Influence of Tire Parameters of a Semi-trailer Truck on Road Surface Friendliness

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Abstract –*The objective of this study is to analyze the influence of tire parameters of a semi-trailer truck such as stiffness and damping coefficients on road surface friendliness. A half-dynamic vehicle model with 12 degrees of freedom established under random road excitation and dynamic load coefficient (DLC) is chosen as an objective function which is simulated and calculated with Matlab/Simulink software. The effect of the tire stiffness and damping coefficients on DLC value is analyzed respectively. The influence of the stiffness and damping parameters of tires on DLC value is shown to be very obvious. The more the vehicle ride comfort and road surface friendliness decrease, the more the tire stiffness increases. Therefore, the study provides a theoretical basis for the dynamic system design of semi-trailer trucks.*

Key Words: Semi-trailer truck, Tire, Stiffness coefficient Damping coefficient, Road surface friendliness.

1. INTRODUCTION

Bridges and roads play an important role in transport systems. Therefore, investigating dynamic loads of heavy vehicles and roads is necessary for maintaining roads and bridges as well as solving problems in design of the vehicle dynamic systems. Based on the analysis of nonlinear geometric characteristics of the suspension systems and tires, a 3D nonlinear dynamic model of a typical heavy truck is established to evaluate the dynamic interaction between heavy vehicles and roads under the condition of random road surface roughness [1]. The three-dimensional (3D) vehicle-pavement coupled model is used to simulate the pavement dynamic loads induced by the vehicle-pavement interaction where both the vehicle vibration and pavement deformation are considered to analyze the effects of different operating conditions and vehicle parameters on pavement dynamic loads. The developed methodology can be used for further study on the vehicle-induced pavement response, and the loading information may be useful in analyzing vehicle-induced pavement damage [2]. To evaluate influence of heavy truck operating condition on dynamic load coefficient, a three-dimensional nonlinear dynamic model of heavy with 15 DOF (degree of freedom) is established according to Zhou Changfeng model for simulation and analysis [3] and the three-dimensional vehicle-pavement coupled model with 14 degrees of freedom is established for simulation and calculation. The influence of the different vehicle operating conditions on the dynamic tire loads, dynamic load coefficient (DLC) and road-friendliness which include road surfaces, vehicle speeds, vehicle loads are analyzed [4]. The 3D dynamic model for a semi-trailer truck with 14 degrees of freedom is developed to analyze the performance of two air suspension systems such as traditional and new air suspension systems for reducing the negative impacts on the road surface[5] and the full dynamic model of a heavy vehicle equipped with three different suspension systems including the hydro-pneumatic, rubber and air spring suspension systems is established to evaluate the effect of suspension characteristics on dynamic load coefficient (DLC)[6]. To analyze and evaluate the performance of the air suspension system of heavy trucks with semi-active fuzzy control, a three-dimensional nonlinear dynamical model of a typical heavy truck with 16-DOF (degree of freedom) is established on Matlab/Simulink software. The weighted root-mean-square (RMS) acceleration responses of the vertical driver's seat, the pitch and roll angle of the cab, and the dynamic load coefficient (DLC) are chosen as objective functions, and the air suspension system is optimized and analyzed by the semi-active fuzzy control algorithm when vehicles operate under different operation conditions[7]. The dynamic model with 8-DOF of a heavy truck is established, Matlab/Simulink software is used to simulate and calculate the objection functions under two types of step and random road surfaces. The FLC is then applied to control semi-active isolation systems of the vehicle including the controlled seat, cab, and vehicle [8]. The semi-active suspension system with a rolling lobe air spring is firstly modeled and a novel front axle vertical acceleration-based road prediction model is constructed. By adopting a sensor on the front axle, the road prediction model can be used to predredict more reliable road information for the rear wheel. After filtering useless signal noise, the proposed FPW (Fuzzy-wheelbase preview controller with wavelet denoising filter) can generate a noise-insensitive control damping force. Simulation results show that the ride quality, the road holding, the handling capability, the road friendliness, and the comprehensive performance of the semi-active air suspension with FPW perform with the traditional active suspension with PID-wheelbase preview controller (APP)[9].

