

Handling and Braking Analyses of a Heavy Vehicle with a Cross-Axle Fluidically-Coupled Suspension

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ABSTRACT

The handling and braking responses of a heavy vehicle equipped with a cross-axle fluidically-coupled hydro-pneumatic suspension concept are investigated. The proposed fluidically-coupled suspension is conceived by diagonally interconnecting different hydraulic fluid chambers of the four suspension struts of the vehicle. The analytical formulations of suspension forces are derived based on fluid flows through the couplings and damping valves. A generalized full-vehicle model is developed and validated to evaluate the handling and braking responses to two critical vehicle maneuvers: (i) braking-in-a-turn; and (ii) split- μ straight-line braking. The responses of the vehicle model with the coupled suspension are compared with those of the uncoupled suspension under various inputs to demonstrate the potential benefits of the proposed cross-axle fluidic coupling of the suspension struts.

INTRODUCTION

Heavy vehicles, with considerably higher location of center of gravity (c.g.) and larger load variations, exhibit roll and directional control performance characteristics that are significantly different from those of the light vehicles [1-3]. Constrained by the regulations on vehicle track width, heavy vehicles generally yield significantly larger lateral load transfers during cornering, and thus lower roll stability limit [1, 2]. The braking/acceleration is known to induce vehicle pitch motions and substantial longitudinal load transfers, particularly in vehicles with floating cargoes, as in the case of partly-filled tank trucks [4-6]. The dynamic load variations cause significant variations in normal loads on the tires, which directly affect the longitudinal and cornering forces developed by the tires during cornering and/or braking, and thus the handling and braking performance in an adverse manner.

Considering the lower control and stability limits of heavy vehicles, further efforts for improving ride and handling qualities have been considered desirable [7-9]. A multitude of advanced control systems have evolved to enhance the directional and stability performance of road vehicles through control of forces and moments in

the yaw plane [6, 12, 13]. The applications of such control systems for heavy vehicles, however, have been limited due to associated high cost, weight, complex packaging and system reliability concerns [7-9]. The design of suspension also affects the vehicle performance in a significant manner by tuning the forces transmitted to the vehicle body and the dynamic wheel loads, although it involves complex compromises among different measures related to ride, road-holding and directional control [3, 6, 14, 15]. Furthermore, the anti-roll and anti-pitch properties of a vehicle suspension also form a vital design objective to limit the lateral and longitudinal load transfers. The primary goal for heavy vehicle suspension design is thus to seek a satisfying compromise among ride vibration, road-holding, handling, and roll and pitch performances of the vehicle under different maneuvers and external excitations.

The current heavy vehicle suspensions generally utilize leaf or air springs in conjunction with passive anti-roll bars to realize a compromise among ride vibration, roll stability and directional control performance [1-3]. Hydraulic dampers are also commonly used to achieve control of the resonant oscillations as well as high frequency vibration [15-17]. Owing to the unreasonable risks associated with potential vehicle rollover, relatively greater emphasis is placed on the suspension roll stiffness, which could yield stiffer suspension with relatively poor ride comfort. Multiple axle (tandem and tri-axle) suspensions employ load equalizers to evenly distribute the loads on individual tires, while the equalization is limited to either static or low-speed load transfers [18, 19]. The transverse mechanical interconnections, such as anti-roll bars, are commonly employed to achieve high effective roll stiffness and thus lower roll deflection of the vehicle body during steering maneuvers [1-3, 21]. The anti-roll bars, however, tends to add weight, and deteriorate the ride responses to some extent [21].

Various hydro-pneumatic struts have been developed to realize compact suspension designs with integrated pneumatic spring and hydraulic damping to achieve improved ride comfort and ride height control. Such suspension struts have been employed in military vehicles for about half a century, and are considered to

offer the most significant potential for commercial vehicle applications [14, 22-24]. Moreover, these suspensions offer greater design and tuning flexibilities in either passive or active manners. A few studies have shown that transverse and fore-aft interconnections between the struts can help achieve desirable roll and pitch attitude control [25-32]. A number of reported studies, however, have employed very large size struts with small effective working area, which would require excessive working pressure [25, 27, 28, 31, 32]. The use of a hydro-pneumatic strut in parallel with passive load-sharing springs has also been suggested in order to reduce the operating pressure [29, 31, 32]. The results revealed that the load sharing by the additional springs significantly limits the effectiveness of the roll-interconnections [29]. Moreover, the majority of the reported studies have considered strut designs with external gas chamber and damping valves. The reported experimental and analytical studies, however, have clearly shown that hydraulically interconnected suspension struts could help enhance the anti-roll and anti-pitch performance without affecting the vertical ride.

