

Experimental and Analytical Evaluations of a Torsio-Elastic Suspension for Off-Road Vehicles

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ABSTRACT

The ride performance potentials of a prototype torsio-elastic axle suspension for an off-road vehicle were investigated analytically and experimentally. A forestry vehicle was fitted with the prototype suspension at its rear axle to assess its ride performance benefits. Field measurements of ride vibration along the vertical, lateral, fore-aft, roll and pitch axes were performed for the suspended and an unsuspended vehicle, while traversing a forestry terrain. The measured vibration responses of both vehicles were evaluated in terms of unweighted and frequency-weighted rms accelerations and the acceleration spectra, and compared to assess the potential performance benefits of the proposed suspension. The results revealed that the proposed suspension could yield significant reductions in the vibration magnitudes transmitted to the operator's station. The field evaluations revealed that suspended vehicle could yield 35%, 43% and 57% lower frequency-weighted rms accelerations in the x -, y - and z -axis, respectively, compared to the unsuspended vehicle, when loaded. A 13-degrees-of-freedom model of the suspended vehicle was subsequently developed and validated using the measured data, which could serve as a design tool for deriving optimal suspension designs for a wide range of vehicles. The model results revealed reasonably good agreements with the measured vibration spectra. From the simulation results, it was further concluded that a reduction in the vertical stiffness of the torsio-elastic member would yield

beneficial effects on the overall weighted vertical and pitch rms acceleration magnitudes.

INTRODUCTION

The logging vehicles (or skidders) used for harvesting of trees in the forestry sectors are generally unsuspended vehicles, which yield highly complex ride vibration along the translational and rotational axes arising from dynamic interactions of the vehicle body with terrains. The whole-body vibration (WBV) environment of such vehicles can be characterized by high magnitudes of low frequency vibration and shocks [1], which may exceed the health caution guidance zone for a 4-h work day, as defined in the current international standard, ISO-2631-1 [2]. Neitzel and Yost [3] further showed that the 8-h equivalent vibration exposure of forestry workers, including the skidder drivers, exceeds the 8-h exposure limit defined by the Commission of the European Communities [4]. These studies have also shown that the whole-body vibration of a forestry skidder tends to dominate in the low frequency range (< 5 Hz), partly attributed to large diameter soft tires with light damping.

Exposure of high intensity and low frequency whole-body vibration has been widely associated with several musculoskeletal disorders among the human operators [5]. A few epidemiological studies have also established high prevalence of low back pain among the forestry vehicle

operators [6, 7]. Despite the high intensity WBV environment of forestry skidders and the associated health and safety risks, only limited efforts have been made to control the transmission of vibration to the operator. Considering the light damping due to large diameter soft tires, the ride vibration properties of a skidder may be characterized by the response of a lightly damped mechanical system. Furthermore, vehicle operation on relatively rough terrains coupled with large size tires and high location of the operator contribute to high magnitudes of lateral and fore-and-aft vibration at the operator's location.

A variety of primary/secondary suspension designs have evolved for different wheeled off-road vehicles [8,9,10,11]. The implementation in forestry skidders, however, has been mostly limited to low natural frequency vertical seat suspension [12]. This may be partly attributed to increased vehicle rollover or tipover risk associated with a primary suspension operating on forestry terrains with considerable unevenness, grade and cross-grade [13, 14]. The suspension seats are designed with low natural frequency ($\approx 1.5\text{ Hz}$) to attenuate vertical ride vibration, while the suspension travel is generally limited to $\pm 5\text{ cm}$ by introducing elastic end-stops [15]. The vehicle interaction with rough forest terrains may cause the suspension seat to exceed its free travel and transmit high intensity vibration or shock motions to the operator due to end-stop impacts. A few studies have shown that the suspension seat do not offer significant reduction in vertical vibration [1, 16]. These studies further stated that a vertical suspension seat may somewhat amplify the vibration transmitted to the operators.

