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Riding comfort Simulation and Analysis of Solid Tire equipped Forklift Truck

Penpisut Saraithong*, Sawangtit Srikitsuwan and Arisara Chaikittiratana

Department of Mechanical and Aerospace Engineering, King Mongkut's University of Technology North Bangkok, Thailand.
* Corresponding Author, E-mail: penpisutsarai@hotmail.co.th

Abstract

Solid tires are commonly used on forklift trucks in many industries. These forklift trucks occasionally work on rough surfaces and with no absorbers on their suspensions, vibrations during operations are inevitably prominent. Riding comfort, and consequently, long term health of the drivers are becoming increasingly concerned. Riding comfort is affected by vibrations during services and the characteristics of the solid tires used are one of the important influences. The present work aims to develop a methodology for quick accessing the influence of solid tires on vibratory behavior of a forklift truck by means of the numerical simulations using MATLAB-Simulink program and the analyses of experimental data. The results obtained from such methodology can be used to compare the riding comfort affected by solid tires from different manufacturers and thus improvements on the quality of solid tires can be suggested.

Keywords*:* Forklifts, Solid tires, Riding comfort, MATLAB, Simulations

1. Introduction

Solid tires are commonly used in the industry for forklift trucks. These trucks generally work on many types of ground surfaces and irregularities. As they are not equipped with suspension systems, mechanical vibrations emitted during operations can be considerably high. Solid tires can be considered as a few suspension parts in a forklift truck and thus primarily influence the vehicle's vibration behavior. Vibration of a forklift are generally caused by the interaction of the wheels and the road surfaces. Vibration greatly affects riding comfort, and consequently, can lead to fatigue and long term health problems of the drivers [1]. The exposure time to the whole body vibration caused by the vehicle's vibration emission should not be greater than the recommended standard set by ISO (ISO 2631-1, 1997) [2]. Therefore, to improve the ride quality, the vibration of vehicle body as well as the passenger seat should be kept minimum. These issues could be tackled with the help of mathematical models to study the vibration behavior of vehicle systems. The widely used dynamic vehicle models are such as quarter car model, 2-DOF half car model, 4-DOF half car model, and full car model [3]-[6].

The goal of this study is to propose a simple methodology for a quick and inexpensive investigation of the influence of solid tire properties on the mechanical vibration behavior of a forklift truck. The methodology consists of a simple method for determining the mechanical properties of solid tires from bouncing experiment and a simple numerical simulation model using Matlab/Simulink for analyzing the dynamic vibration behavior of the vehicle using 4-DOF half car model and to evaluate the influence of solid tire properties on the vibration behavior of a forklift truck.

2. Determination of solid tire properties from tire bouncing experiment

A simple method for determining the mechanical properties of solid tires was devised for a quick determination of solid tires mechanical properties. These properties are used as the input parameters in the proposed numerical simulation model of a forklift, described in the later section. The tire properties determined from the method proposed were linear stiffness *k*, and viscous damping *c*. These parameters were determined using the data obtained from a simple dropped tire bouncing experiment explained below.

The system of a solid tire fitted to a wheel can be represented by a mass *m*, viscous damping *c*, and linear stiffness *k*. In our approach, the bouncing behaviour of a dropped tire is modelled using the equation of motion, similarly to the case of bouncing ball [7] as shown schematically in Fig. 1.

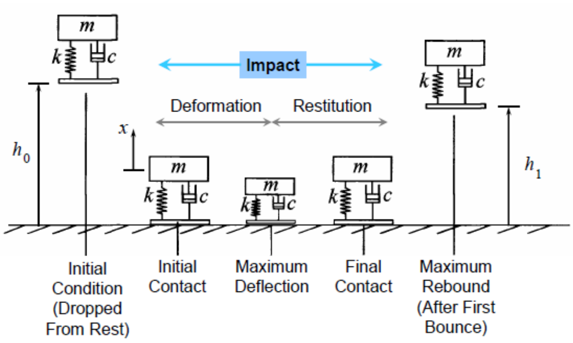

Fig.1 A mass-spring-damper model of a bouncing system [7]