In this study, the handling and braking dynamics responses of a heavy vehicle employing fluidically-coupled hydro-pneumatic suspension struts are explored. The proposed coupled suspension system involves diagonal interconnections among the different hydraulic fluid chambers of the four suspension struts. A generalized full-vehicle model is developed together with analytical formulations for the suspension forces. The validity of the model is examined and the dynamic responses due to different coupled or uncoupled suspension strut configurations are evaluated under a variety of inputs. The dynamic responses are assessed under two critical vehicle maneuvers, braking-in-a-turn and split- μ straight-line braking.

COUPLED SUSPENSION STRUT FORCES

A generalized 14-DOF model of a two-axle vehicle is developed and validated to investigate the dynamic responses of the vehicle with different coupled or uncoupled suspensions to excitations arising from steering and braking, road roughness and crosswinds, as presented in Appendix A. The vehicle model and its validity have been presented in [42]. Figure 1(a) illustrates the schematic of a strut design concept that integrates the gas chamber and damping valves within the same unit [10, 11]. The shim disc valves are employed within the primary piston to achieve variable resistance to flows between hydraulic fluid chambers 1 and 2. Additional bleed damping orifices in the main piston are also provided for flows between chambers 1 and 2, and 1 and 3. The floating piston isolates the hydraulic fluid in chamber 2 from the nitrogen gas in chamber 4. This compact design provides considerably larger effective working area when compared to those reported in the literature [25, 27, 28, 31, 32], and therefore helps reduce the operating pressure.

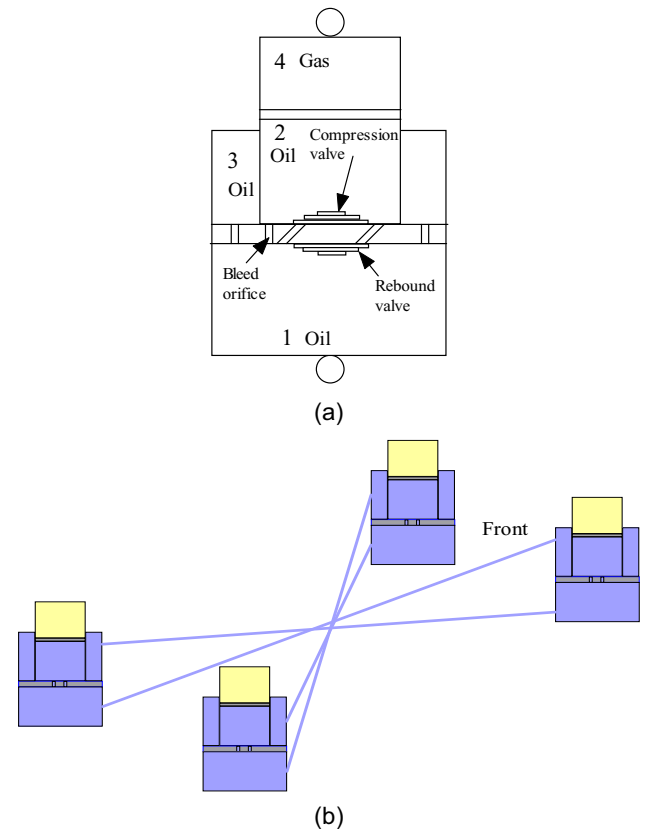


Fig. 1: Schematics of: (a) the hydro-pneumatic strut concept; and (b) a cross-axle fluidically-coupled suspension (B_x).

Figure 1(b) shows the proposed full-vehicle suspension system comprising cross-axle fluidically-coupled struts, referred to as B_x . The proposed configuration involves couplings between the hydraulic fluid chambers 1 and 3 of the four suspension struts, as shown. Chambers 1 and 3 of the rear-left strut are coupled to 3 and 1 of the front-right strut, respectively, and are referred to as $1rl3fr$ and $3rl1fr$. In a similar manner, the chambers 1 and 3 of the rear-right strut are coupled to 3 and 1 of the front-left strut, respectively, and are referred to as $1rr3fl$ and $3rr1fl$. An uncoupled suspension arrangement, referred to as B_u , is also considered to evaluate the relative properties of the fluidic couplings. Both the suspension arrangements employ struts with identical geometry and charge pressure, and thereby yield identical load carrying capacity and suspension rates. The parameters related to damping properties of the two arrangements, however, differ due to additional flows through fluidic couplings in the B_x configuration.

Although the coupled suspension B_x involves complex cross-axle fluid flows across different suspension struts, the forces developed by each strut can be conveniently derived using a generalized model, presented in [10]. The model permits the evaluation of restoring and dissipative suspension forces due to hydro-pneumatic struts, uncoupled or coupled in the roll and/or pitch planes. The forces developed by the unconnected suspension struts (B_u) are identical to those derived in [11], while those developed by the coupled struts depend upon flows through the couplings and the feedback effects in a complex manner. The forces

developed by the front-left (fl) and -right (fr) struts in the coupled suspension (B_X) are derived as:

$$F_f = P_0 \left[A_{1f} \left(\frac{V_{40f}}{V_{40f} + A_{1f}x_f - A_{3r}x_r} \right)^n - A_{3f} \left(\frac{V_{40r}}{V_{40r} + A_{1r}x_r - A_{3f}x_f} \right)^n \right] - P_0 A_{2f} - \frac{\rho A_{1f} |A_{1f}\dot{x}_f - A_{3r}\dot{x}_r| (A_{1f}\dot{x}_f - A_{3r}\dot{x}_r)}{2C_d^2 a_{12f}^2} - \frac{128\mu L A_{3f}^2 \dot{x}_f}{\pi D^4} + \frac{\rho A_{3f} |A_{1r}\dot{x}_r - A_{3f}\dot{x}_f| (A_{1r}\dot{x}_r - A_{3f}\dot{x}_f)}{2C_d^2 a_{12r}^2} \quad (1a)$$

where $r=rr$ for the front-left strut ($f=fl$), and $r=rl$ for the front-right strut ($f=fr$). In a similar manner, the forces developed by the rear struts are also derived, such that:

$$F_r = P_0 \left[A_{1r} \left(\frac{V_{40r}}{V_{40r} + A_{1r}x_r - A_{3f}x_f} \right)^n - A_{3r} \left(\frac{V_{40f}}{V_{40f} + A_{1f}x_f - A_{3r}x_r} \right)^n \right] - P_0 A_{2r} - \frac{\rho A_{1r} |A_{1r}\dot{x}_r - A_{3f}\dot{x}_f| (A_{1r}\dot{x}_r - A_{3f}\dot{x}_f)}{2C_d^2 a_{12r}^2} - \frac{128\mu L A_{3r}^2 \dot{x}_r}{\pi D^4} + \frac{\rho A_{3r} |A_{1f}\dot{x}_f - A_{3r}\dot{x}_r| (A_{1f}\dot{x}_f - A_{3r}\dot{x}_r)}{2C_d^2 a_{12f}^2} \quad (1b)$$

where $f=fl$ for the rear-right strut ($r=rr$), and $f=fr$ for the rear-left strut ($r=rl$). In the above formulations, F_i is the dynamic force developed by strut i ($i=fl, fr, rl, rr$), A_{ji} is the effective piston area reflected on chamber j ($j=1,2,3$) side of the strut i , P_0 is the static charge pressure, x_i and $\dot{x}_i$ are the relative displacement and velocity across the strut i , respectively, assuming small motions with positive direction being upward. V_{40i} is the initial volume of gas in chamber 4 of strut i , C_d is the discharge coefficient, ρ is mass density of the hydraulic fluid, μ represents the dynamic viscosity of hydraulic fluid, and L and D represent the length and diameter of the coupling pipes, respectively. a_{12i} is the effective orifice area due to bleed orifices and damping valves.