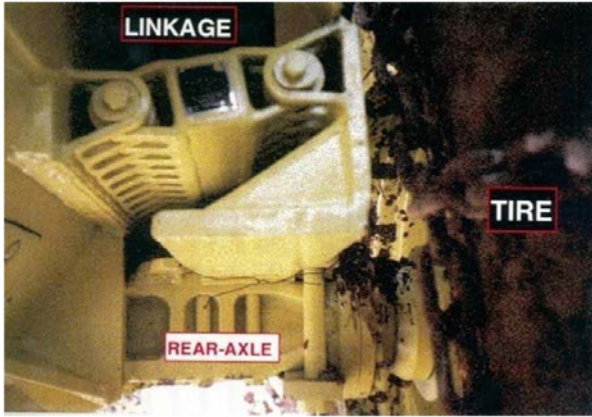
A number of cab suspension designs and concepts have also evolved to attenuate vibration along the translational as well as rotational axes [17,18,19]. The cab suspensions generally cannot provide attenuation of low frequency lateral vibration (predominant around 1 Hz). Furthermore, the implementation of a suspension at the cab could be quite complex in small-size forestry vehicles, where the cab is often welded to the chassis. Alternatively, a number of primary suspension designs have been explored for agricultural and construction vehicles to achieve improved ride comfort for the operators and reduce the stresses in the vehicle structure and components [8, 9, 20,21,22]. Similar developments in primary suspensions for forestry skidders have been lacking, which may be attributed to a few factors: (i) the low frequency characteristics of ride vibration, which would require low natural frequency suspension and thus reduced roll and lateral stability limits; (ii) the requirements of high roll and pitch stiffness to ensure adequate directional/rotational stability limits of the vehicle operating on uneven forestry terrains; (iii) complexities associated with implementation of a wheel suspension since the axles are generally welded to the chassis structure; and (iv) extreme variations in the axle loads, which pose considerable design

challenge and may limit the suspension performance to particular operating conditions.

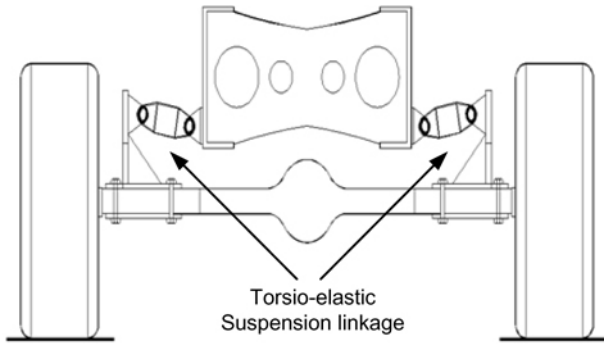
This study proposes a concept in a torsio-elastic linkage suspension in an attempt to enhance vehicle ride performance coupled with high lateral and roll stiffness for improved roll stability. The performance potentials of the proposed suspension concept are investigated analytically and through field measurements of a prototype suspension implemented in a forestry skidder. The potential benefits of the suspension are demonstrated by comparing the field-measured vibration characteristics of the suspended skidder with those of the conventional unsuspended skidder. A ride dynamic full-vehicle model of a skidder employing the proposed torsio-elastic suspension is formulated and analyzed to investigate the influences of variations in the suspension design parameters on the ride performance.

MODELING OF A SKIDDER WITH TORSIO-ELASTIC REAR-AXLE SUSPENSION

The torsio-elastic suspension comprises a linkage mechanism, oriented along the lateral axis of the vehicle that couples the chassis to the unsprung mass. The suspension linkage mechanism envelopes two 0.6 m long tubular shafts of elliptical cross section ($43\text{ mm} \times 38\text{ mm}$), oriented along the longitudinal axis of the vehicle, directly coupling the sprung and unsprung masses through a series of clevises, as illustrated in the pictorial and schematic views in [Figure 1](#). Each tubular shaft comprises an elastomeric bar reinforced with a 16 mm metallic shaft in the center. The nominal width of the suspension link is 160 mm , while its length could be varied depending upon the desired load carrying capacity. The proposed torsio-elastic suspension is introduced at the rear axle, which eliminates the rigid coupling between the chassis and the rear axle, while providing high effective roll stiffness due to restoring moment capability of the torsional shafts. The laterally-oriented linkage suspension undergoes roll motion due to relative motions between the unsprung and sprung masses, which yields a kinematic advantage and absorption of energy due to torsional deflections of each composite shaft. Furthermore, the composite shaft could also undergo radial deflections.



(a)



(b)

Figure 1. (a) A pictorial view of the rear-axle torsio-elastic suspension; and (b) roll-plane schematic of the suspension system.

