured as follows. Section 2 gives some background knowledge on heavy truck air suspension systems and liquid slosh modeling. Section 3 introduces the development of the multi-domain model that couples shifted load-pneumatic suspension-vehicle dynamics by a co-simulation technique. In Section 4, the body roll responses of the truck with the proposed suspension are computed and compared with the conventional pneumatic suspension for a steering maneuver under solid shifted loads and liquid slosh. At last, a discussion of conclusions closes the paper.

## 2. Background

In the past decades, airsprings have been extensively used for heavy truck primary suspensions in place of steel springs due to their benefits of reduced weight, ability to adjust ride height, increased ride comfort, lower road damage, and reduced structurally-borne noise [13]. The airsprings are connected to an air supply through a leveling valve(s) to maintain a nearly constant ride height regardless of load changes. As shown in Figure 2, the leveling valve is generally mounted to the vehicle frame, while providing a lever arrangement attached to the bottom of the suspension in such a way to provide an auto-adjustment in height. If the suspension jounces (compresses) beyond the ride height, the lever arm rotates upward and opens the valve to supply additional air to the airsprings, thereby increasing internal pressure and returning the suspension to its appropriate riding level. Similarly, if the suspension rebounds (extends), the lever arm rotates in the opposite direction and activates a purge valve to exhaust air from the airsprings, therefore decreasing the internal pressure and lowering the suspension back to the specified ride height.

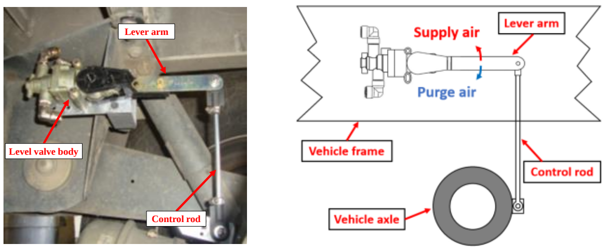
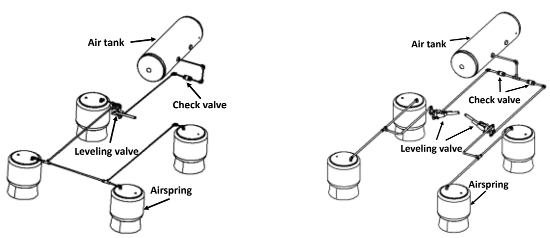



Figure 2. Installation and working process of the leveling valve

For the semi-trailer truck application, all axles except the steering axle are typically equipped with the pneumatic suspension system for improving ride comfort. The originally-equipped truck suspension is based on a single leveling valve (Figure 3a), which is less efficient in providing resistance to the body inclination. A novel suspension is recently proposed with two leveling valves and a symmetric plumbing arrangement (Figure 3b) to provide a balanced airflow and air pressure in the airsprings [14-16]. This dual leveling valve arrangement intends to contribute to a balanced force distribution among the axles that enables the suspension to maintain the body in the leveled position. This novel arrangement could be a good approach to meet the required performance in the transportation of shifted load. The objective of the paper is to provide a simulation evaluation of such arrangements on improving any dynamic imbalance that occurs because of the shifted load.




(a) (b)

Figure 3. Plumbing configurations of (a) single and (b) dual leveling valve suspensions on the tractor's tandem axles or trailer's axles

For this study, multi-domain modeling is required to couple the fluid dynamics of pneumatic suspensions and shifted-load impacts with the multi-body dynamics of a semi-truck [17]. In particular, two general categories of shifted load – static (rigid cargo) and dynamic (liquid) – are studied. In order to appraise the performance of suspensions in the presence of a constant uneven load, the static shifted load is considered fixed to the trailer floor. The other condition (dynamic liquid) includes the liquid slosh within a tank, which affects the roll dynamics of the trailer in a complex manner. A group of researchers has developed an analytical model for the prediction of the effect of fluid sloshing on the roll dynamics of a partially filled tank truck [8, 18-19]. Similarly, Azadi et al. [10] and Cheli et al. [20] simulated the dynamic liquid slosh using CFD techniques. Some limitations, such as extreme computational demands, are associated with the CFD approach, which makes it less suitable as compared with the quasi-static fluid slosh model [21]. The quasi-static fluid slosh model can help in predicting the steady-state motion of the liquid free-surface and roll response characteristics of the truck [9]. It is also a fact that the quasi-static slosh model yields a lower roll stability limit than that predicted by the rigid cargo model in the case of a partially-filled tank truck [22].

## 3. Model Description

In the current study, a model of a semi-trailer truck with a rigid shifted load is established by using TruckSim. TruckSim is a commercial software package that has been well recognized for providing accurate and realistic predictions of truck dynamic behavior. The truck model is then coupled with a pneumatic suspension model developed by Chen et al. [14-16]. The main intent of this coupling is to capture the contribution of the suspension pneumatic system on vehicle dynamics. To provide validity of the pneumatic suspension system model, a static tractor test is conducted. Additionally, the vehicle-pneumatic-suspension model is coupled with a roll-plane quasi-static slosh model, resulting in a pneumatic-suspension-integrated tank truck model, which has not established in any prior research. The models developed above are used to perform extensive simulations to explore the potential benefit of the suggested pneumatic suspension on alleviating the destabilizing effect of static (rigid) and dynamic (liquid) shifted loads. The following sections describe the development of truck models that are considered in the study, including (1) a 53-ft semi-trailer truck with a rigid off-center load, and (2) a 43-ft partially-filled tank truck.

### *3.1. 53-ft Semi-truck with Rigid Off-center Load*

As mentioned earlier, a 5-axle, 53-ft semi-trailer truck model with an off-center load is established in TruckSim software to investigate the effect of the suspension on counteracting the rigid shifted load. Parameters used for the truck simulation are determined from reference [16]. The off-center load (rigid shifted load) is assumed to be completely fixed to the trailer, thereby excluding the dynamic load shifting contribution. During the cornering, the off-center load results in the vehicle being subjected to additional overturning moment as compared with an in-center load, as shown in Figure 4.

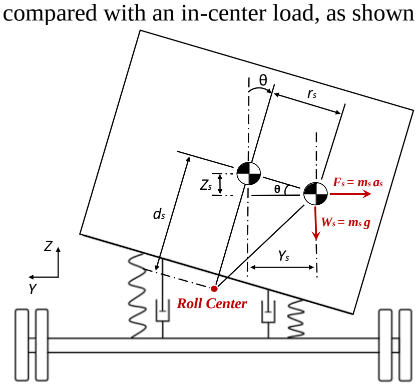



Figure 4. Roll-plane schematic of the truck with a rigid shifted load

The distance of the center of gravity (CG) between the off-center and in-center loads along the lateral ($Y_s$) and vertical ($Z_s$) axes can be expressed as:

$$Y_s = r_s cos|\theta| \tag{1}$$

$$Z_s = r_s sin|\theta| \tag{2}$$

where $\theta$ is the roll angle and $r_s$ is the lateral movement of the off-center load CG. The change in roll moment due to the off-center loading can be determined by:

$$\triangle M_{Fs} = -m_s a_s r_s sin|\theta| \quad \text{(due to the centrifugal force)} \tag{3}$$

$$\triangle M_{Ws} = m_s g r_s cos|\theta| \quad \text{(due to the load weight)} \tag{4}$$

where $m_s$ is the sprung mass. The lateral movement of the load makes the CG far away from the roll center, which increases the roll inertia of the cargo to the roll center. The axis theorem is then applied to help to calculate the roll inertia of the in-center and off-center loads to the roll center:

$$I_i = I_{i0} + {d_s}^2 m_s \quad \text{(in-center load)} \tag{5}$$

$$I_s = I_{s0} + ({d_s}^2 + {r_s}^2) m_s \quad \text{(off-center load)} \tag{6}$$

where $d_s$ is the height of the in-center load CG to the roll center, and $I_{i0}$ and $I_{s0}$ are the roll inertias of the in-center and off-center loads, which are considered to be identical in this study. The change in the roll inertia due to the lateral movement of the load can be obtained as:

$$\triangle I_x = (I_s - I_i) = m_s {r_s}^2 \tag{7}$$

Combining Equations (3), (4), and (7) yields the overturning moment caused by the off-center load relative to that of the in-center load:

$$M_{Tl} = \triangle M_{Ws} + \triangle M_{Fs} - \ddot{\theta} \triangle I_x$$

$$= m_s g r_s cos|\theta| - m_s a_s r_s sin|\theta| - \ddot{\theta} m_s {r_s}^2 \tag{8}$$

As indicated in Equation (8), the off-center load results in an overturning moment tilting the vehicle body, even under normal driving conditions, such as straight driving without the centrifugal force. During cornering conditions, an additional overturning moment may be produced due to the off-center load, depending on the direction of the body roll. If the body rolls to the side of the off-center load, an additional overturning moment is generated to promote the roll motion, as indicated in Equation (8), while in the opposite roll, the off-center load provides an anti-roll moment that resists the body roll. According to the equations above, The TruckSim model is developed to represent the roll moment contribution of the shifted load, as shown in Figure 5.

Figure 5. Back view of the truck model with shifted load in TruckSim

### *3.2. Pneumatic Suspension Model and Validation*

For a reasonable representation of the non-linear characteristics of the pneumatic suspension system, AMESim is used to develop the models of the pneumatic suspensions with single and dual leveling valves (Figure A1a and A1b of Appendix A). More details of the model development can be found in the previous study [16], which indicate a lack of validation for the pneumatic suspension system model. Therefore, a static tractor test is conducted for the validation of the pneumatic suspension system model. The test setup (Figure 6) includes a stack of steel plates (9450 lb) placed on the back of the tractor, where the fifth wheel is resident to simulate vertical trailer load. The airsprings on the tractor's tandem axles are connected by pipes and are controlled by one leveling valve set on the table, as shown in Figure 6. A precise rotation movement of the lever arm of the leveling valve can be controlled by a servomotor through a linkage arrangement. The linkage arrangement is completed by connecting the lever arm to the motor using two pieces of steel angle, as shown in Figure 6. The Hitec HS-5805MG servo motor (6V power supply) is used in the test due to its high torque capability (1.8 lb·ft), which is beyond the torque required to rotate the lever arm (about 1.2 lb·ft). The rotational speed for the server motor is 0.14s/60deg that is also beyond the requirement of this test.

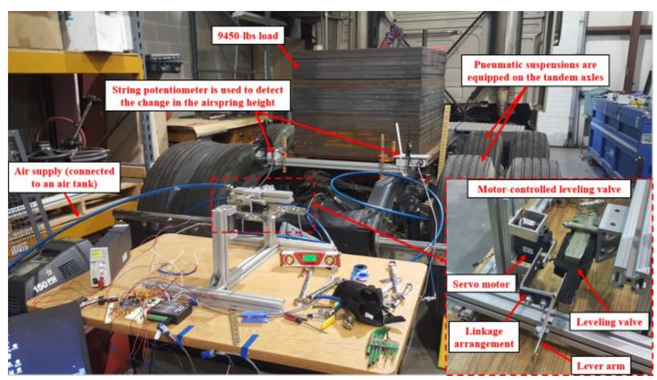


Figure 6. The pictorial view of a tractor static test setup

String potentiometer, a draw-wire sensor widely used to measure linear position and velocity by using a flexible cable, a spring-loaded spool, and a rotational sensor, is applied to capture the change in the airspring height during the test. The string potentiometer is fixed to the vehicle frame while providing its measuring cable connected to the vehicle axle, as shown in Figure 7. This setup results in the measuring cable capable of extending along with the relative change between the vehicle frame and the axle, which creates an electrical signal proportional to the change in the airspring height.