The above equations clearly show that the fluidically-coupled struts exhibit strong couplings in the stiffness as well as damping terms. The variations in the damping forces could be realized by varying the effective flow areas of the damping valves. Modeling of damping variations in a piecewise-linear manner [20] would yield non-differential variations in the fluid pressures in the vicinity of the transition pressures. Consequently, a refined model is used to smoothen the flow variations around the transition region [33].

RESPONSES TO DIFFERENT MANEUVERS

The response characteristics of the vehicle model integrating the coupled suspension B_X are evaluated under selected braking and steering inputs. The responses are compared to those of the vehicle with uncoupled suspension B_U to illustrate the performance potentials of fluid couplings. The equations of motion for the full-vehicle model equipped with different suspension configurations are analyzed through simultaneous

solutions of Eqs. (A.1) and (A.2), together with the dynamic strut forces presented in Eqs. (1a) and (1b).

Two critical handling maneuvers, braking-in-a-turn and split- μ straight-line braking, are used for the relative analyses of handling and braking dynamic responses of the vehicle [4, 5, 34, 35]. Two road surfaces with different friction characteristics are considered for the analyses: (i) a dry road with adhesion coefficient of 0.9, referred to as 'dry surface'; and (ii) a wet road with adhesion coefficient of 0.5, referred to as 'wet surface' [36]. The initial vehicle speed for the analyses is set as 28 m/s, while the braking torque distribution is selected to be proportional to the static weight distribution between the two axles of the vehicle [4, 37]. The braking torque and steer angle inputs applied during a braking-in-a-turn maneuver are presented in Fig. 2. Figure 3 illustrates the front and rear braking-torque inputs for the split- μ straight-line braking maneuver, where the left- and right-vehicle tracks are assumed to have road friction coefficients of 0.9 and 0.5, respectively.

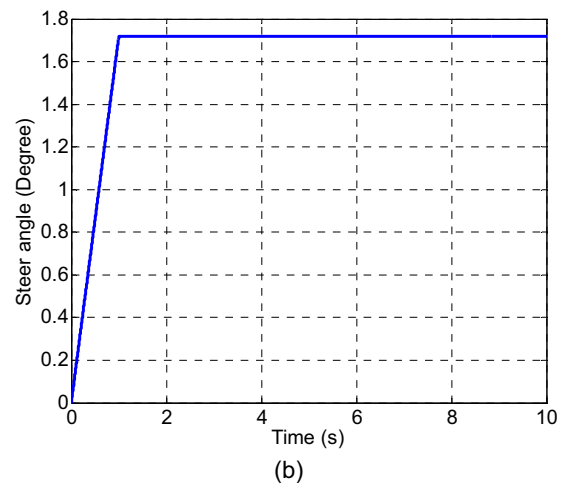
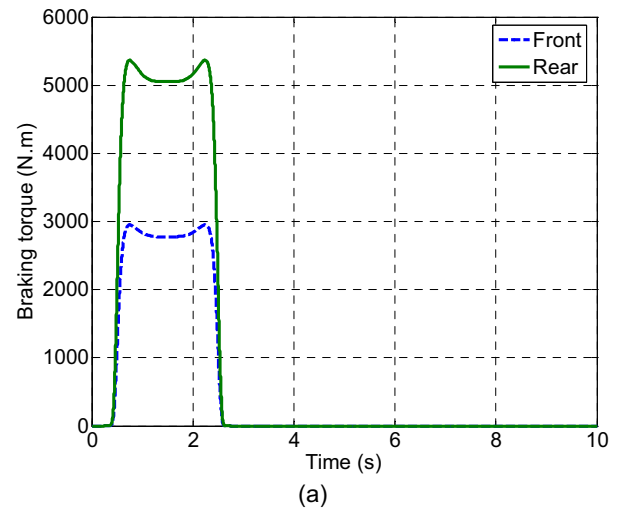


Fig. 2: Representations of the idealized braking-in-a-turn inputs: (a) braking torques; and (b) steer angle.