A forestry vehicle incorporating the linkage suspension at the rear-axle is analytically modeled as a 13-DOF dynamic system. Figures 2(a) and 2(b) illustrates the pitch and roll planes of the vehicle model with the rear-axle suspension, respectively, while the detailed linkage suspension model is shown in Figure 2(c). The vehicle body, chassis, cab, and the mounted equipment together with the unsprung front axle are represented by a lumped sprung mass (m_s) with four-DOF: vertical (z_s), roll (ϕ_s), pitch (θ_s), and lateral (y_s). The rear axle is represented by a rigid body with three-DOF: vertical (z_{ur}), lateral (y_{ur}), and roll (ϕ_{ur}). Each suspension linkage mechanism on the right and left side is considered to possess three-DOF motion along the vertical ($z_{3,4}$), roll ($\phi_{3,4}$), and lateral ($y_{3,4}$) directions. Each tire is modeled as a parallel combination of linear spring and viscous damping, assuming a point contact model. The torsio-elastic suspension is characterized by its radial and torsional visco-elastic properties. The radial visco-elastic properties are represented by linear stiffness and damping elements along the vertical (k_z, c_z) and lateral (k_y, c_y) axes, while the torsional properties

are characterized by linear torsional stiffness and damping (k_θ, c_θ). The equations of motion for the vehicle model subject to terrain excitations are formulated as follows:

$$\begin{aligned}
 m_s \ddot{z}_s + \sum_{i=1}^2 F_{Ti} + \sum_{i=3}^4 (F_{zi}) &= 0; & m_s \ddot{y}_s + \sum_{i=3}^4 F_{yi} + \sum_{i=1}^2 F_{TYi} &= 0 \\
 I_{XS} \ddot{\phi}_s + M_{\theta 4s} + M_{\theta 3s} + (F_{z4} - F_{z3}) L_1 - \sum_{i=3}^4 F_{yi} h + T(F_{T2} - F_{T1}) + \sum_{i=1}^2 F_{TYi} (h + h_1 + R) &= 0 \\
 I_{YS} \ddot{\theta}_s + \sum_{i=3}^4 F_{zi} W B_2 - \sum_{i=1}^2 F_{Ti} W B_1 &= 0 \\
 m_{ur} \ddot{z}_{ur} + \sum_{i=3}^4 F_{Ti} - \sum_{i=3}^4 F_{zii} &= 0; & m_{ur} \ddot{y}_{ur} - \sum_{i=3}^4 F_{yiu} + \sum_{i=3}^4 F_{TYi} &= 0 \\
 I_r \ddot{\phi}_{ur} - (F_{T3} - F_{T4}) T - \sum_{i=3}^4 [R + F_{yiu} h_1 + F_{zii} (L_1 + L_0)] - M_{\theta 3u} - M_{\theta 4u} &= 0 \\
 m_3 \ddot{z}_3 + F_{z3u} - F_{z3} &= 0; & m_4 \ddot{z}_4 + F_{z4u} - F_{z4} &= 0 \\
 I_3 \ddot{\theta}_3 - (F_{z3u} + F_{z3}) \frac{L_0}{2} - M_{\theta 3s} + M_{\theta 3u} &= 0; \\
 I_4 \ddot{\theta}_4 - (F_{z4u} + F_{z4}) \frac{L_0}{2} - M_{\theta 4s} + M_{\theta 4u} &= 0 \\
 m_3 \ddot{y}_3 + F_{y3u} - F_{y3} &= 0; & m_4 \ddot{y}_4 + F_{y4u} - F_{y4} &= 0
 \end{aligned} \tag{1}$$

<figure 2 here>

where I_{XS} and I_{YS} , respectively, are the roll and pitch mass moments of inertia of the sprung mass. I_r , I_3 and I_4 are the roll mass moment of inertia due to rear axle, and right and left suspension links, respectively. The geometric parameters: L_0 , L_1 , R , h and h_1 are illustrated in Figure 2. In the above equations, the subscripts “3” and “4” of the forces or moments refer to those developed by the right- and left-suspension units, respectively. F_{z4} and F_{z3} are the vertical forces acting on the sprung mass developed due to left- and right-axle suspensions, respectively; F_{z3u} and F_{z4u} are those acting at the unsprung mass attachments; (F_{y3} , F_{y4}), and (F_{y3u} , F_{y4u}) are the lateral forces acting on the sprung and unsprung masses, respectively, due to radial stiffness and damping properties of the shafts; ($M_{\theta 3s}$, $M_{\theta 4s}$), and ($M_{\theta 3u}$, $M_{\theta 4u}$) are roll moments acting on the sprung and unsprung masses due to torsional stiffness and damping properties of the left and right torsio-elastic suspension; and F_{TYi} ($i = 1, \dots, 4$) are the lateral forces developed at the tire-terrain interfaces. The forces and the moments developed due to the tire and suspensions are described in details in Appendix I. The coupled differential equations of motion of the 13-DOF skidder model can be expressed in the following matrix form:

