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ANALYSIS OF DYNAMIC WHEEL LOADS OF A SEMI-TRAILER TRUCK WITH AIR-SPRING AND LEAF- SPRING SUSPENSION SYSTEMS

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ABSTRACT

In order to analyze the vertical dynamic tire loads of a semi-trailer truck acting on road surface equipped with air- and leaf- spring suspension systems, a half- vehicle dynamic model with 12 degrees of freedom is developed under random road excitation. A dynamic load coefficient (DLC) is chosen as objective function and Matlab/Simulink software is used to simulate and analyze the vehicle dynamic model. The performance dynamic wheel load of both air- and leaf- spring suspension systems are analyzed under different operating conditions such as different vehicle speeds, road surface roughness and vehicle loads. The obtained results indicate that the air-spring suspension systems reduced the dynamic tire loads acting on road surface better than the leaf-spring suspension systems under all operating conditions. Especially, the DLC values at all axles with air-spring suspension systems are significantly reduced by 3.72%, 55.91%, 50.53%, 31.47% and 25.45% in comparison with the leaf-spring suspension systems, respectively when vehicle moves on the ISO class B road surface roughness at v=20 m/s and full load.

Keywords: semi-trailer truck, air spring, leaf spring, suspension system, dynamic load coefficient (DLC), and road friendliness.

1. INTRODUCTION

The vehicle suspension system has an important role in reducing the negative effects of vehicle vibration- the analysis affects the suspension parameters on vehicle ride comfort and the road surface has been interested by many researchers in the world. The effects of suspension parameters on tire dynamic forces and dynamic load coefficient (DLC) using three-dimensional vehicle models were presented and analyzed in [1-3]. The effects of driving conditions as well as suspension parameters on dynamic load-sharing and road-friendliness of the semi-trailer were analyzed in [4]. To reduce the negative impact on the road surface, an optimum concept to design "road-friendly" vehicles with the recognition of pavement loads as a primary objective function of vehicle suspension design was presented [5]. The optimum design parameters of the suspension systems were searched by a genetic algorithm [6].

Air suspension system is widely used in heavy vehicles due to its nonli ear characteristics of stiffness. The nonlinear stiffness of air spring is determined by means of thermodynamic and fluid dynamic constitutions [7]. Two different air spring models (classic air spring; dynamic air spring model) were used for analyzing the performance of suspension types based on a 2-DOF quarter mathematical model and the obtained results indicated that the values of the body acceleration, suspension deflection and dynamic tire load with the dynamic air spring model reduced respectively, which provides more comfort and friendly performances comparing to the classic air spring model [8]. A 3D dynamic model for a semi-trailer truck with 14 degrees of freedom was developed to analyze the performance of two air suspension systems (the traditional and new air springs) and the obtained results indicated that the values of the dynamic load coefficient (DLC) of wheels at all axles of vehicle with the new air spring reduced respectively comparing to the traditional air spring when vehicle moves on the different road surface conditions [9]. Similarly, a 3D dynamic model with 15 d.o.f. was used to analyze the performance of the hydro-pneumatic suspension system compared to two others (rubber and air springs) and the obtained results indicated that the values of the dynamic load coefficient (DLC) of wheels at all axles of vehicle with the hydro-pneumatic suspension system reduced respectively comparing to two others when vehicle operates under the different conditions [6].

The major goal of this study is to develop the half - vehicle dynamic model of a semi-trailer truck with 12 degrees of freedom based on Spivey C model [7] equipped with air- and leaf- spring suspension systems for analyzing the vertical dynamic tire loads of the semi-trailer truck acting on road surface. The dynamic models of air- spring suspension systems at 2nd, 3rd, 4th and 5thaxles are proposed to compare its performance with leaf- spring suspension systems in research literature [7]. In order to analyze the effect of dynamic wheel loadings on the road surface, a dynamic load coefficient (DLC) is chosen as objective function. Matlab/Simulink software is used to simulate and determine the values of objective function. The effective reduction of dynamic tire loads of suspension systems acting on road surface was respectively analyzed when operates under different conditions.

