In Proceedings of the 1995 SAE International Truck and Bus Meeting and Exposition, 1:1-8, Nov 1995.

Ride Comfort and Roll Characteristics of Tandem – Axled Trucks

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ABSTRACT

Usually, vehicles driven under different road conditions are subjected to various forces and moments. An uneven input force originating from road surface could be one of the major load affecting ride comfort or body vibration. In order to investigate the ride and handling characteristics of a tandem axle truck, a three dimensional mathematical model is developed, and the equations of motion for the proposed model with fourteen degrees-of-freedom are derived. Various numerical techniques are used to solve this complex matrix equation. A computer program, capable of vibration isolation and dynamic analysis of tandem-axle truck, is presented. The sensitivity analysis for the parameter variation of the simulation model is investigated and the overall effects of anti-roll bar is also examined.

Introduction

Vehicle suspensions for road vehicles are typically designed such that the sprung mass is isolated from the roadway unevenness. Moreover, they must control the low frequency body motions, i.e, heave, roll and pitch, in order to maintain uniform wheel to road contact. The suspension geometry plays the most significant role in keeping wheels aligned with the direction of travel, and hence, the suspension characteristics affect this, too. Furthermore, it plays a major role in maintaining uniform contact force between the wheels and road, [1, 12].

There are many market requests for design and traction improvement. In particular, improvement of ride comfort is more and more requested by truck users and drivers, since, they drive trucks continuously for a long time. Moreover, they want to keep the damages inflicted on cargo, such as commodities or production materials, caused by vibration shocks, to a minimum level, [3, 13].

Generally, a vehicle suspension system may be categorized as either passive, semi-active or fully-active systems.

Passive suspension systems include the conventional springs and shock absorbers used in most cars. The springs are assumed to have almost linear characteristics; while, most of the shock absorbers exhibit nonlinear relationship between force and velocity. In passive systems, these elements have fixed characteristics and, hence, have no mechanism for feedback control, [10].

Semi-active suspensions provide controlled real-time dissipation of energy, [2, 10]. For an automotive suspension this is achieved through a mechanical device called active damper which is used in parallel with a conventional spring. The main advantage of this system is to adjust the damping of the suspension system, without any use of actuators. This type of system requires some form of measuring devices with a controller board in order to tune the damping properly.

Active suspension employs pneumatic or hydraulic actuators which in turn creates the desired force in the suspension system, [5, 11]. The actuator is secured in parallel with a spring and shock absorber. Active suspension requires sensors to be located at different points of the vehicle to measure the motions of the body, suspension system and/or the unsprung mass. This information is used in the online controller to command the actuator in order to provide the exact amount of force required. Active suspensions may consume large amounts of energy in providing the control force, and therefore, in the design procedure for the active suspension the power limitations of actuators should also be considered as an important factor.

In this study the conventional passive suspension system is considered and a sensitivity analysis is forwarded, which is the basic tool for these kind of suspension system design. Many researcher have studied the semi-active or active suspension for improving ride comfort and road handling characteristics, [7, 9, 14]. However, to have a good active design it is necessary to carefully analyse the passive system first, [8]. Then it is more effective to add a control strategy to remedy the drawbacks of passive systems, [6, 16].

The ride environment of the truck driver is a combination of the input excitation and the output response properties of the truck. There are various sources from which truck ride vibrations may be excited. These generally fall into two categories: on-board sources, and road roughness, [4]. The on-board sources arise from rotating components and these include the tire/wheel assemblies, the drive line and the engine.

Roughness of the road surface is generally considered as one of the main problems in evaluating vehicle dynamic performance on rough roads. In order to achieve efficient transportation, the vehicle needs to be operated at high speed with a minimum expenditure of energy. However, for rough road operations, several important problems should be considered which, in turn, result in changing the vehicle performance. The main parameters which effect vehicle behavior on rough road surfaces are, ride quality, controllability, tractive effort and durability, i.e. the vehicle's life cycle cost.

An uneven input force originating from the road surface irregularities is divided into the following categories:

(i) Random input force corresponding to the smooth or rough road.
(ii) Periodic input force resembling the periodically uneven road surface.
(iii) Single input force corresponding to the protruding surface.

Depending on the type of the input force from the road surface, the relevant evaluation technique must be adopted. In this paper, the ride comfort and handling characteristics of a tandem-axle truck are investigated. The equations of motion for the proposed model, with fourteen-degrees-of-freedom, are derived. Various numerical techniques are employed in order to solve the complex matrix equation. The sensitivity analysis for the parameter variations is carried out in this study and the overall effects of anti-roll bar are also examined.