Assuming no aerodynamic drag, when the tire is not in contact with the ground the equation of motion can be written as

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$$m\ddot{x} = -mg \tag{1}$$

The initial conditions are $x(0) = h_0$ and $\dot{x}(0) = 0$ for a tire released from rest from height h_0, as shown in Fig.1 ,where x is measured vertically up from the tire's center of mass with $x = 0$ corresponding to initial contact i.e., assuming that the deformation at the contact of the tire and the ground is very small and can be neglected. And when the tire is in contact with the ground, deformation and restitution occur. The equation of motion is then:

$$m\ddot{x} + c\dot{x} + kx = -mg \tag{2}$$

The initial conditions are $x(0) = 0$ and $\dot{x}(0) = -v_0$ where v_0 is the velocity of the ball just prior to contact with the ground. Solving Eq.(2) gives:

$$x(t) = \left[\frac{cg - 2kv_0}{2k\omega_d}\sin(\omega_d t) + \frac{mg}{k}\cos(\omega_d t)\right] \times \exp\left(\frac{-c}{2m}t\right) - \frac{mg}{k} \tag{3}$$

where ω_d is the damped natural frequency. The coefficient of restitution, ε is the ratio of speed of separation to speed of approach at the first bounce can be written as:

$$\varepsilon = \left|\frac{\dot{x}(\Delta T)}{\dot{x}(0)}\right| = \left|\frac{v_1}{v_0}\right| \tag{4}$$

where ΔT is the contact time and v_1 is the rebound speed just after leaving the ground.

By imposing the assumption $\frac{mg}{k} << \left|\frac{v_0}{\omega_d}\right|$, the coefficient of restitution can then be related to the contact time, ΔT , the stiffness, k, and damping, c as:

$$\varepsilon = \exp\left(-\frac{c\Delta T}{2m}\right) \tag{5}$$

$$k = m\left(\frac{\pi}{\Delta T}\right)^2\left[1 + \left(\frac{\ln\varepsilon}{\pi}\right)^2\right] \tag{6}$$

Contact time and the coefficient of restitution are assumed to be constant at each bounce and the total number of bounce is n, the contact time can be expressed as

$$\Delta T = \varepsilon^n\sqrt{\frac{h_0}{g}\left(\pi^2 + (\ln\varepsilon)^2\right)} \tag{7}$$

Contact time ΔT and the total number of bounce n can be found from the experimental data and, with known mass m and height h_0, the restitution confident ε can be determined. Damping, c and stiffness, k, and can then be evaluated from Equations (5) and (6), respectively.

In our experiment set up, a solid tire sample was fitted to a wheel and was dropped vertically from the height of 25 cm. measured from lowest point of the tire to the ground and allowed to freely bounce until rest. Tests were performed for a front tire of size 7.00-12, weighing 58.1 kg (including wheel) and a size 6.00-9 rear tire, weighing 30 kg (including wheel). In the setup, the accelerometer (Bruel & Kjear Accelerometer Type 4513-B) was mounted on the wheel at the position shown in Fig.2 and was configured to only measure the vertical acceleration. The data logger used was Bruel & Kjear, Input Module 50 kHz 4 channels Type 3050-B-040 as shown in Fig.3.

Fig.2 the position of accelerometer mounting

Fig.3 vertical acceleration measurement equipment

The parameters of the front and the rear tire (mounted to wheels) evaluated are listed in Table. 1.

Table. 1 The parameters of the front and the rear tire

Tire	Parameters		
	Mass, m (kg)	Stiffness, k (MN/m)	Damping, c (kN/ms^{-1})
Front (Dia. 640 mm)	58.1	2.053	2.497
Rear (Dia. 510 mm)	30.0	1.570	1.661

A simple Matlab/Simulink model was constructed to simulate the motion of dropped-bouncing tire based on a mass-spring-damping system and the Eqs. (1) – (2) with the initial conditions and the tire parameters described. This is to validate the tire-wheel evaluated parameters and the assumptions used for the model. The results of acceleration at the probe position on the wheel obtained from the simulations for both front and rear tire-wheel are plotted and compared with the measured data as shown in Fig.4 and Fig.5 . It can be seen that the simulation results are in good agreement with the measured data.

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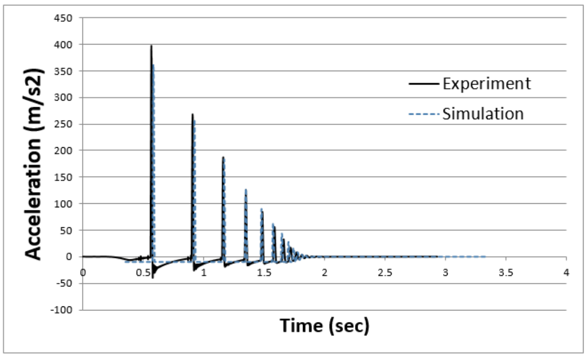

Fig.4 The vertical acceleration of the front tire obtained from the bouncing test and the Matlab/Simulink simulation

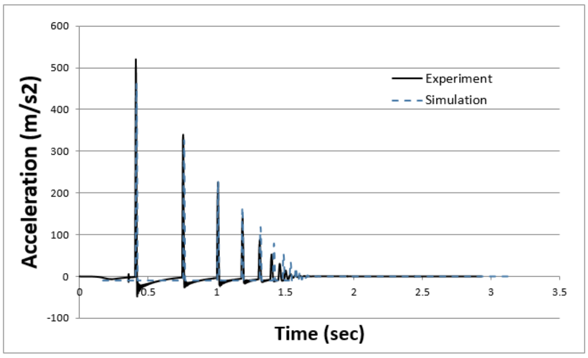

Fig.5 The vertical acceleration of the rear tire obtained from the bouncing test and the Matlab/Simulink simulation

3. Dynamic Simulation Model of Forklift truck

A dynamic simulation model of a TCM25 Forklift truck was constructed to analyses the vibration behavior of the vehicle while running over obstacles made of flat bars of width 25mm, thickness 4.5mm at the speed of 0.92 m/s. The schematic diagram of the truck and the test track are as depicted in Fig 6.

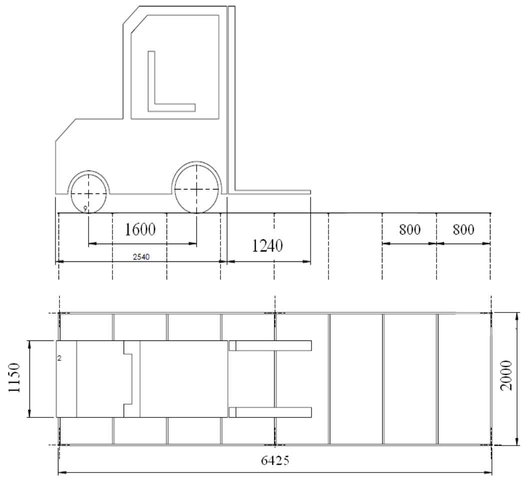

Fig.6 Schematic diagram of forklift truck and testing track (dimensions are in mm.)

A half car dynamic model with 4-DOF was applied to represent the dynamic vehicle of a forklift as shown in Fig.7 . It is assumed that the sprung mass was regarded as the rigid body. The suspension and tires are considered as spring and damper systems. Only the vertical vibration is assumed to influence the comfort of the ride, thus the transverse vibration and the horizontal vibration are not considered. The state variables and input descriptions are provided in Table.2 .