2. VEHICLE DYNAMIC MODEL

2.1 Full vehicle dynamic model

A 5- axle semi-trailer truck with a dependent leaf spring suspension system for the front steer axle and the rear axles of tractor and trailer is selected in this paper. A half semi-trailer truck dynamic model with 12 degrees of freedom was developed based on Spivey C model [11] for analyzing the vertical dynamic tire loads acting on pavement, as shown in Figure-1.

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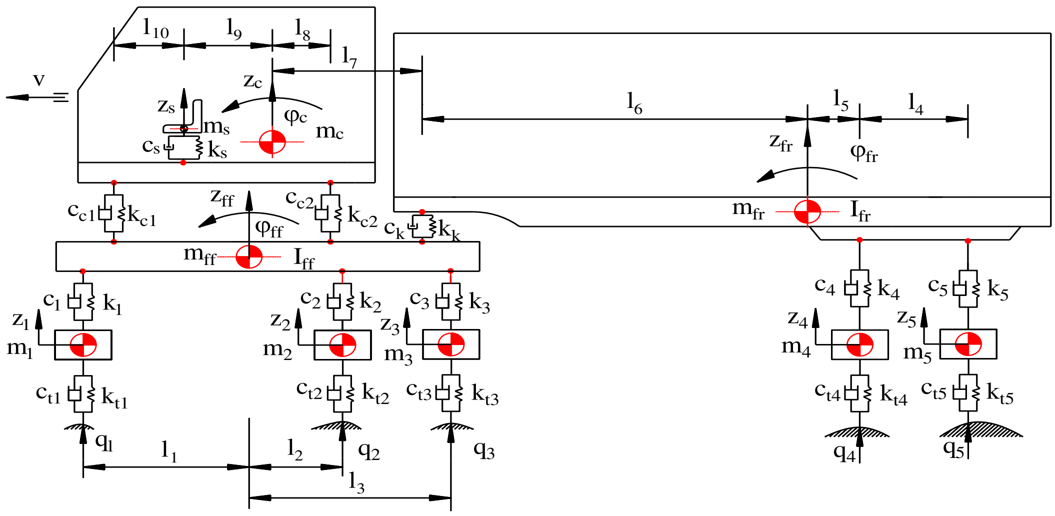


Figure-1. Articulated truck semi-trailer dynamic model.

In Figure-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 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 ($i=1\div5$; $j=1\div10$).

Equation of motion: The equations of vehicle motion are obtained using many different methods such as the Lagrange equation type II [12], the combined method of the multi-body system theory and D'Alembert's principle [13]; Jourdain's principle [14]. However, Newton's second law are used to describe the dynamics differential equation of vehicle system in this study. The general dynamic differential equation for the 5-axle heavy truck is given by the following matrix form:

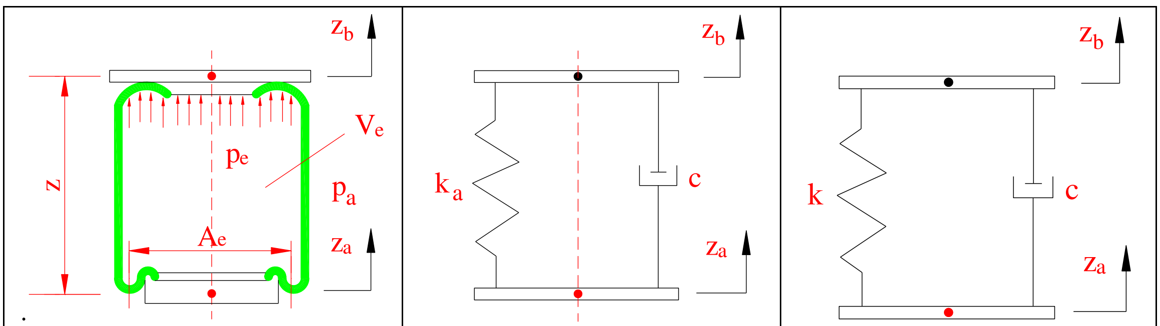
$$[m]\{\ddot{z}\}+[c]\{\dot{z}\}+[k]\{\dot{z}\}=\{F_t\} \tag{1}$$

where [m], [c], and [k] are the (12×12) mass, damping, and stiffness matrices of vehicle, respectively; {z} is the (12×1) vector of displacement of vehicle; $\{F_t\}$ is vector of wheel-road contact forces acting on the vehicle.