Vehicle Model

Various vehicle models are used in the practice of dynamic simulation, their complexity depends upon the type of problem being investigated. The analysis have revealed that the indicated accuracy does not increase proportionally with the model complexity level, which means that the model used should be matched with the purpose of the detailed investigation.

Bearing in mind that the objective of this study was simultaneous minimization of bounce, roll, and pitch motions of the body together with the vertical oscillations of the wheels, it is appropriate to use the spatial truck vibration model as shown in Fig. 1.

The computer simulation model, having fourteen-degrees-of-freedom, makes it possible to analyze the following oscillatory motions of the truck:

(a) Sprung mass bounce, x; pitch, p; roll, R; lurch, y; and, yaw, w;
(b) Front axle bounce, x_1; roll, R_1 ;and, lurch, y_1;
(c) First rear tandem axle bounce, x_3; roll, R_3 ; and, lurch, y_3;
(d) Second rear tandem axle bounce, x_5 ;roll, R_5 ; and lurch, y_5;

The oscillatory movements of the truck are caused by the excitation generated by the roughness of longitudinal micro-profile of the road. The input to the left wheels is $y_i = E_i \sin(\omega t) + F_i \cos(\omega t)$, and the corresponding input to the right wheels is $y_j = E_j \sin(\omega t) + F_j \cos(\omega t)$. For the front axle i=1 and j=2, while for the first tandem axle the values are i=3 and j=4, and for the second tandem axle i=5 and j=6.

Referring to the mathematical model presented in Fig. 2, the designations of Tab. 1. will apply. Equations of motion for the fourteen-degrees-of-freedom model are derived and explained in detail in Reference [15]. These equations could be written in the matrix form as:

$$\mathbf{M}\,\ddot{\mathbf{x}} + \mathbf{C}\,\dot{\mathbf{x}} + \mathbf{K}\,\mathbf{x} = \mathbf{u} \qquad (1)$$

M	Sprung mass
M_1	Mass of the front axle
M_2	Mass of the first rear tandem axle
M_3	Mass of the second rear tandem axle
I_R, I_P, I_W	Moments of inertia of the sprung mass
I_{R1}, I_{R3}, I_{R5}	Moments of inertia of the axles
a_i, a_j b_1, b_2, b_3	Positions of the center of gravity of the sprung mass
h_i, h_j, l_i, l_j	Linear joint dimensions

Table 1: Parameter designation of the model

where $\mathbf{M}$ is a (14×14) mass matrix, $\mathbf{C}$ is a (14×14) damping matrix and $\mathbf{K}$ is a (14×14) stiffness matrix. Furthermore, $\mathbf{u}$ is a (14×1) vector representing the input motion, due to the road irregularities, subjected to the vehicle, and $\mathbf{x}$ being the degrees of freedom. The detail description of the system matrices is given in Appendix.

The dynamic matrix in the absence of damping mechanism forms a square symmetrical matrix, and, is a general representation of the 14 DOF model.

Computer Simulation Results

A computer program is developed in order to solve the characteristic equation, from which the eigenvalues and eigenvectors are determined. The fourteen undamped natural frequencies are evaluated and shown in Tab. 2.

ω_{n1}	5.584 (rad/s)	ω_{n2}	5.651 (rad/s)
ω_{n3}	10.572	ω_{n4}	14.774
ω_{n5}	23.136	ω_{n6}	53.516
ω_{n7}	93.88	ω_{n8}	101.00
ω_{n9}	101.50	ω_{n10}	101.53
ω_{n11}	110.43	ω_{n12}	116.36
ω_{n13}	292.85	ω_{n14}	298.27

Table 2: Natural frequencies of tandem-axle truck

Different numerical techniques have been considered for this analysis and, finally, the modified Gauss-Jordan numerical method has been preferably employed due to its rather fast and accurate convergence.

A detailed explanation with its flow chart has been fully explained in [15], and, its merits has been demonstrated. The values of parameters and the technical data associated with this truck are taken from an Iranian truck manufacturing company. Figures 3 to 7 show the frequency response of the bounce, roll, pitch, yaw, and lurch motions of the truck body. Frequency variations of the bounce, lurch, and roll of the front axle of the truck are presented in Figures 8, 9, and 10. The bounce, lurch, and roll motions of the first rear axle as a function of frequency are illustrated in Figures 11, 12, and 13. The corresponding diagrams for the second rear axle are shown in Figures 14, 15, and 16, respectively.