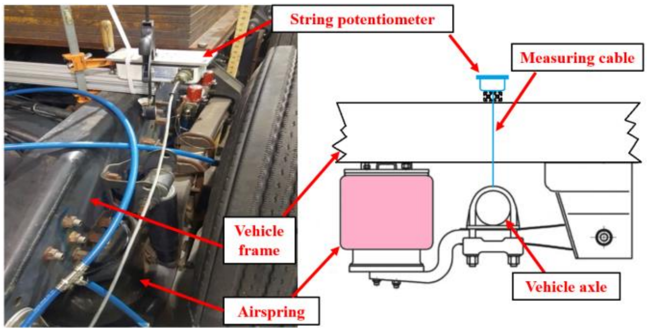
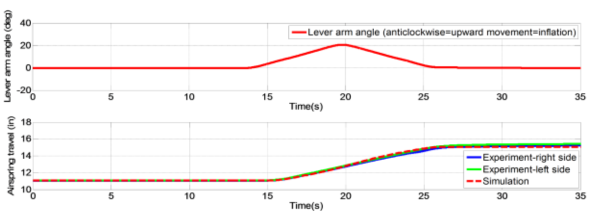
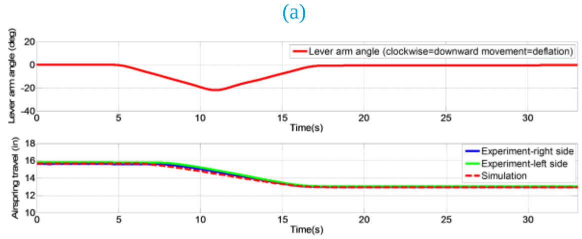



Figure 7. Installation of the string potentiometer

In this test, the lever arm moves upward (supplying air to the airsprings) and downward (purging air out of the airsprings) as an indication of raising and sinking of the truck. The model is validated through the comparison of airspring height variation between experimental and simulation data. For this purpose, the rotation data of the lever arm, which is collected during the test, is applied as an input to this model in AMESim. This model includes the pneumatic suspension arranged identically to the experimental configuration and vehicle dynamics in heave, pitch, and roll. Good agreement of suspension deflection between the simulation and experiments is found for both supply and purge tests (Figures 8a and 8b).




(a)




(b)

Figure 8. Comparison of simulation and experiment results of airspring displacement for (a) the supply and (b) purge tests

After validation, the suspension model is integrated into the truck model in TruckSim by a co-simulation technique (Figure 9). It should be noted that the AMESim model is technically merged into the TruckSim model through Simulink, where they work together as S-functions. A Microsoft Visual C++ compiler is needed to convert the AMESim into the S-function. Furthermore, the communication between the AMESim and TruckSim models is accomplished by sending the suspension deflection from TruckSim to AMESim while feeding TruckSim with the airspring force. It is worth noting that the pneumatic suspension system is placed on the tractor's tandem axles and trailer's axles, while the steering axle is equipped with the leaf spring, which is of the common suspension configuration for those trucks operated in the U.S. The input to the co-simulation model includes driving speed and a predetermined path in TruckSim.

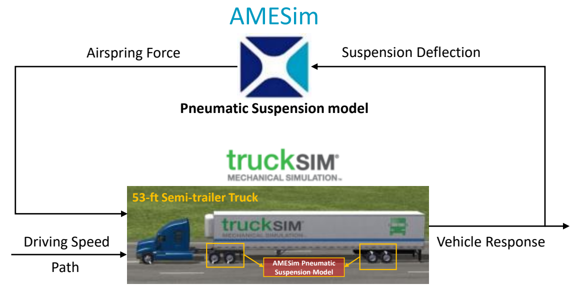



Figure 9. Co-simulation scheme for the roll dynamics analysis of the semi-trailer truck with a rigid off-center load

### 3.3. 43-ft Partially-filled Tank Truck

This is a case of the dynamic-shifted load, where the liquid slosh occurs in a cylindrical tank. A 43-ft partially-filled tank truck model is developed to include the contribution of liquid slosh on vehicle roll dynamics (Figure 10). Dimensional parameters of the tank truck are selected from those commonly operated on U.S. highways, which are provided in Table B1 of Appendix B.

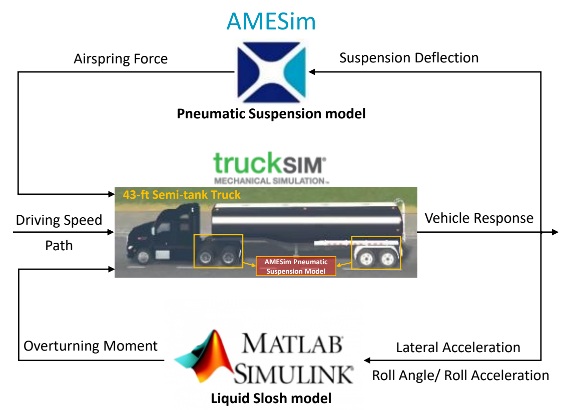



Figure 10. Co-simulation scheme for roll dynamics analysis of a 43-ft partially-filled tank truck