Table. 2 States variables and Input description

Symbols	Description	Units
m_{wf}	Mass of front wheel	kg
m_{wr}	Mass of rear wheel	kg
m_b	Mass of body truck	kg
θ	Pitch angle	Degree
k_{wf}	Front wheel stiffness	N/m
k_{wr}	Rear wheel stiffness	N/m
k_{bf}	Body front stiffness	N/m
k_{br}	Body rear stiffness	N/m
c_{bf}	Body front damping coefficient	N-s/m
c_{br}	Body rear damping coefficient	N-s/m
c_{wf}	Front wheel damping coefficient	N-s/m
c_{wr}	Rear wheel damping coefficient	N-s/m
l_f	distance between center of mass and front wheel	m
l_r	distance between center of mass and rear wheel	m
x_{rf}	Vertical displacement of road excited on front wheel	m
x_{rr}	Input vertical displacement of road on the rear wheel	m
x_{wf}	Input vertical displacement on the front wheel	m
x_{wr}	Vertical displacement of rear wheel	m
x_{bf}	Vertical displacement of body at front	m
x_{br}	Vertical displacement of body at rear	m

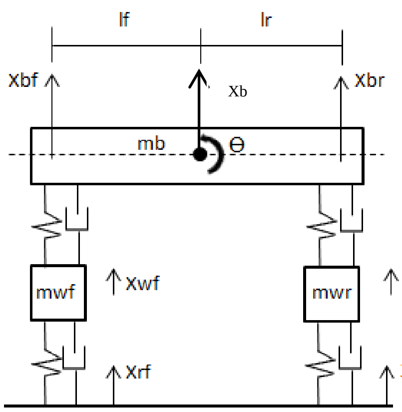

Fig.7 4-DOFS half car dynamic model of forklift truck

The equations of motion for the vehicle body and the front/rear wheels are given by

$$k_{bf}(x_{bf}-x_{wf})+c_{bf}(\dot{x}_{bf}-\dot{x}_{wf})+k_{br}(x_{br}-x_{wr})+c_{br}(\dot{x}_{br}-\dot{x}_{wr})=-m_b\ddot{x}_b \quad (8)$$

$$\begin{aligned}&k_{bf}(x_b-l_f\theta)l_f-k_{bf}x_{wf}l_f+c_{bf}(\dot{x}_b-l_f\dot{\theta})l_f-c_{bf}\dot{x}_{wf}l_f\\&-k_{br}(x_b+l_r\theta)l_r+k_{br}x_{wr}l_r-c_{br}(\dot{x}_b+l_r\dot{\theta})l_r+c_{br}\dot{x}_{wr}l_r=I\ddot{\theta}\end{aligned} \quad (9)$$

$$\begin{aligned}&k_{bf}(x_{bf}-x_{wf})+c_{bf}(\dot{x}_{bf}-\dot{x}_{wf})-k_{wf}(x_{wf}-x_{rf})\\&-c_{wf}(\dot{x}_{wf}-\dot{x}_{rf})=m_{wf}\ddot{x}_{wf}\end{aligned} \quad (10)$$

$$\begin{aligned}&k_{br}(x_{br}-x_{wr})+c_{br}(\dot{x}_{br}-\dot{x}_{wr})-k_{wr}(x_{wr}-x_{rr})\\&-c_{wr}(\dot{x}_{wr}-\dot{x}_{rr})=m_{wr}\ddot{x}_{wr}\end{aligned} \quad (11)$$

The equations of motion can also be written in state space matrix form as:

$$\begin{bmatrix}[C] & [M]\\ [M] & [0]\end{bmatrix}\{\dot{Y}\}+\begin{bmatrix}[K] & [0]\\ [0] & -[M]\end{bmatrix}\{Y\}=\begin{bmatrix}f\\0\end{bmatrix} \quad (12)$$