Parameter Sensitivity Analysis

In order to study the sensitivity of the truck behavior to the variation in the vehicle parameters, two different models have been investigated, the results of which are given in [15]. In comparing these results, it was shown that for the bounce of the sprung mass, the resonant frequency tends to decrease when the suspension compliances are increased. Moreover, as a result, the body roll decreases rapidly.

It was also shown that the anti-roll bar plays an important role on the overall dynamic stability of the truck. In order to verify this effect, one may completely eliminate the anti-roll bar from the model by just setting the value of its stiffness equal to zero. It was found that the roll, yaw and lurch motions associated to the sprung mass are increased considerably, when the anti-roll bar is removed. Therefore, the overall dynamic stability of the truck is severely deteriorated.

Consequently, the inclusion of anti-roll bar is shown to be detrimental to the ride, handling and stability of road trucks, and therefore, it is most appropriate to design a suitable anti-roll bar for every truck.

Conclusions

Through the vehicle modeling and simulation analysis of a tandem axle truck the relationship between body roll and bounce, together with the vehicle stability and ride have been investigated. It was shown that the computer simulation model, in its general form, provides a good basis for studying the ride, handling and stability of trucks. Parameter sensitivity analysis shows that the suspension compliance and the stiffness of the main springs play important roles on the overall dynamic stability of the truck. Moreover, the presence of anti-roll bar provides better ride, dynamic stability and overall controllability of the truck.

Figure 1: Three dimentional view of a tandem-axle road truck

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Appendix

The governing equation of the system has been given in matrix form as:

$$\mathbf{M}\,\ddot{\mathbf{x}} + \mathbf{C}\,\dot{\mathbf{x}} + \mathbf{K}\,\mathbf{x} = \mathbf{u}$$

where mass matrix M is a diagonal matrix in the form of:

$$\mathbf{M} = \mathrm{diag}(M, I_P, M_1, M_3, I_R, I_{R1}, I_{R3}, M, I_W, M_1, M_3, M_5, I_{R5})$$

The stiffness matrix $\mathbf{K}$ is a 14×14 matrix which is illustrated in the next two pages. The damping matrix $\mathbf{C}$ has similar structure to $\mathbf{K}$, but the K's element should be replaced by C's respectively. Finally, the road input vector $\mathbf{u}$ could be represented by:

$$\mathbf{u} = \begin{bmatrix} 0 \\ 0 \\ K_{t1}(E_1+E_2)\sin(\omega t) + C_{t1}\omega(F_1+F_2)\cos(\omega t) \\ K_{t3}(E_3+E_4)\sin(\omega t) + C_{t3}\omega(F_3+F_4)\cos(\omega t) \\ 0 \\ K_{t1}a_2(E_1-E_2)\sin(\omega t) + C_{t1}\omega a_2(F_1-F_2)\cos(\omega t) \\ K_{t3}a_4(E_3-E_4)\sin(\omega t) + C_{t3}\omega a_4(F_3-F_4)\cos(\omega t) \\ 0 \\ 0 \\ 0 \\ 0 \\ K_{t5}(E_5+E_6)\sin(\omega t) + C_{t5}\omega(F_5+F_6)\cos(\omega t) \\ 0 \\ K_{t5}a_6(E_5-E_6)\sin(\omega t) + C_{t5}\omega a_6(F_5-F_6)\cos(\omega t) \end{bmatrix}$$

Figure 2: Mathematical model of a tandem–axle truck

Figure 3: Frequency responce for BOUNCE motion of sprung mass

Figure 5: Frequency responce for ROLL motion of sprung mass

Figure 4: Frequency responce for PITCH motion of sprung mass

Figure 6: Frequency responce for LURCH motion of sprung mass

Figure 7: Frequency responce for YAW motion of sprung mass

Figure 8: Frequency responce for BOUNCE motion of front axle

Figure 9: Frequency responce for LURCH motion of front axle

Figure 10: Frequency responce for ROLL motion of front axle

Figure 11: Frequency responce for BOUNCE motion of first rear axle

Figure 12: Frequency responce for LURCH motion of first rear axle

Figure 13: Frequency responce for ROLL motion of first rear axle

Figure 14: Frequency responce for BOUNCE motion of second rear axle

Figure 15: Frequency responce for LURCH motion of second rear axle

Figure 16: Frequency responce for ROLL motion of second rear axle