In this study, a half-dynamic vehicle model with 12 degrees of freedom is established under random road excitation and dynamic load coefficient (DLC) is chosen as objective function which is simulated and calculated with Matlab/Simulink software. The tire parameters such as the stiffness and damping coefficients are analyzed respectively based on DLC values when vehicle moves on the ISO class B road surface at v=20 m/s and full load.

2. VEHICLE DYNAMIC MODEL

Full vehicle dynamic model

A 5- axle semi-trailer truck with dependent leaf spring suspension systems for all axles of vehicle is selected for analyzing the influence of tire parameters of vehicle on road surface friendliness. A half semi-trailer truck dynamic model with 12 degrees of freedom is established, as shown in Fig-1.

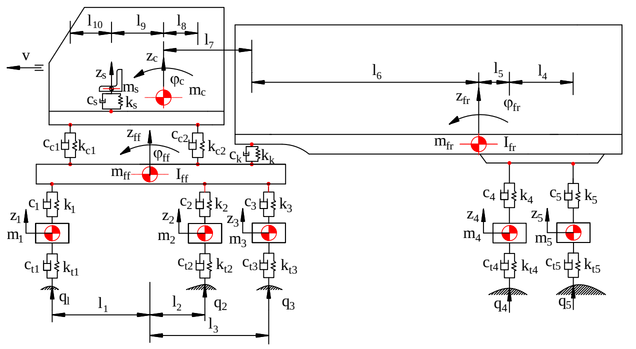


Fig-1: Articulated truck semi-trailer dynamic model

In Fig-1, k_{ti} and c_{ti} are the stiffness and damping coefficients of the tires, respectively; k_i and c_i are the stiffness and damping coefficients of the passive suspension systems of vehicle, respectively; k_{c1}, k_{c2}, k_s and c_{c1}, c_{c2}, c_s are the passive suspension systems of cab and driver seat respectively; m_i are the unsprung mass of the tractor and trailer axles, respectively; m_{ff} and m_{fr} are the sprung mass of the tractor and trailer bodies, respectively; m_c and m_s are the mass of the cab body and the driver seat; I_{ff}, I_{fr} and I_c are the mass moment of inertia of tractor, trailer and cab; z_i, z_{ff}, z_{fr}, z_c và z_s are the vertical displacements of the axles, tractor body, trailer body, cab and driver seat, respectively; φ_c, φ_{ff} and φ_{fr} are the pitch angle displacements of the cab, tractor and trailer bodies, respectively; l_j are the distances; v is the speed of vehicle ($i=1\div5$; $j=1\div10$).

Equations of motion: The equations of vehicle motion can be formulated in different ways such as Lagrange's equation, Newton-Euler equation, Jourdain's principle. However, in order to facilitate the description of vehicle dynamic systems using computer simulation, a combined method of the multi-body system theory and D'Alembert's principle is chosen for this study. The multi-body system theory is used to separate the system into subsystems which are linked by the force and moment equations. D'Alembert's principle is used to set up force and moment equations to describe vehicle dynamic subsystems.

The equations of motion for steering axle and drive axles for tractor are written as follows:

$$m_{a1}\ddot{z}_{a1}=\left[k_1\left(z_{b1}-l_1\varphi_{b1}-z_{a1}\right)+c_1\left(\dot{z}_{b1}-l_1\dot{\varphi}_{b1}-\dot{z}_{a1}\right)\right] -\left[k_{t1}\left(z_{a1}-q_1\right)+c_{t1}\left(\dot{z}_{a1}-\dot{q}_1\right)\right] \quad (1)$$

$$m_{a2}\ddot{z}_{a2}=\left[k_{a2}\left(z_{b1}+l_2\varphi_{b1}-z_{a2}\right)+c_2\left(\dot{z}_{b1}+l_2\dot{\varphi}_{b1}-\dot{z}_{a2}\right)\right] -\left[k_{t2}\left(z_{a2}-q_2\right)+c_{t2}\left(\dot{z}_{a2}-\dot{q}_2\right)\right] \quad (2)$$

$$m_{a3}\ddot{z}_{a3}=\left[k_{a3}\left(z_{b1}+l_3\varphi_{b1}-z_{a3}\right)+c_3\left(\dot{z}_{b1}+l_3\dot{\varphi}_{b1}-\dot{z}_{a3}\right)\right] -\left[k_{t3}\left(z_{a3}-q_3\right)+c_{t3}\left(\dot{z}_{a3}-\dot{q}_3\right)\right] \quad (3)$$