$$[\bar{A}]\{\dot{Y}\}+[\bar{B}]\{Y\}=[b]\{u\} \quad (13)$$

$$\{\dot{Y}\}=-[\bar{A}]^{-1}[\bar{B}]\{Y\}+[\bar{A}]^{-1}[b]\{u\} \quad (14)$$

where

$$[M]=\begin{bmatrix}I & 0 & 0 & 0\\ 0 & m_b & 0 & 0\\ 0 & 0 & m_{wf} & 0\\ 0 & 0 & 0 & m_{wr}\end{bmatrix}$$

$$[C]=\begin{bmatrix}(c_{bf}\times l_f^2)+(c_{br}\times l_r^2) & (-c_{bf}\times l_f)+(c_{br}\times l_r) & c_{bf}\times l_f & -c_{br}\times l_r\\ (-c_{bf}\times l_f)+(c_{br}\times l_r) & c_{bf}+c_{br} & -c_{bf} & -c_{br}\\ c_{bf}\times l_f & -c_{bf} & c_{bf}+c_{wf} & 0\\ -c_{br}\times l_r & -c_{br} & 0 & c_{br}+c_{wr}\end{bmatrix}$$

$$[K]=\begin{bmatrix}(k_{bf}\times l_f^2)+(k_{br}\times l_r^2) & (-k_{bf}\times l_f)+(k_{br}\times l_r) & k_{bf}\times l_f & -k_{br}\times l_r\\ (-k_{bf}\times l_f)+(k_{br}\times l_r) & k_{bf}+k_{br} & -k_{bf} & -k_{br}\\ k_{bf}\times l_f & -k_{bf} & k_{bf}+k_{wf} & 0\\ -k_{br}\times l_r & -k_{br} & 0 & k_{br}+k_{wr}\end{bmatrix}$$

$$[b]=\begin{bmatrix}0 & 0 & 0 & 0\\ 0 & 0 & 0 & 0\\ c_{wf} & 0 & k_{wf} & 0\\ 0 & c_{wr} & 0 & k_{wr}\end{bmatrix}$$

$$\{u\}=\begin{Bmatrix}\dot{x}_{rf}\\ \dot{x}_{rr}\\ x_{rf}\\ x_{rr}\end{Bmatrix},\ \{x\}=\begin{Bmatrix}\theta\\ x_b\\ x_{wf}\\ x_{wr}\end{Bmatrix},\ \{Y\}=\begin{Bmatrix}\{x\}\\ \{\dot{x}\}\end{Bmatrix},\ \{\dot{Y}\}=\begin{Bmatrix}\{\dot{x}\}\\ \{\ddot{x}\}\end{Bmatrix} \quad (15)$$

The above equations were implemented into Matlab/Simulink and solved to simulate the response of the vehicle to the excitation imposed by the obstacles. The displacement input to the front tire due to the forklift running over flat bar obstacles can be approximated to a rectangular pulse train profile and can be represented by Fourier series as

$$x_{rf}(t)=0.00013+\sum_{n=1}^{\infty}\left[\frac{0.009}{n\pi}\sin(0.0287n\pi)\cos(7.23nt)\right] \quad (16)$$

The input to the rear tire, x_{rr} is delayed by the time shift of 1.472 seconds.

The simulations were performed for case 1: the forklift carrying no load and case 2: with 1,130 kg load on the front carriage. The input parameters for the dynamic model for the two cases are listed in Table.3.

Table.3 Model parameters

Parameters	Magnitude
Forklift mass (Free loading)	3,720 kg
Forklift mass (with load)	4,850 kg
Front wheel mass	58.1 kg
Front wheel stiffness	2,052,917.617 N/m
Front wheel damping	2,497.282 N-s/m
Rear wheel mass	30 kg
Rear wheel stiffness	1,570,032.172 N/m
Rear wheel damping	1,661.452 N-s/m
Center of gravity of mass (Free loading)	0.52 m measured from center of front wheel
Center of gravity of mass (with load)	0.18 m measured from center of front wheel
Moment of Inertia (Free loading)	1,738.294 kg-m^2
Moment of Inertia (with load)	2,266.324 kg-m^2

The result of vertical acceleration at center of mass of the forklift obtained from the simulations in time domain are as shown in Fig.8 and Fig.9.