The vehicle dynamics simulation in TruckSim can predict the influence of the rigid cargo but not applicable for emulating the CG (center of gravity) motion due to liquid slosh. Therefore, a model is developed in Simulink using quasi-static slosh theory, which is coupled with the truck model to emulate the lateral slosh effect on vehicle dynamics. It can be seen in Figure 10 that the slosh model at each integration instant receives the lateral acceleration and roll angle from the TruckSim model. In return, the slosh model feeds the TruckSim model with the overturning moment, which includes:

- The overturning moment produced by the centrifugal force applied on the CG that moves in a vertical direction due to the slosh
- The overturning moment generated by the cargo weight acting at the position that laterally offsets from the center line due to the slosh
- The overturning moment produced by the change in roll moment of the deflected liquid load

This model is derived from the implementation of the quasi-static slosh equations to calculate the above overturning moments. For this purpose, a circular-section tank and inviscid and incompressible fluid flow conditions are assumed. The quasi-static formulation is illustrated in Figure 11.

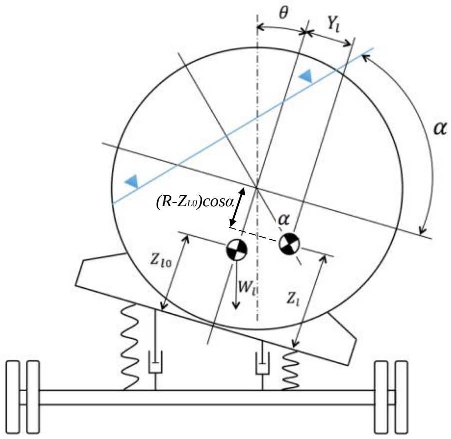



Figure 11. Roll-plane schematic of a partially-filled circular tank

The gradient of the liquid free surface ($\alpha$) can be described as a function of the lateral acceleration ($a_l$) and vehicle roll angle ($\theta$) [22]:

$$\alpha = tan^{-1}\left(\frac{g tan\theta - a_l}{g + a_l tan\theta}\right) \tag{9}$$

where is $g$ is the gravitational acceleration. The motion of the liquid free surface contributes to shifting the liquid CG and therefore alters its inertia properties. Since a circular cross-section tank is assumed, the translation of instantaneous CG of the liquid relative to that of an equivalent rigid cargo, along with lateral ($Y_l$) and vertical ($Z_l$) axes in the body-fixed coordinate system, are given by:

$$Z_l = R - (R - Z_{l0})cos\alpha \tag{10}$$

$$Y_l = (R - Z_{l0})sin\alpha \tag{11}$$

where R is the tank radius, and $Z_{L0}$ is the height of liquid CG in the absence of tank tilt. The change in overturning moment associated with the centrifugal force ($F_l$) is shown in Figure 12 and can be calculated as:

$$\triangle M_{Fl} = F_l(d_1 - d_2 - b) \tag{12}$$

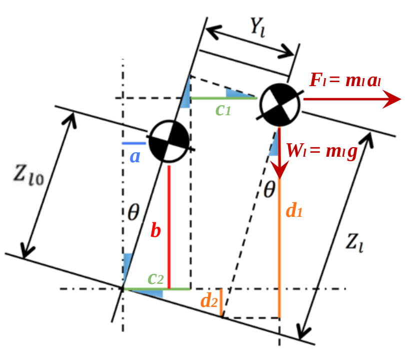



Figure 12. Instantaneous CG position of the liquid cargo

Substituting Equation (12) with the position variables of liquid CG yields:

$$\triangle M_{Fl} = m_l a_l(Z_l cos\theta - Y_l sin\theta - Z_{l0} cos\theta) \tag{13}$$

where $m_l$ is the liquid mass, and $a_l$ is the lateral acceleration imposed on the shifting load center. Similarly, the increase in overturning moment related to the liquid weight ($W_l$) is provided by the following equations:

$$\triangle M_{Wl} = W_l(c_1 + c_2 - a) \tag{14}$$

$$\triangle M_{Wl} = m_l g(Y_l cos\theta + Z_l sin\theta - Z_{l0} sin\theta) \tag{15}$$

where g is the acceleration of gravity. As shown in Figure 13, the roll moment of inertia to the roll center for liquid cargo ($I_l$) and rigid equivalent representation ($I_r$) can be calculated by the parallel axis theorem:

$$I_l = I_0 + [{Y_l}^2 + (Z_l + h_r)^2]m_l \tag{16}$$

$$I_r = I_0 + (Z_{l0} + h_r)^2 m_l \tag{17}$$

where $I_0$ is the initial roll moment of inertia of the liquid, and $h_r$ is the distance from tank bottom to the roll center.

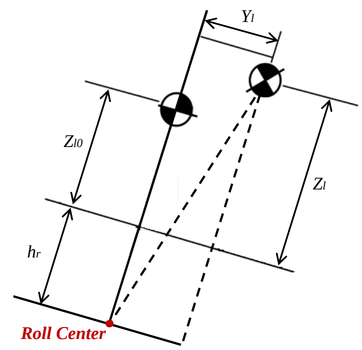



Figure 13. The position of liquid sloshing CG to the roll center

Subtracting Equation (17) from Equation (16) produces the variation in roll inertia due to liquid sloshing:

$$\triangle I_x = (I_l - I_r) = m_l({Y_l}^2 + {Z_l}^2 - {Z_{l0}}^2 + 2h_r Z_l - 2h_r Z_{l0}) \tag{18}$$

Multiplying Equation (18) with the roll acceleration of the vehicle produces an expression of the overturning moment due to the change in the inertia of liquid roll:

$$\triangle M_{Rl} = -\ddot{\theta} \triangle I_x \tag{19}$$

Combining Equations (13), (15), and (19) yields the total overturning moment due to liquid slosh ($M_{Tl}$) as:

$$M_{Tl} = \triangle M_{Fl} + \triangle M_{Wl} + \triangle M_{Rl} \tag{20}$$

According to Equations (9)-(20), Simulink blocks are programmed and integrated into the TruckSim model to represent the effect of the liquid slosh on vehicle dynamics, which can be seen in Figure A2 of Appendix A. The CAD models of liquid cargo for various fill volumes are created in SolidWorks (Figure A3 of Appendix A) to determine the liquid properties such as weight, moment of inertia, and CG position. The details of the parameters for the slosh simulation are provided in Table B2 in Appendix B.

## 4. Simulation Results and Discussion

The effect of the pneumatic suspension is evaluated under straight-ahead and steady-turning (radius= 262.5ft) driving at 20 mph, as shown in Figure 14. Because the rigid shifted load considered in the study causes an incline of the vehicle body to the right side, a left-hand turn is performed to represent the worst-case scenario. The road surface is assumed to be perfectly flat. The vehicle response parameters, such as roll angle and roll rate, are studied for investigating the influence of the suspension on roll dynamics. In the case of liquid slosh, the analytical solutions in terms of liquid CG shifts, roll inertia, and effective moment arm are also discussed.

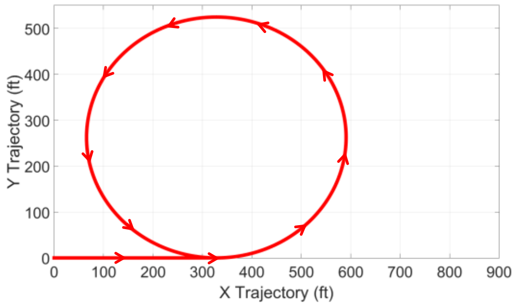
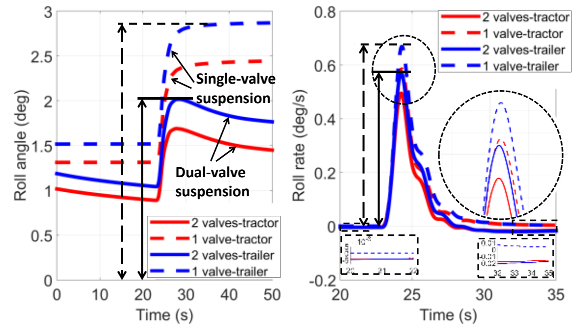



Figure 14. Path of a steady-turning maneuver

### 4.1. Roll Dynamic Performance of a 53-ft Semi-truck with Rigid Shifted Load

Figures 15a and 15b show the time responses of roll angle and roll rate for the conventional suspension (single-valve suspension) and the proposed suspension (dual-valve suspension) when a rigid load is placed 15 in away from the centerline. There is a tilt of the vehicle body caused by the shifted load, as indicated by the tractor and trailer roll angle in Figure 15a, before entering the cornering (t<23s). As compared to the conventional suspension, the dual-valve suspension provides counteractive effect in reaction to the body roll by inflating the airsprings on the jounce side while the rebound side is deflated. Consequently, side-to-side balanced suspension forces are generated to reasonably diminish the body imbalance. In contrast, the single-valve suspension does not provide any active correction to the body imbalance, as shown in Figure 15a. For the transient steering occurring at t=23s, both tractor and trailer equipped with dual-valve suspensions experience an approximately 30% smaller peak roll angle than those of single-valve suspensions. In addition, the dual-valve suspension yields a lower changing rate of rolling motion (less roll-angle overshoot), as illustrated in Figure 15b. This amounts to better body control and stability when confronted with the shifted load. In addition, the dual-valve suspension results in the reduction in settling time, as seen in Figure 15b. It is because the dual-valve suspension uses extra pneumatic energy (balanced airflow) to level the vehicle body during cornering, results in an increase on the dynamic bandwidth of the suspension system making the vehicle body more quickly settle within a small changing rate of the body roll. When the truck enters into the steady turning, the dual-valve suspension enables a sustained return of the body to the leveled position, which is not, however, observed for the single-valve suspension. A preliminary conclusion can be drawn that the dual-valve suspension is capable of counterbalancing the imbalance of the vehicle body resulting from the shifted load and centrifugal forces, thereby improving the vehicle roll stability.




(a) (b)

Figure 15. Simulation results of (a) roll angle and (b) roll rate of the truck with 15-in off-center load subjected to the steady-state turning

Peak values extracted from the roll angle of the tractor and the trailer corresponding to different lateral offsets of the load are provided in Figures 16 and 17. The results show that the suspension with dual leveling valves causes less roll for entire loading conditions. The peak roll angle is improved by nearly 23%-37% for the tractor and trailer. The improvement becomes more pronounced as the load is placed far away from the centerline, since greater pressure is produced on the heavier side to counteract the uneven load. More interestingly, the rate of increase in peak roll angle with increasing cargo offset (from 8 to 25 in) is between 48.2%-52.1% for the tractor and trailer with the dual-valve suspension, which is considerably smaller than those of single-valve suspension (77.6%-77.9%). The findings in Figures 16 and 17 intensify the certainty that the dual-valve suspension is capable of diminishing the destabilizing effect of the shifted load.

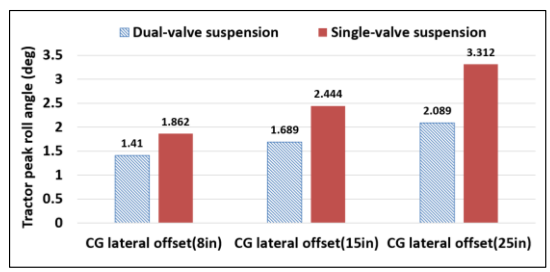



Figure 16. Summary results of tractor peak roll angle for 8-, 15-, and 25-in off-center loads

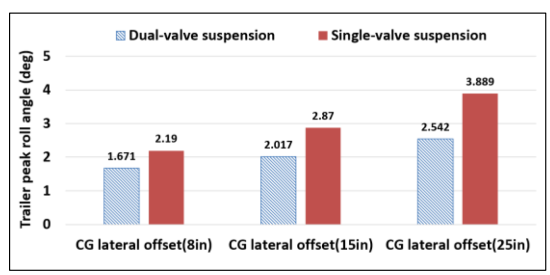
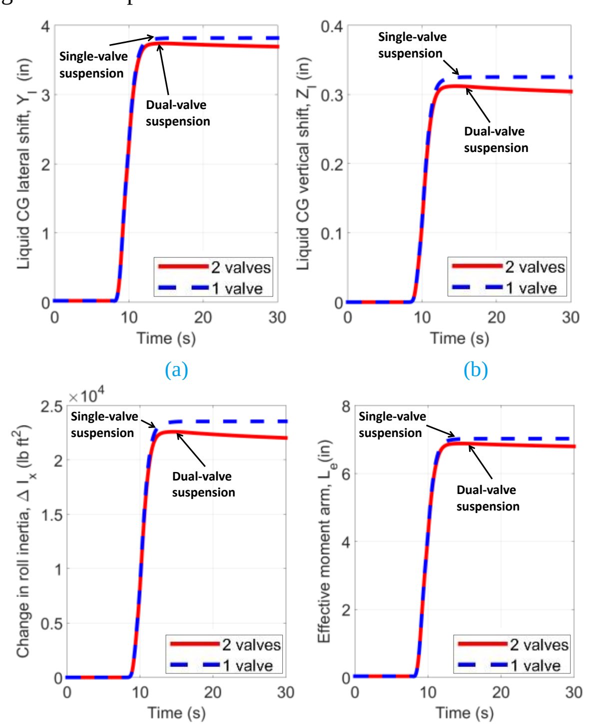



Figure 17. Summary results of trailer peak roll angle for 8-, 15-, and 25-in off-center loads

### 4.2. Roll Dynamic Performance of the 43-ft Partially-filled Tank Truck

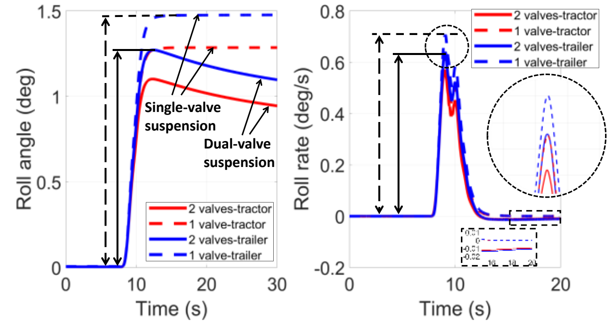
Roll responses of the tank truck, considering a 30% fill volume, are simulated for the steady-turning maneuver shown in Figure 14. The fill level is defined as a ratio of the liquid volume to the tank volume. Figures 18a and 18b present the simulation results of liquid CG shifting in lateral and vertical axes for the single- and dual-valve suspensions. The results show that the liquid CG shifting occurs when the tank is subjected to centrifugal force, while the amount of the shift in the lateral axis is substantially larger than that in the vertical axis. In comparison with the single-valve suspension, the dual-valve suspension results in a reduction of CG shift in both lateral and vertical directions. As shown in Figure 18c, the restraint of load shifting allows the deflected liquid to have a smaller change in roll inertia to the roll center, which is consistent with Equation (18). The dual leveling valve suspension also causes a shorter effective moment arm, which denotes the overturning moment normalized to the load weight ($L_e = (M_{Fl} + M_{wl})/W_l$), as shown in Figure 18d. Overall, the tank truck with the dual-valve suspension exhibits a better restraint of the sloshing behavior than that with the single-valve suspension.



(a) (b) (c) (d)

Figure 18. Simulation results of (a) lateral and (b) vertical CG shifts, (c) changes in roll inertia, and (d) effective moment arm for a 30% volume-filled tank truck for the steady-state turning

Figures 19a and 19b show the comparison of roll angles and roll rates between tank trucks equipped with dual- and single-valve suspensions. As expected, the suspension with dual leveling valves results in less roll angle and roll rate for both tractor and trailer. Notably, the body inclination in the steady-state cornering is considerably diminished when the truck is equipped with the dual-valve suspension, similar to the case of the rigid shifted load.



(a) (b)

Figure 19. Simulation results of (a) roll angle and (b) roll rate for a 30% volume-filled tank truck subjected to the steady-state turning

More simulations are performed to evaluate the suspension under larger fill volumes (60% and 90%). The peak roll angles experienced by the tractor and trailer for the various tank fill volumes are summarized in Figures 20 and 21. The dual leveling valve suspension (blue columns) provides better control in peak roll than the single leveling valve (red columns) over the entire fill range. Specifically, the suspension with dual leveling valves improves the peak roll angle by 14.3%-19.5% for tractors, and 13.9%-18.8% for trailers. Interestingly, as the tank fill volume increases from 30% to 90%, the rate of increase in peak roll angles for the single-valve suspension is 56% and 52.3% for the tractor and trailer, respectively, which are larger than those of the dual-valve suspension (46.6% and 43.7%). This implies that the dual-valve suspension diminishes the rate of roll-stability deterioration due to increasing tank fill volume. It is because the more load transfer occurs during cornering, the better roll control the dual-valve suspension provides (by generating more side-to-side pressure difference thereby more balanced forces). As a result, a better improvement of body roll is also observed for the dual-valve suspension in the case of the rigid shifted load in Section 4.1 as compared with the case of liquid slosh.

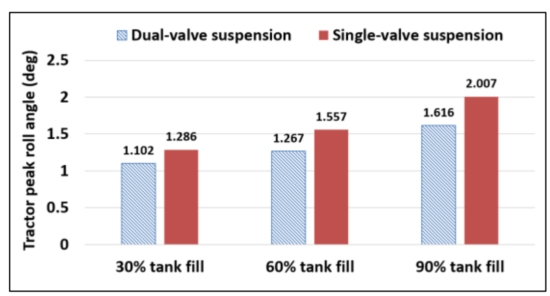



Figure 20. Summary results of tractor peak roll angle for 30%, 60%, and 90% fill volumes

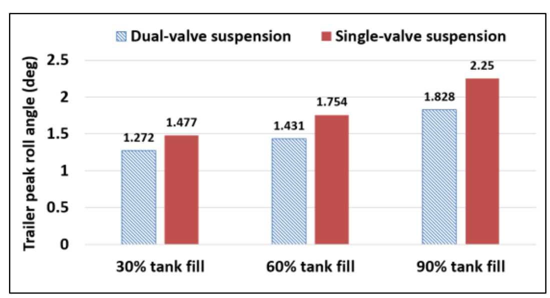



Figure 21. Summary results of trailer peak roll angle for 30%, 60%, and 90% fill volumes

## 5. Conclusions

This paper has presented a simulation investigation into the pneumatic suspension design with dual leveling valves for the resistance of destabilizing impacts of shifted loads. A novel multi-domain modeling technique is proposed and developed using TruckSim, AMESim, and Simulink. Roll responses of trucks equipped with dual-valve suspensions, computed for a steady-turning maneuver, are compared with those with single-valve suspensions. The simulation results indicate that the dual-valve suspension is more effective in reducing the roll angle and rolling rate as compared with the conventional single-valve suspension for both shifted load cases (rigid and liquid shifted loads). It concludes that, as compared to the conventional heavy truck pneumatic suspension system, the proposed pneumatic suspension system (with dual leveling valves) is more favorable in countering the destabilizing effects of shifted loads, conducive to keep the vehicle balanced and provide additional stability. It is also found that a shorter settling time for the changing rate of the body roll can be achieved using the dual-valve suspension. In addition, this study contributes to a co-simulation platform useful for efficient and accurate analyses of the coupled shifted load-pneumatic suspension-vehicle system dynamics.

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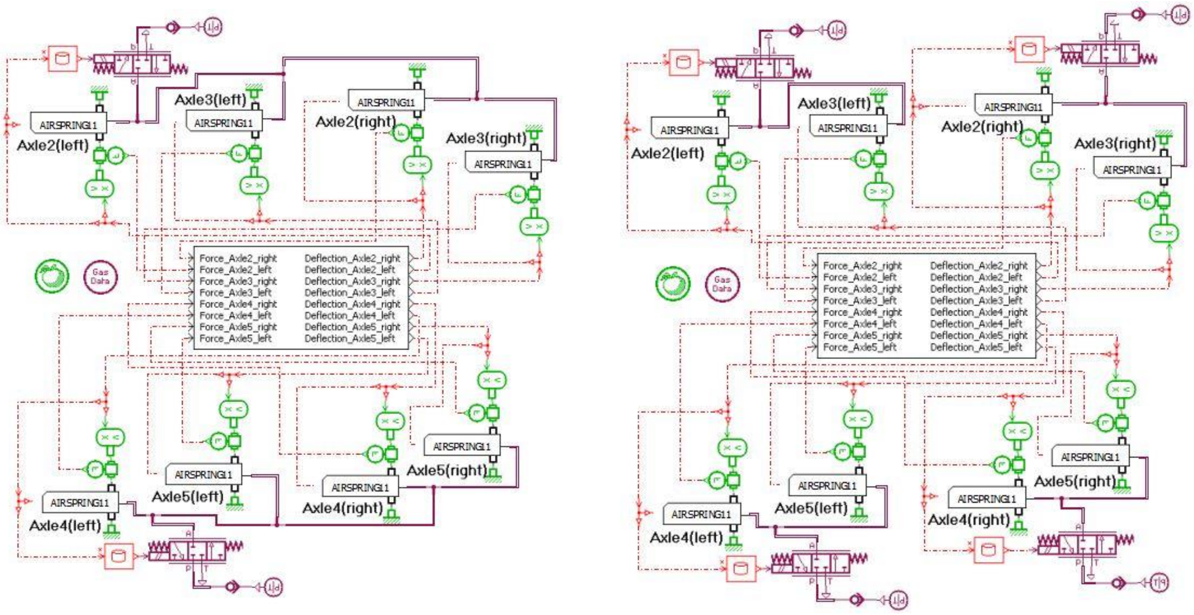
# Appendix A




(a) (b)

Figure A1. Pneumatic suspension models in AMESim: (a) single and (b) dual leveling valves