Dynamic model of suspension system

Case 1: Using air suspension systems for the rear axles of tractor and trailer

Air suspension is widely used in heavy vehicle applications such as buses, heavy trucks, semi-trailer truck. Many different kinds of vertical air spring dynamic model were proposed by Nishimura, VAMPIRE, SIMPAC, and GENSYS [3]. In this study, the dynamic model of the classic air spring of suspension system is selected and shown in Figure-2(a), (b). In Figure-2(a), (b), p_e is the absolute pressure in the air chamber (Pa), p_a is the atmospheric pressure (Pa), V_e and A_e are the effective volume and area; z_a and z_b are the displacements of axle and vehicle body, k_a is stiffness coefficients of air spring, c is damping coefficients of suspension system



(a) Schematic of an air spring. (b) Dynamic model of air suspension. (c) Dynamic model of leaf spring suspension.

Figure-2. Dynamic model of air suspension system.

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The schematic of an air spring is shown in Figure-2(a). The pressure in the air bag is p_e and pressure atmosphere is p_a. The air spring elastic force F_a in the axial direction provided by the compressed air in the bellow is expressed as:

$$F_a = \left(p_e - p_a\right) A_e \tag{2}$$

From Figure-2(a), there is no air exchange between the air spring component and the other system [15], according to the thermodynamics law and ideal gas state equation given by:

$$p_e V_e^n = const \tag{3}$$

where, n is the polytropic exponent and depends on the following working conditions: n = 1 for the isothermal condition, n = 1.4 for the adiabatic condition, and 1 < n < 1.4 for the polytropic condition.

Differentiating Equation (3) with respect to z yields:

$$\begin{aligned} & V_e^{\,n} \frac{dp_e}{dz} + n p_e V_e^{\,n-1} \frac{dV_e}{dz} = 0 \\ & \frac{dp_e}{dz} = -n \frac{p_e}{V_e} \frac{dV_e}{dz} = n \frac{p_e A_e}{V_e} \end{aligned} \tag{4}$$

Differentiating Equation (2) and then substituting Equation (4) into it yields the air spring nonlinear elastic stiffness:

$$k_a = \frac{dF_a}{dz} = \frac{dp_e}{dz} A_e + \left(p_e - p_a\right) \frac{dA_e}{dz} = n \frac{p_e A_e^2}{V_e} + \left(p_e - p_a\right) \frac{dA_e}{dz} \tag{5}$$

The air-spring dynamic model of suspension system is shown in Figure-2 (b); the vertical force of the air-spring suspension is defined as

$$F = k_a\left(z_b - z_a\right) + c\left(\dot{z}_b - \dot{z}_a\right) \tag{6}$$

Case 2: leaf -spring suspension systems for the front steer axle and the rear axles of tractor and trailer

The leaf-spring dynamic model of suspension system is shown in Figure-2(c). In Figure-2(c), k is stiffness coefficients of leaf spring. The vertical dynamic force of the leaf-spring suspension system is defined as

$$F = k(z_b - z_a) - c(\dot{z}_b - \dot{z}_a) \tag{7}$$

2.2 Road surface excitation

Road surface excitation function is described by various mathematical functions such as the harmonic function, impulse function and random function. The articulated truck semi-trailer dynamic model is simulated under a random road surface. The road surface roughness irregularities can be represented with a normal stationary ergodic random process described by its Power Spectral Density (PSD). According to the International Standards Organization (ISO) 8608 [16], PSD of road roughness can be defined as:

$$\begin{cases} G_q(n) = G_q(n_0)\left(\dfrac{n}{n_0}\right)^{-\mathrm{w}} \\ G_q(\Omega) = G_q(\Omega_0)\left(\dfrac{\Omega}{\Omega_0}\right)^{-\mathrm{w}} \end{cases} \tag{8}$$

where, n is a function of spatial frequency in m^{-1} ($n=\Omega/2\pi$ cycles. m-1), Ω is angular spatial frequency in rad.m^{-1}, n_0 is a function of reference spatial frequency with a value of 0.1 m^{-1}, Ω_0 is reference angular spatial frequency with a value of 0.1 rad.m^{-1}, $G_q(n_0)$ and $G_q(\Omega_0)$ are PSD value for reference spatial and angular spatial frequencies in m^3, w is termed waviness, and reflects approximate frequency structure of the road roughness, 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 $G_q(\Omega_0)$ and $G_q(n_0)$ (see Table-1).

Table-1. Road classification proposal according to the ISO 8608[16].

Road class	$G_q(n_0)$ (10^{-6} m^3)		$G_q(\Omega_0)$ (10^{-6} m^3)	
	Lower limit	**Upper limit**	**Lower limit**	**Upper limit**
A	-	32	-	2
B	32	128	2	8
C	128	512	8	32
D	512	2048	32	128
E	2048	8192	128	512
F	8192	32768	512	2048
G	32768	131072	2048	8192
H	13107	-	8192	-

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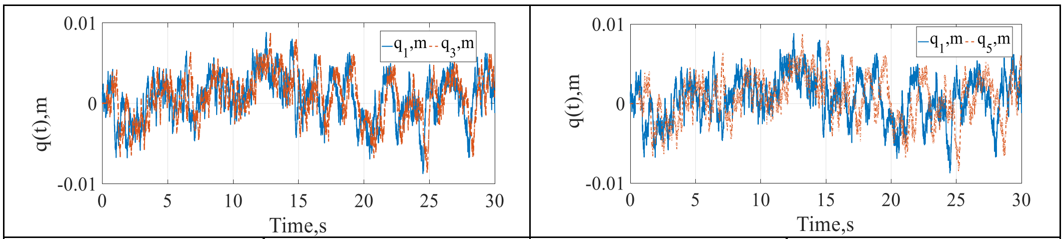
The time domain excitation of the uneven road surface is generated as the sum of a series of harmonics:

$$q(t) = \sum_{k=1}^{N} \sqrt{\frac{2vn_0^2 G_q(n_0)}{f_{mid,k}^2}\Delta f}\,.\cos\left(2\pi f_{mid,k} t + \varphi_k\right) \tag{9}$$

where ϕ_k is the random phase angle following an uniform probabilistic distribution within the 0-2π range; $G_q(n_0)$ is the values in Table-1, v=20 m/s, t is sampling time as t=30s, f = v.n is the road excitation frequency as f_1=0.1 Hz and f_2=50Hz, $f_{mid,k}$ is determined the following formula:

$$f_{mid,k} = f_1 + \frac{2i-1}{2}\Delta f \tag{10}$$

In order to solve equation 9, a program is written in Matlab with different road surface data according to the standard ISO 8068 [8]. The results of the typical random road excitation are shown in Figure-3.



At 1st and 3rd wheel axles.	At 1st and 5th wheel axles.

Figure-3. Random road excitation according to the standard ISO 8068 class B.

3. DYNAMIC LOAD COEFFICIENT

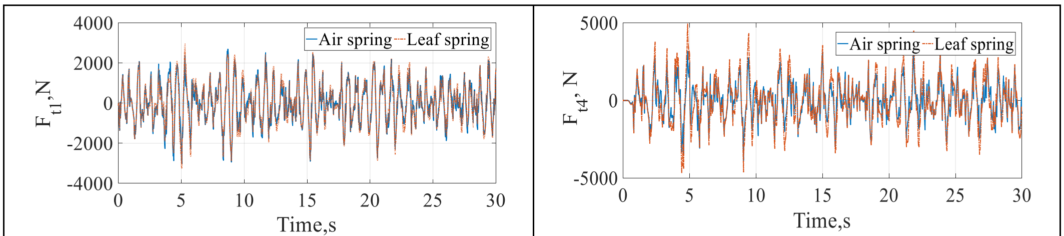
Through research we found that the tire dynamic load is strongly dependent on vehicle suspension design and operating conditions. Dynamic tire load would lead to stress and strain of road surface. The long-term accumulation of road surface plastic deformation causes the destruction of roads, such as cracks and rutting. There are several indicators that are proposed by the researcher such as IM and DLC indicators [1], DLSC (dynamic load-sharing coefficient) and DLC indicators [4]. To analyze the performance of the air spring of the suspension systems compared with the leaf spring of the suspension systems of a vehicle, the DLC is selected as an indicator in this study. The dynamic load coefficient DLC [1-4], [9-10, 17] is defined as the ratio of a ratio of the root mean square (r.m.s) of the vertical dynamic tire force over static load.

$$DLC = \frac{F_{t,rms}}{F_s} \tag{11}$$

where, $F_{t,rms}$ and F_s are the r.m.s of the vertical dynamic and the static tire force, respectively.