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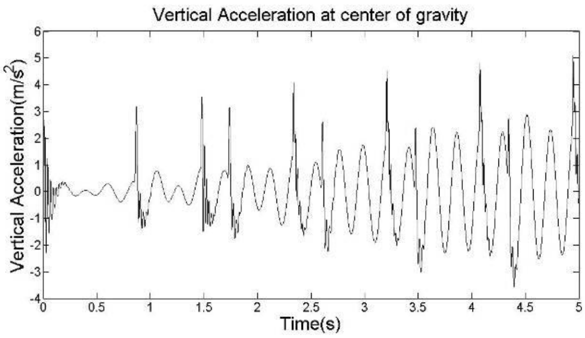

Fig.8 Acceleration Response at center of mass of the forklift body (no load carrying)

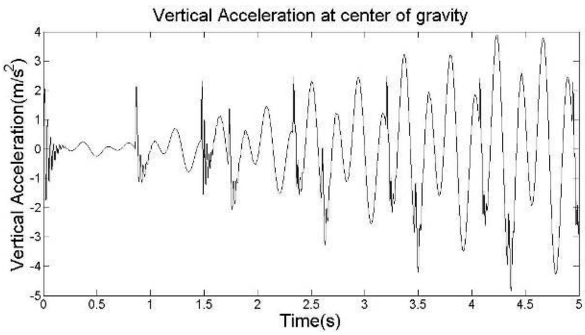

Fig.9 Acceleration Response at center of mass of forklift body (carrying load of 1,130 kg)

The simulated vertical acceleration time history at the center of mass of the forklift truck obtained from the two cases are converted to response in the frequency domain as shown in Fig.10 and Fig.11.

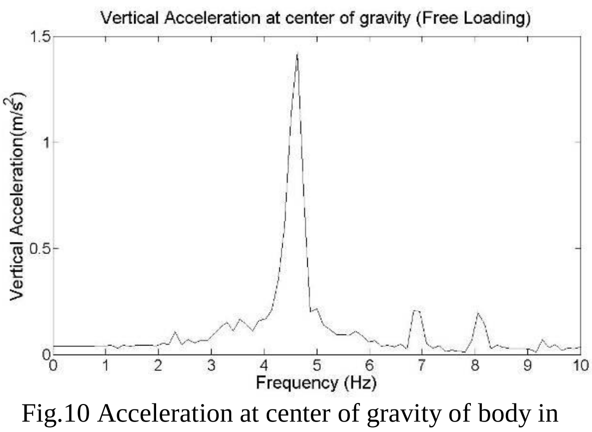

Fig.10 Acceleration at center of gravity of body in frequency domain (no load)

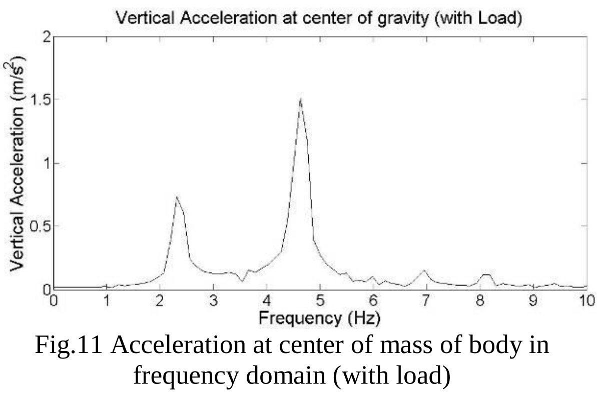

Fig.11 Acceleration at center of mass of body in frequency domain (with load)

From the simulation results, the highest vertical acceleration for both cases occurring in the range of 4-5 Hz with the maximum magnitude of 1.45 m/s2 for the forklift without load and 1.6 m/s2 for loaded case. Also there is a smaller peak of acceleration observed at the frequency range of 2-3 Hz for the loaded case. This is the effect of change in the mass and the position of the forklift's center of mass due to the carrying load. According to the recommended standard set by ISO (ISO2631-1, 1997) [1], the whole body vibrations transmitted to the human body between 2.5 and 5.5 Hz generate strong resonance in the vertebra of the neck and lumbar region and the exposure time should not be greater than the recommended and the acceleration magnitude should be kept below 0.5m/s^2 for a comfortable ride.

4. Parametric study of the influence of tires on the vibration behavior

The dynamic model described in previous section was used to investigate the effect of solid tire's properties on the vertical vibration response. The simulations were performed for case of forklift carrying no load. Firstly, the influence of the stiffness of the tire were investigated. Three different values of the stiffness were used for both the front and the rear tires, with the damping coefficients listed in Table 1, and the simulation results are shown in Fig.12. The effect of the damping were also studied. Three different values of the damping were used for both the front and the rear tires, with the stiffness listed in Table 1, and the simulation results are shown in Fig. 13.

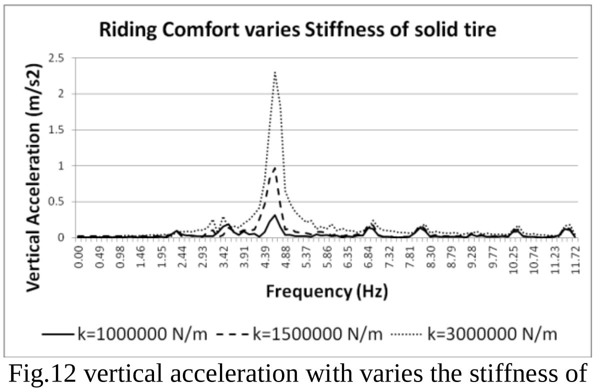

Fig.12 vertical acceleration with varies the stiffness of solid tire (constant damping of front wheel at 2,497.282 N-s/m and rear wheel at 1,661.452 N-s/m)

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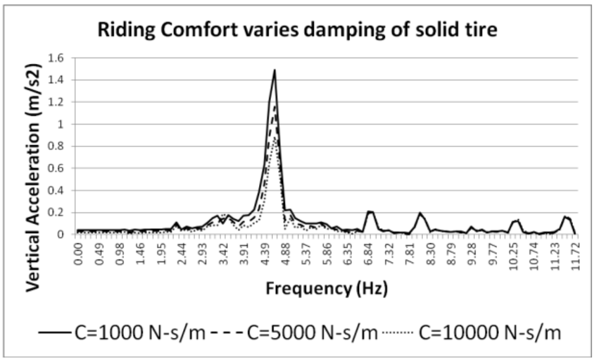

Fig.13 vertical acceleration with varies the damping of solid tire (constant stiffness of front wheel at 2,052,917.617 N/m and rear wheel at 1,570,032.172 N/m)

It can be seen that the increase of stiffness leads to the significant increase in vibration emission. In contrast, the increase in damping leads to the reduced vibration emission but not so drastically. In addition, it is not realistic to have high value of damping over 2000 Ns/m for conventional tires. Thus, for the improvement of riding comfort, the tires stiffness can be lowered to reduce vibration. However, it is to be noted that decreasing the stiffness may affect the forklift's controllability and stability.

5. Conclusions

In this work, a simple methodology for a quick and inexpensive investigation of the influence of solid tire properties on the mechanical vibration behavior of a forklift truck was proposed. The methodology consists of a simple method for determining the mechanical properties of solid tires from bouncing experiment and a simple numerical simulation model using Matlab/Simulink for analyzing the dynamic vibration behavior of the vehicle using 4- DOF half car model. It was demonstrated that the proposed methodology was sufficient for the study of the influence of solid tire properties on the vibration behavior of a forklift truck. The results obtained are useful for the quality improvement of solid tires towards a more comfortable ride.

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