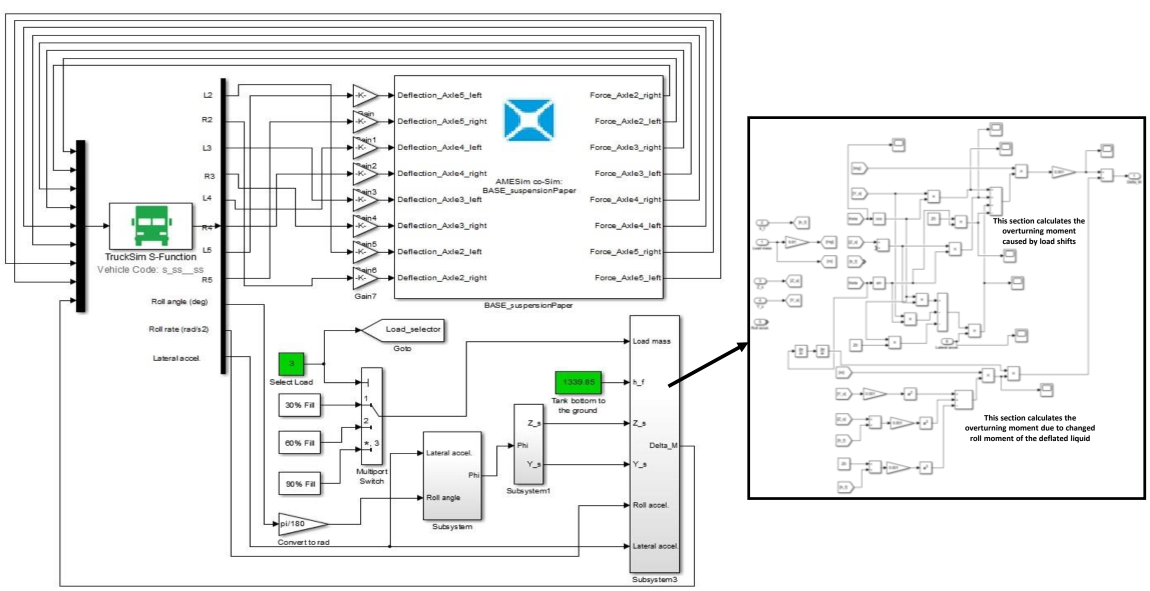
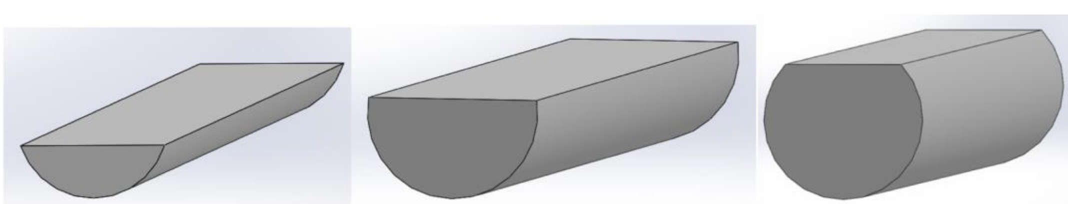



Figure A2. Co-simulation model established in Simulink for dynamic analysis of the tank truck



**30% fill volume** **60% fill volume** **90% fill volume**

Figure A3. CAD models for determining the parameters for the slosh simulation

# Appendix B

Table B1. Parameters for the 43-ft tank truck simulation

| Parameter | Value (English unit) | Value (SI unit) |
|---|---|---|
| 43-ft tank trailer tare weight | 12900.0 lb | 5851.3 kg |
| The vertical distance from the kingpin to trailer axles | 31.0 in | 78.7 cm |
| Distance from kingpin to the front trailer axle | 384.5 in | 976.6 cm |
| Distance from kingpin to the rear trailer axle | 439.5 in | 1116.3 cm |
| The track width of trailer axles | 77.5 in | 196.9 cm |
| Damping coefficient on trailer axles | 2124.2 lb·s/ft | 31.0 kN·s/m |
| The lateral spacing between airsprings on trailer axles | 39.4 in | 100.1 cm |
| The lateral spacing between dampers on trailer axles | 31.5 in | 80.0 cm |
| The vertical distance from tank bottom to roll center | 31.9 in | 81.0 cm |
| The longitudinal distance between kingpin and CG | 255.9 in | 650.0 cm |
| Empty tank roll inertia | 236354.4 lb·ft$^2$ | 9960.0 kg·m$^2$ |
| Empty tank yaw inertia | 4271275.0 lb·ft$^2$ | 179992.0 kg·m$^2$ |
| Empty tank pitch inertia | 4065865.0 lb·ft$^2$ | 171336.0 kg·m$^2$ |
| Tank diameter | 38.0 in | 95.5 cm |
| Liquid cargo density | 51.8 lb/ft$^3$ | 830.4 kg/m$^3$ |

Table B2. Parameters for liquid cargo simulation (30%, 60%, and 90% fill volumes)

| Fill volume | Parameter | Value (English unit) | Value (SI unit) |
|---|---|---|---|
| 30% fill volume of the tank | Liquid mass | 16595.7 lb | 7527.7 kg |
| | Fluid surface to the tank bottom | 25.5 in | 64.8 cm |
| | CG height to the tank bottom | 15.2 in | 38.6 cm |
| | Roll inertia | 49432.5 lb·ft$^2$ | 2083.1 kg·m$^2$ |
| 60% fill volume of the tank | Liquid mass | 33191.5 lb | 15055.4 kg |
| | Fluid surface to the tank bottom | 43.4 in | 110.2 cm |
| | CG height to the tank bottom | 24.7 in | 62.7 cm |
| | Roll inertia | 134223.0 lb·ft$^2$ | 5656.2 kg·m$^2$ |
| 90% fill volume of the tank | Liquid mass | 49787.2 lb | 22583.1 kg |
| | Fluid surface to the tank bottom | 63.3 in | 160.8 cm |
| | CG height to the tank bottom | 34.1 in | 86.6 cm |
| | Roll inertia | 259630.0 lb·ft$^2$ | 10940.8 kg·m$^2$ |