The equations of motion for semitrailer axles are written as follows:

$$m_{a4}\ddot{z}_{a4}=\left[k_{a4}\left(z_{b2}+l_5\varphi_{b2}-z_{a4}\right)+c_4\left(\dot{z}_{b2}+l_5\dot{\varphi}_{b2}-\dot{z}_{a4}\right)\right] -\left[k_{t4}\left(z_{a4}-q_4\right)+c_{t4}\left(\dot{z}_{a4}-\dot{q}_4\right)\right] \quad (4)$$

$$m_{a5}\ddot{z}_{a5}=\left[k_{a5}\left(z_{b2}+l_4\varphi_{b2}-z_{a5}\right)+c_5\left(\dot{z}_{b2}+l_4\dot{\varphi}_{b2}-\dot{z}_{a5}\right)\right] -\left[k_{t5}\left(z_{a5}-q_5\right)+c_{t5}\left(\dot{z}_{a5}-\dot{q}_5\right)\right] \quad (5)$$

The equations of motion for tractor vertical and pitch motions are written as follows:

$$\begin{aligned}
m_{b1}\ddot{z}_{b1}=&\left[k_{c1}\left(z_c-l_9\varphi_c-l_{10}\varphi_c-z_{b1}+l_{11}\varphi_{b1}\right)\right.\\
&\left.+c_{c1}\left(\dot{z}_c-l_9\dot{\varphi}_c-l_{10}\dot{\varphi}_c-\dot{z}_{b1}+l_{11}\dot{\varphi}_{b1}\right)\right]\\
&+\left[k_{c2}\left(z_c+l_8\varphi_c-z_{b1}-l_{12}\varphi_{b1}\right)+c_{c2}\left(\dot{z}_c+l_8\dot{\varphi}_c-\dot{z}_{b1}-l_{12}\dot{\varphi}_{b1}\right)\right]\\
&+\left[k_k\left(z_{b2}-l_6\varphi_{b2}-z_{b1}-l_7\varphi_{b1}\right)+c_k\left(\dot{z}_{b2}-l_6\dot{\varphi}_{b2}-\dot{z}_{b1}-l_7\dot{\varphi}_{b1}\right)\right]\\
&-\left[k_1\left(z_{b1}-l_1\varphi_{b1}-z_{a1}\right)+c_1\left(\dot{z}_{b1}-l_1\dot{\varphi}_{b1}-\dot{z}_{a1}\right)\right]\\
&-\left[k_{a2}\left(z_{b1}+l_2\varphi_{b1}-z_{a2}\right)+c_2\left(\dot{z}_{b1}+l_2\dot{\varphi}_{b1}-\dot{z}_{a2}\right)\right]\\
&-\left[k_{a3}\left(z_{b1}+l_3\varphi_{b1}-z_{a3}\right)+c_3\left(\dot{z}_{b1}+l_3\dot{\varphi}_{b1}-\dot{z}_{a3}\right)\right]
\end{aligned} \quad (6)$$

$$\begin{aligned}
I_{b1}\ddot{\varphi}_{b1}=&\left[k_1\left(z_{b1}-l_1\varphi_{b1}-z_{a1}\right)+c_1\left(\dot{z}_{b1}-l_1\dot{\varphi}_{b1}-\dot{z}_{a1}\right)\right]l_1\\
&+\left[k_{c2}\left(z_c+l_8\varphi_c-z_{b1}-l_{12}\varphi_{b1}\right)+c_{c2}\left(\dot{z}_c+l_8\dot{\varphi}_c-\dot{z}_{b1}-l_{12}\dot{\varphi}_{b1}\right)\right]l_{12}\\
&-\left[k_{a2}\left(z_{b1}+l_2\varphi_{b1}-z_{a2}\right)+c_2\left(\dot{z}_{b1}+l_2\dot{\varphi}_{b1}-\dot{z}_{a2}\right)\right]l_2\\
&-\left[k_{a3}\left(z_{b1}+l_3\varphi_{b1}-z_{a3}\right)+c_3\left(\dot{z}_{b1}+l_3\dot{\varphi}_{b1}-\dot{z}_{a3}\right)\right]l_3\\
&-\left[\left(z_c-l_9\varphi_c-l_{10}\varphi_c-z_{b1}+l_{11}\varphi_{b1}\right)\right.\\
&\left.+c_{c1}\left(\dot{z}_c-l_9\dot{\varphi}_c-l_{10}\dot{\varphi}_c-\dot{z}_{b1}+l_{11}\dot{\varphi}_{b1}\right)\right]l_{11}\\
&-\left[k_k\left(z_{b2}-l_6\varphi_{b2}-z_{b1}-l_7\varphi_{b1}\right)+c_k\left(\dot{z}_{b2}-l_6\dot{\varphi}_{b2}-\dot{z}_{b1}-l_7\dot{\varphi}_{b1}\right)\right]l_7
\end{aligned} \quad (7)$$

The equations of motion for semi-trailer vertical and pitch motions are written as follows:

$$\begin{aligned}
m_{b2}\ddot{z}_{b2}=&-\left[k_k\left(z_{b2}-l_6\varphi_{b2}-z_{b1}-l_7\varphi_{b1}\right)\right.\\
&\left.+c_k\left(\dot{z}_{b2}-l_6\dot{\varphi}_{b2}-\dot{z}_{b1}-l_7\dot{\varphi}_{b1}\right)\right]\\
&-\left[k_{a4}\left(z_{b2}+l_5\varphi_{b2}-z_{a4}\right)+c_4\left(\dot{z}_{b2}+l_5\dot{\varphi}_{b2}-\dot{z}_{a4}\right)\right]\\
&-\left[k_{a5}\left(z_{b2}+l_4\varphi_{b2}-z_{a5}\right)+c_5\left(\dot{z}_{b2}+l_4\dot{\varphi}_{b2}-\dot{z}_{a5}\right)\right]
\end{aligned} \quad (8)$$

$$\begin{aligned} I_{b2}\ddot{\varphi}_{b2} = &\Big[k_k\left(z_{b2}-l_6\varphi_{b2}-z_{b1}-l_7\varphi_{b1}\right) \\ &+c_k\left(\dot{z}_{b2}-l_6\dot{\varphi}_{b2}-\dot{z}_{b1}-l_7\dot{\varphi}_{b1}\right)\Big]l_7 \\ &-\Big[k_{a4}\left(z_{b2}+l_5\varphi_{b2}-z_{a4}\right)+c_4\left(\dot{z}_{b2}+l_5\dot{\varphi}_{b2}-\dot{z}_{a4}\right)\Big]l_5 \\ &-\Big[k_{a5}\left(z_{b2}+l_4\varphi_{b2}-z_{a5}\right)+c_5\left(\dot{z}_{b2}+l_4\dot{\varphi}_{b2}-\dot{z}_{a5}\right)\Big]l_4 \end{aligned} \tag{9}$$

The equations of motion for cab body vertical and pitch motions are written as follows:

$$\begin{aligned} m_c\ddot{z}_c = &\Big[k_s\left(z_c-l_9\varphi_c-z_s\right)+c_s\left(\dot{z}_c-l_9\dot{\varphi}_c-\dot{z}_s\right)\Big] \\ &-\Big[k_{c1}\left(z_c-l_9\varphi_c-l_{10}\varphi_c-z_{b1}+l_{11}\varphi_{b1}\right) \\ &k_{c1}\left(z_c-l_9\varphi_c-l_{10}\varphi_c-z_{b1}+l_{11}\varphi_{b1}\right)\Big] \\ &-\Big[k_{c2}\left(z_c+l_8\varphi_c-z_{b1}-l_{12}\varphi_{b1}\right) \\ &+c_{c2}\left(\dot{z}_c+l_8\dot{\varphi}_c-\dot{z}_{b1}-l_{12}\dot{\varphi}_{b1}\right)\Big] \end{aligned} \tag{10}$$

$$\begin{aligned} I_c\ddot{\varphi} = &\Big[k_{c1}\left(z_c-l_9\varphi_c-l_{10}\varphi_c-z_{b1}+l_{11}\varphi_{b1}\right) \\ &c_{c1}\left(\dot{z}_c-l_9\dot{\varphi}_c-l_{10}\dot{\varphi}_c-\dot{z}_{b1}+l_{11}\dot{\varphi}_{b1}\right)\Big]\left(l_9+l_{10}\right) \\ &-\Big[k_{c2}\left(z_c+l_8\varphi_c-z_{b1}-l_{12}\varphi_{b1}\right)+c_{c2}\left(\dot{z}_c+l_8\dot{\varphi}_c-\dot{z}_{b1}-l_{12}\dot{\varphi}_{b1}\right)\Big]l_8 \\ &-\Big[k_s\left(z_c-l_9\varphi_c-z_s\right)+c_s\left(\dot{z}_c-l_9\dot{\varphi}_c-\dot{z}_s\right)\Big]l_9 \end{aligned} \tag{11}$$

The equations of motion for seat vertical motion are written as follows:

$$m_s\ddot{z}_s = -\Big[k_s\left(z_c-l_9\varphi_c-z_s\right)+c_s\left(\dot{z}_c-l_9\dot{\varphi}_c-\dot{z}_s\right)\Big] \tag{12}$$

Road Surface Roughness

Surface roughness plays an important role in evaluating the dynamic interaction between vehicles and road. It is simulated in space domain and acts as an input to the vehicle-road model. The road surface roughness irregularities can be represented with a normal stationary argotic random process described by its Power Spectral Density (PSD). According to the International Standards Organization (ISO) 8608 [11], PSD of road roughness can be defined as Eq. (2):

$$G_q(n) = G_q(n_0)\left(\frac{n}{n_0}\right)^{-w} \tag{13}$$

n is spatial frequency in m^{-1}, n_0 is reference spatial frequency with a value of $0.1m^{-1}$, $G_q(n_0)$ is PSD value for reference spatial frequency in m^3, w is termed waviness, and reflects approximate frequency structure of the road profile, commonly taken as w=2. The classification of road roughness is based on the index of International Organization for Standardization ISO 8608. The ISO has proposed road roughness classification from class A - very good to class H - very poor, according to different values of $G_q(n_0)$. The road profile is generated as the sum of a series of harmonics:

$$q(t) = \sum_{k=1}^{N}\sqrt{2G_q\left(n_k\right)\Delta n}\cos\left(2\pi n_k t+\phi_k\right) \tag{14}$$

ϕ_k is the random phase uniformly distributed from 0 to2π; $G_q(n_k)$ is the power spectral density (PSD) function (m^3/cycle) for the road surface elevation; n_k is the wave number (cycle/m). In order to solve Eq.9, a program is written in Matlab with different road surface data according to the standard ISO 8068[11]. The results of the typical road surface roughness are shown in Fig-2.

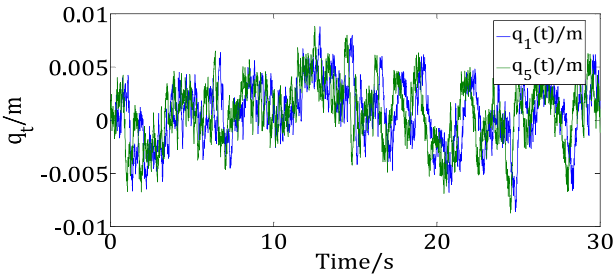


Fig- 2: Road surface roughness according to the standard ISO 8068 class B

3. DYNAMIC LOAD COEFFICIENT

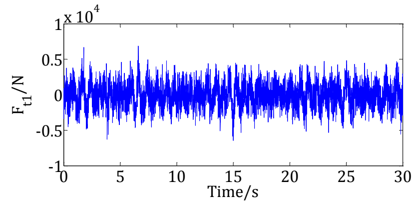
Dynamic tire load would lead to stress and strain of road surface. The long-term accumulation of road surface plastic deformation causes the destruction of roads, such as cracks and rutting. In this paper, in order to analyze the performance of the air-spring suspension systems compared with the leaf-spring suspension systems of semi-trailer truck, the DLC is chosen as objective function which is defined by a ratio of the root mean square of the vertical dynamic tire force over static load [1-7] as follows:

$$DLC = \frac{F_{t,rms}}{F_s} \tag{15}$$

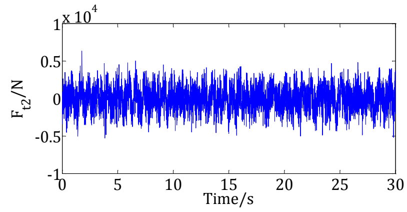
$F_{t,rms}$ and F_s are the root mean square of the vertical dynamic and the static tire force. The value of the DLC is in range of 0.05 to 0.3 under normal operating conditions. It may reach zero when the wheels move on a special smooth road or increase up to 0.4 when the tires of the axles spend a significant proportion of their time disconnecting the road surface [12].

4. SIMULATION AND DISCUSSION

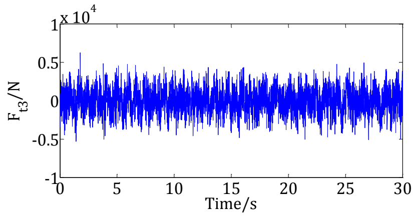
In order to analyze the vertical dynamic tire loads of a semi-trailer truck acting on road surface equipped with leaf spring suspension systems, the mathematical model of the proposed half-vehicle semi-trailer suspension is developed in the MATLAB/Simulink to solve the truck differential equations in section 2 with a set of parameters of the articulated truck semi-trailer by the references [10].



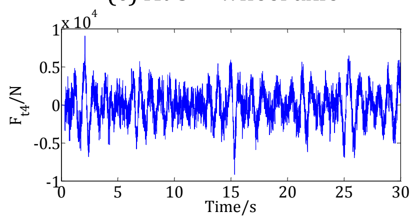
(a) At 1st wheel axle



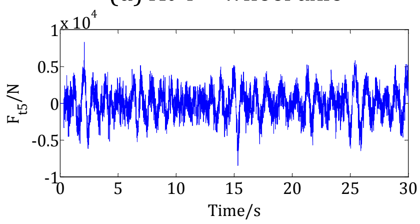
(b) At 2nd wheel axle



(c) At 3rd wheel axle



(d) At 4th wheel axle



(e) At 5th wheel axle

Fig-3. The vertical dynamic tire forces acting on road surface

When vehicle moves on the ISO class B road surface at v=20 m/s and full load, the simulation results of the vertical dynamic tire loads acting on road surface at 1st, 2nd, 3rd, 4th and 5th axles are shown in Fig-3 and the DLC values at all axles are shown in Tab.1

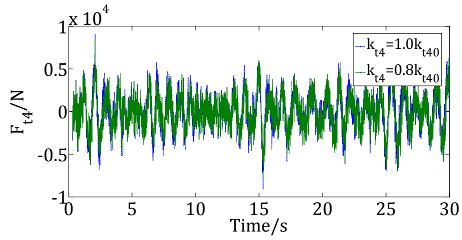
Tab-1. DLC values at all axles

indexes	DLC_{t1}	DLC_{t2}	DLC_{t3}	DLC_{t4}	DLC_{t5}
DLC	0.022	0.029	0.027	0.049	0.044

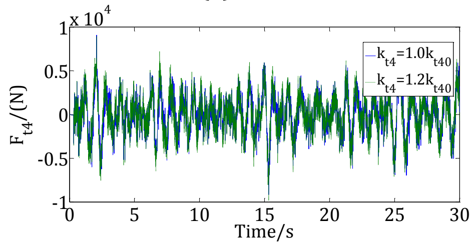
From the results in Tab.1, it is shown that the values of DLC at 1st, 2nd, 3rd, 4th and 5th axles of vehicle are satisfied in the allowed range (in section 3). The influence of tire parameters of a semi-trailer truck such as stiffness and damping coefficients on DLC values at all axles of wheel is being presented in the following in sections.

Effect of tire parameters

Effect of stiffness coefficients of tires: To analyze the effect of stiffness coefficients of tires on *DLC* values, the values of the stiffness coefficients of tires k_t=[0.6, 0.8, 1.0, 1.2, 1.4] xk_{t0}, where $k_{t0}=[k_{t1}, k_{t2}, k_{t3}, k_{t4}]^T$ are analyzed when vehicle moves the ISO class B road surface at v=20 m/s and full load, where k_{t0} are used to designate the tire stiffness coefficients in the reference document[10]. The influence of the tire stiffness coefficients including case 1: $k_t=0.8k_{t0}$ and $k_t=1.0k_{t0}$ and cas2: $k_t=1.0k_{t0}$ and $k_t=1.2k_{t0}$ on *DLC* values of wheels at 4th axle of vehicle are shown in Fig-4. From Fig-4 we can see that the tire stiffness coefficient increases from $0.8k_{t0}$ to $1.2k_{t0}$, the DLC value of wheel at the 4th axle of vehicle increases from 3.8 to 7.4%.



(a) Case 1



(b) Case 2

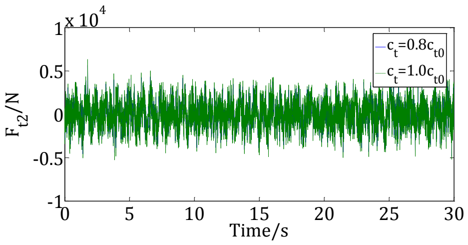
Fig-4. Comparing the vertical dynamic tire forces acting on road surface at 4th axle with case 1 and case 2

Tab-2. DLC values at all axles

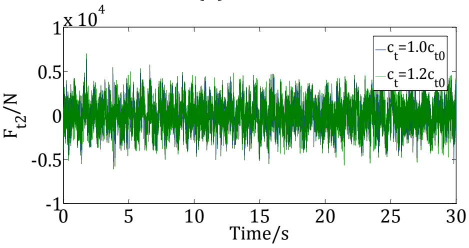
indexes	DLC_{t1}	DLC_{t2}	DLC_{t3}	DLC_{t4}	DLC_{t5}
$k_t=0.6k_{t0}$	0.019	0.026	0.024	0.039	0.035
$k_t=0.8k_{t0}$	0.021	0.027	0.026	0.045	0.041
$k_t=1.0k_{t0}$	0.022	0.029	0.027	0.049	0.044
$k_t=1.2k_{t0}$	0.023	0.031	0.029	0.052	0.046
$k_t=1.4k_{t0}$	0.024	0.032	0.030	0.054	0.048

From the results in Tab.2, it is shown that the tire stiffness coefficient increases, the values of DLC at 1st, 2nd, 3rd, 4th and 5th axles increase, which means that the road-friendliness of vehicle reduce. However, the tire stiffness coefficient (tire pressure) is not in accordance with the manufacturer's regulations, it will affect the driving dynamics and the economics of vehicle.

Effect of damping coefficients of tires: To analyze the influence of the tire damping coefficients on DLC values, case 3: $c_t=0.8c_{t0}$ and $c_t=1.0c_{t0}$ and cas4: $c_t=1.0c_{t0}$ and $c_t=1.2c_{t0}$ on DLC values of wheels at 2nd axle of vehicle are shown in Fig-5.



(a) Case 3



(a) Case 4

Fig-5. Comparing the vertical dynamic tire forces acting on road surface at 2nd axle with case 3 and case 4.

Fig-5 reveals that the tire damping coefficient increases from $0.8c_{t0}$ to $1.2c_{t0}$, the DLC value of wheel at the 4th axle of vehicle increases from 7.4 to 10.3%. However, the damping coefficient value of tire prevents intimate relationship with the value of the tire pressure that cannot be changed.

5. CONCLUSIONS

In this study, a half-vehicle dynamic model of a semi-trailer truck is established for analyzing the influence of tire parameters of a semi-trailer truck on road surface friendliness when vehicle moves on the ISO class B road surface at v=20 m/s and full load. The major conclusions can be drawn from the analysis results as follows:

i) The tire stiffness coefficient increases from $0.8k_{t0}$ to $1.2k_{t0}$, the DLC value of wheel at the 4th axle of vehicle increases from 3.8 to 7.4%. The tire stiffness coefficient increases, the values of DLC at 1st, 2nd, 3rd, 4th and 5th axles increase, which means the road-friendliness of vehicle reduced.
ii) The tire damping coefficient increases from $0.8c_{t0}$ to $1.2c_{t0}$, the DLC value of wheel at the 4th axle of vehicle increases from 7.4 to 10.3%.
iii) The optimal design parameters of tire is chosen because it is necessary in improving road surface friendliness, vehicle ride comfort as well as reducing vehicle noise.

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