Dynamic simulation and evaluation

In order to analyze the vertical dynamic tire loads of a semi-trailer truck acting on road surface equipped with air and leaf spring of the suspension systems, the mathematical model of the articulated vehicle is presented in section 2 and MATLAB/Simulink software is used to solve the articulated vehicle differential equations with a set of parameters of the articulated vehicle by the references [11] as a set of parameters of suspension system by the references [11] and [18]. The simulation results of the tire vertical forces acting on road surface at 2^{nd} and 4^{th} axles of vehicle and comparing of the DLC values at all axles with air and leaf springs of suspension systems are shown in Figure-4 and Table-1 when vehicle moves on the ISO class B road surface roughness at v=20 m/s and full load.



at 1st wheel axle	at 4th wheel axle

Figure-4. The tire vertical forces acting on road surface at 2^{nd} and 4^{th} axles of vehicle.

Table-2. Comparing of the DLC values with air- and leaf-spring suspension systems.

Axles	Air-spring suspension (DLC)	Leaf-spring suspension (DLC)	Decrease %
1^{st}	0.0188	0.0195	3.72
2^{nd}	0.0093	0.0145	55.91
3^{rd}	0.0095	0.0143	50.53
4^{th}	0.0143	0.0188	31.47
5^{th}	0.0110	0.0138	25.45

The results are shown in Figure-4, we could be determined the maximum amplitudes of the vertical dynamic tire forces at 1st and 4^{th} axles and its values are 2693.70N, 3415.10 N with the air spring of the suspension systems and 2954.60 N, 4951.20N with the leaf spring of the suspension systems, respectively. From obtained this result, it is shown that the dynamic tire load amplitudes with the air spring of suspension systems are significantly reduced. From the results in Table-2, it is shown that the values of DLC at 1^{st}, 2^{nd}, 3^{rd}, 4^{th} and 5^{th} axles with the air spring of the suspension systems are significantly reduced by 3.72%, 55.91%, 50.53%, 31.47% and 25.45% in comparison with the leaf spring of the suspension systems, respectively. The above cases present that the air spring of the suspension systems reduced the dynamic tire loads acting on road surface better than the leaf spring of the suspension systems. The performance reduction of dynamic tire loads with the air spring of the suspension systems acting on road surface under different operating conditions is being presented in the following in sections.

3.1 Performance of air suspension systems under different operating conditions

3.1.1 Road surface conditions

Road surface roughnesses not only affect movement safety, but also affect vehicle ride comfort as well as road safety. To analyze the effect of road roughness on the DLC value, five road surface conditions from class A (very good) to class E (very poor) in ISO 8608 are used to analyze the performance of the air spring of the suspension systems when vehicle operates under fully load and the velocity of 20 m/s. The DLC values at 4^{th} axle of vehicle with air and leaf springs of the suspension systems are shown in Figure-5(a). Figure-5(a) shows that the DLC values at 4^{th} axles of vehicle with the air spring of the suspension system are reduced by 28.59%, 31.47%, 28.72%, 28.59% and 28.53%, respectively in comparison with the leaf spring of the suspension systems when vehicle operates under the road surface condition changing from the ISO class A to class E road surface with a velocity of 20 m/s and fully load.

3.1.2 Vehicle speed conditions

To analyze the effect of vehicle speed conditions on the DLC value, five vehicle speeds: 5m/s, 10m/s, 15m/s, 20m/s, 25m/s and 30m/s were considered to compare the performance of the air and leaf springs of the suspension systems when vehicle operates under the ISO class B road surface roughness and full load. The DLC values at 4^{th} axle of vehicle with two suspension systems under the different vehicle speed conditions are shown in Figure-5(b). Figure-5(b) shows that the DLC values at 4^{th} axles of vehicle with the air spring of the suspension system are reduced respectively at the all vehicle speed conditions in comparison with the leaf spring of the suspension systems.

3.1.3 Vehicle load conditions

To analyze the effect of vehicle speed conditions on the DLC value, the load values of heavy trucks: empty load, half load, fully load and over 50% load are used to compare the performance of the air and leaf springs of the suspension systems when vehicle operates under the ISO class B road surface roughness with v = 20 m/s. The DLC values at 4^{th} axle with air and leaf springs of the suspension systems under the different vehicle load conditions are shown in Figure-5(c). Figure-5(c) shows that the DLC values at 4^{th} axles of vehicle with the air spring of the suspension system are reduced by 16.86%, 20.88%, 31.47%, and 25.56%, respectively in comparison with the leaf springs of the suspension systems under all vehicle load conditions.

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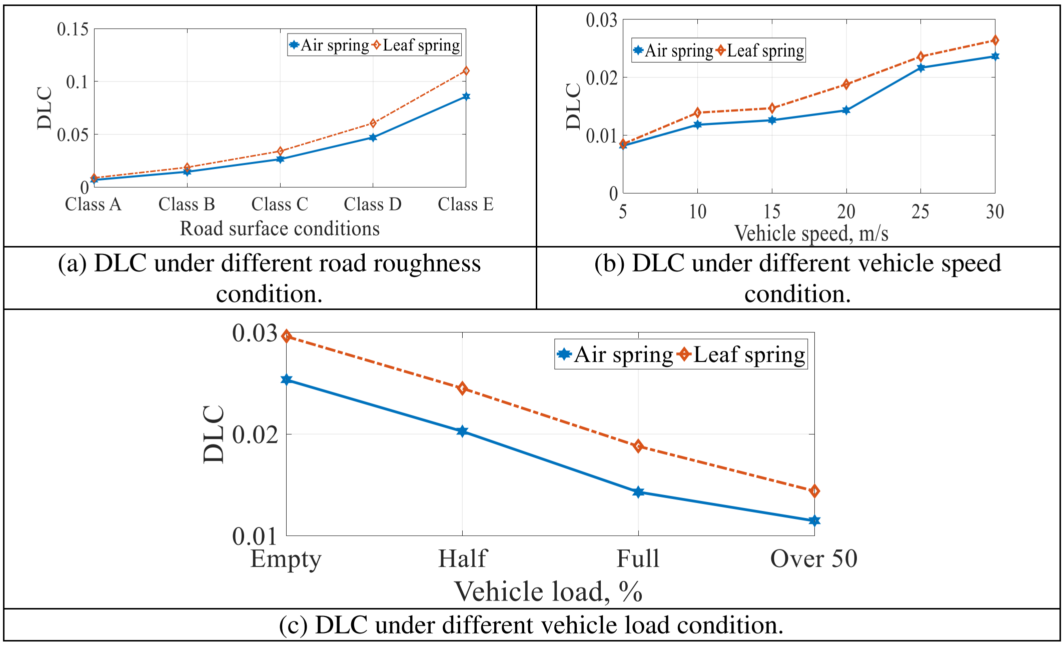

(a) DLC under different road roughness condition.

(b) DLC under different vehicle speed condition.

(c) DLC under different vehicle load condition.

Figure-5. Comparing of the DLC values with the air and leaf springs of suspension systems when vehicle operates under the different conditions.

4. CONCLUSIONS

In this study, a half-vehicle dynamic model of a semi-trailer truck is developed for analyzing the vertical dynamic tire loads of a semi-trailer truck acting on road surface equipped with air and leaf springs of the suspension systems when vehicle operates under the different conditions. Several conclusions that can be drawn from the results as follows:

a) The performance of the road-friendly vehicle suspensions with air springs are significantly improved by 3.72%, 55.91%, 50.53%, 31.47% and 25.45% in comparison with the leaf spring of suspension systems, respectively when vehicle operates under the ISO class B road surface roughness at v=20 m/s and full load.

b) The performance of the road-friendly vehicle suspensions with air springs are improved by 28.59%, 31.47%, 28.72%, 28.59% and 28.53%, respectively in comparison with the leaf spring of the suspension systems when vehicle operates under the road surface condition changing from the ISO class A to class E road surface with a velocity of 20 m/s and fully load. Similar to other operating conditions, the performance of the road-friendly vehicle suspensions with air springs are improved respectively in comparison with the leaf spring of the suspension systems.

c) To improve vehicle ride comfort as well as road surface friendliness, an optimal design of geometrical parameters of air spring should be further considered in future studies.

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