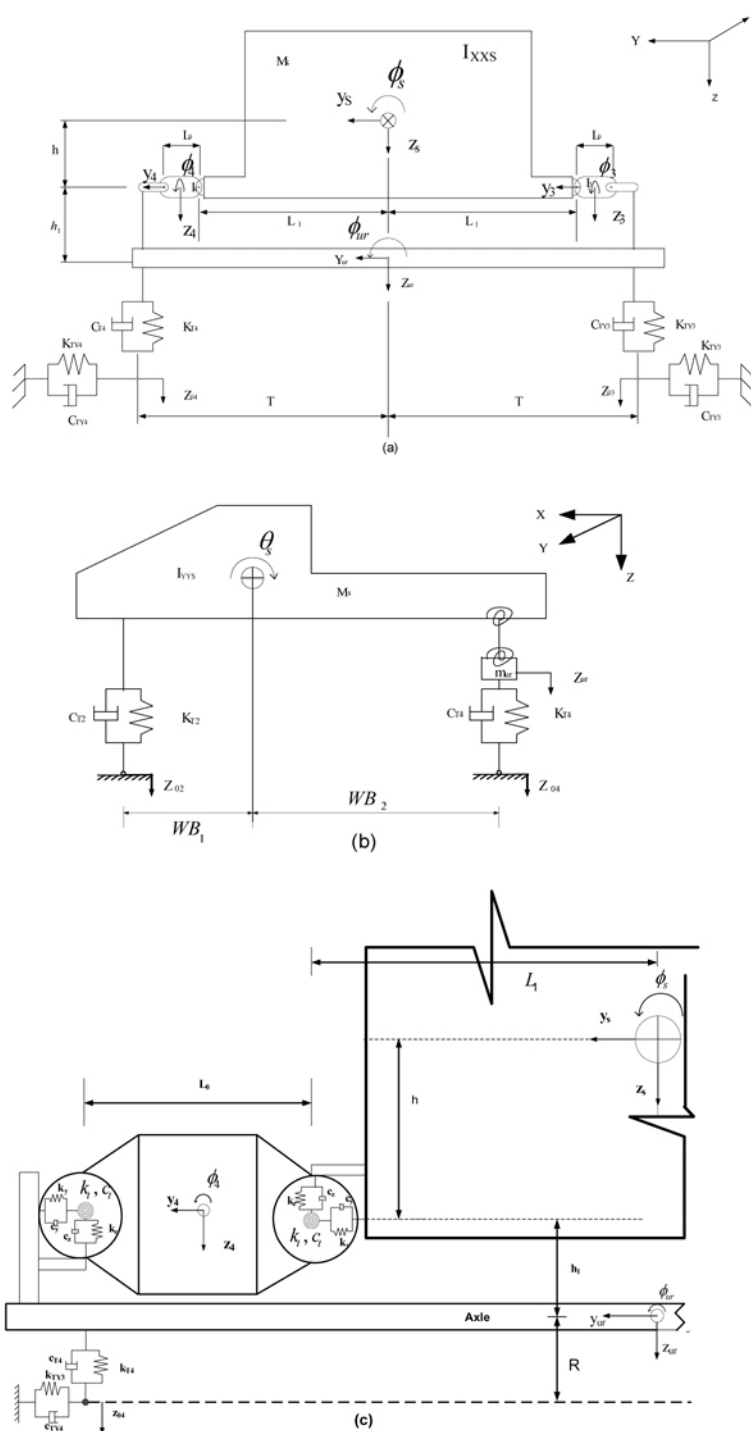


Figure 2. Schematic representation of the 13-DOF suspended vehicle model: (a) roll plane; (b) pitch plane; and (c) torsio-elastic linkage.

$$[M]\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = [C_F]\{\dot{q}_0\} + [K_F]\{q_0\} \quad (2)$$

where $[M]$, $[C]$ and $[K]$ are (13×13) mass, damping and stiffness matrices, respectively, and $[C_F]$ and $[K_F]$ are

(13×4) forced damping and stiffness matrices. $\{q\}$ and $\{q_0\}$ are the vectors of generalized and excitation coordinates, given by: