SHOCK ABSORBER AND SPRING CONTRIBUTION REDUCES VERTICAL VEHICLE LOADS THAT BURDEN THE ROAD STRUCTURE

By Simon Kaka

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SHOCK ABSORBER AND SPRING CONTRIBUTION REDUCES VERTICAL VEHICLE LOADS THAT BURDEN THE ROAD STRUCTURE

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ABSTRACT

Driving comfort for riders and passengers is a key target to be achieved. Fluctuations in vehicle loads, bumps, perforated surfaces, and other road damage will greatly affect the vehicle suspension working system. This study aims to (1)examine more about the effect of vertical dynamic load of vehicles and changes in dimensional barriers on the road surface in its path.(2) obtain the amount of vibration and load reduced by the working of the spring and shock absorber. Load changes arising from the number of non-permanent passengers always burden the vehicle suspension in a fluctuating manner. Experimentally these load fluctuations are replaced by pneumatic actuator forces of varying magnitude based on the regulatory pressure of the regulator. The deviations generated by the varying load work are measured by placing a proximity sensor along the spring movement. The vertical dynamic load transformation up to the road surface is measured using a "Load cell" mounted under the wheels of the vehicle. Characteristics of vertical dynamic vibration occurring due to several dimensional barriers, U (cm) obtained using mathematical modeling method with 2 DOF suspension system transfer function. The results showed a condition on the body and wheels of vehicles experienced a brief overshoot for 0.14 seconds with a deviation of 0.178 m. From the graph shows that the rate of deviation that occurs is large enough that Y2d = 1.03 m / s caused by a sudden shock that occurred on the wheels of the vehicle. This condition does not last long that is only duration t = 0.22 s, because of the reaction of the spring and a shock absorber that can dampen the vibration not less than 25% to the vibration caused by the vertical load of the body and the axle of the vehicle.

Keywords: vibration, spring, shock absorber, deviation, vertical load.

1. INTRODUCTION

Increasing the volume of vehicles, especially the four-wheeled vehicles that cross the road will further increase the load, while the road capacity to support the recurrent load of vehicles decreases with increasing age of use. Vibrations originating from vertical dynamic loads of vehicles often fluctuate due to unstable drivers and passengers. Such conditions will further weaken the ability of the road structure to accept the fluctuating load.

The spring and shock absorber mounted on each of the wheels of the vehicle is expected to be able to overcome and reduce the vertical dynamic load of vehicles overloading the road structure. The vehicle suspension system according to [1-3] is composed of a spring and a shock absorber arranged in parallel. The main function of the suspension system is to support the weight of the vehicle, to provide comfort for the rider, to keep traction of the wheel on the road surface condition, and to maintain the alignment of the front wheel and rear wheel [3-5]. If the suspense spring is very rigid, then the shock absorber will not efficiently absorb the shock that comes from the form of road surface resistance [3]. The shape and mechanism of the suspension system for quarter vehicles are shown in Figure-1 (a), with sprung mass (m_2), un-sprung mass (m_1), suspension spring (k_2), shock absorbers (c) and tire elastic constants on wheels (k_1), [6, 7].

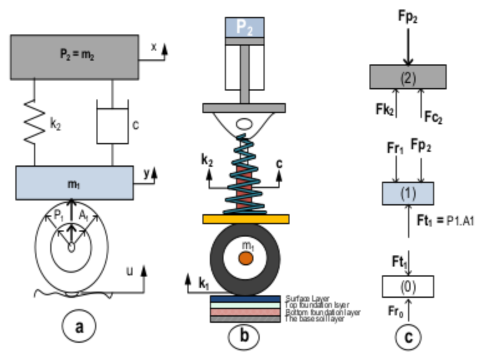

Figure-1. Pneumatic system experimental loading mechanism.

The magnitude of the pneumatic cylinder thrust force generated by the air pressure P_2 (bar) according to [6, 8] is formulated as;

$$F_k = \frac{\pi}{4} D^2 P_2 (\mathrm{N}) \qquad (1)$$

If the dimensions of the cylinder, D = 100 mm (0.100 m) are substituted to equation (1) then the compression force of the suspension mechanism is obtained:

$$F_k = 785\ x\ P_2\ and\ F_{p2} = 706.5\ P_2 = 707\ P_2\ (\mathrm{N}). \qquad (2)$$

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Based on the free-body diagram of Figure-1(c), the equation of the wheel loading the road structure is;

$$\left.\begin{aligned} F_{t1} &= F_{p2} + F_{r1} \\ F_{t1} &= (k_2 + c)x_2 + (m_1.g) \end{aligned}\right\} \text{(N)} \tag{3}$$

If in equation (3), the spring constant k_2 (N / mm), the coefficient of damping c (Ns / mm), the weight of the wheel m_1 (kg) and the gravitational acceleration g (m / s^2) are obtained;

$$F_{t1} = F_{ro} = 707\, P_2 + (m_1.g)\ \text{(N)} \tag{4}$$

The mass sprung of m_2 supported by the spring and the shock absorber with the spring stiffness of k_2 and the damping coefficient c will overload the vehicle wheel as the mass unsprung of m_1 further gives the action force to the contour of the road surface.

The reaction force of the road contour to the tire will be distributed in the direction of u through the tire elasticity constant of k1. Thus the transfer function of a quarter vehicle suspension system for a front wheel with two degrees of freedom (2DOF) is obtained by comparing the output and input as follows:

$$\frac{Y(s)}{U(s)} = \frac{k_1(cs + k_2)}{m_1 m_2 s^4 + (m_1 + m_2)cs^3 + [k_1 m_2 + (m_1 + m_2)k_2]s^2 + k_1 cs + k_1 k_2} \tag{5}$$

Calculation of spring constant value, k_2 by [9, 10] respectively in the book "Machinery's Handbook, 29th Edition and the journal Mechanism and Machine Theory are formulated as follows:

$$k_2 = \frac{Gd^4}{8n_a D^3} \tag{6}$$

G is the modulus of stiffness (N / cm^2), d is the diameter of the wire (cm), n_a is the number of active coils and D is the mean coil diameter (cm) obtained from the difference between the outer diameter of the coil and the diameter of the wire.

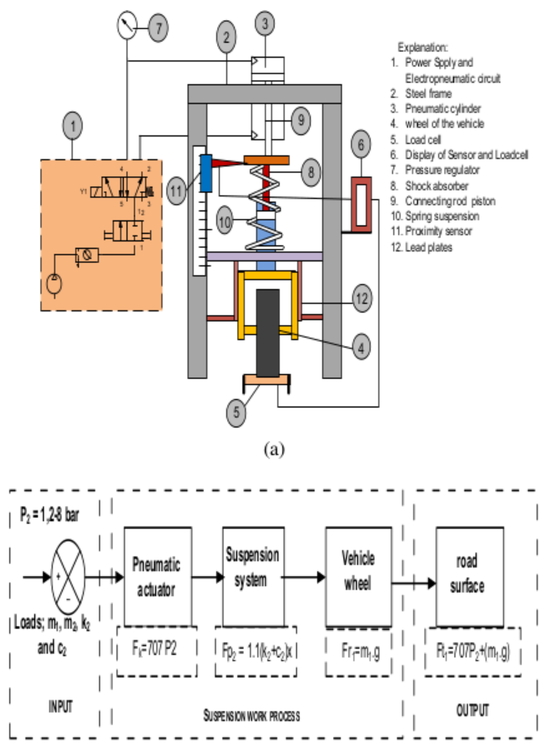
The number of active coils, n_a is usually less than the total coil, n whereas the spring compress area is between the minimum compression of 20% and the maximum compression of 80%. If the number of active coils, n_a = 80% x n, and the mean coil diameter, D = Do - d then the value of the spring constant, k_2 can be obtained by using equation (6). The energy analysis and efficiency of the shock absorber can be produced by the suspension work on the wheels of the vehicle use a linear dynamic equation, [11, 12].

2. RESEARCH METHOD

Experimental tests conducted on the mechanical work of suspension on the wheels of the vehicle is by adjusting the working pressure P2 (bar) on the regulator ranging from 1 to 8 bar [6]. Furthermore, the loading of the spring and the shock absorber by the pneumatic actuator begins to occur when the 5/2 directional control valve is operated.

The change in the position of compression motion on the spring and the shock absorber is measured by placing the proximity sensor along the spring movement.

The amount of deviation detected by the sensor can be transferred to the LCD in digital form. It should be noted that the change in the size of the readings on the LCD screen is correlated with the regulatory pressure settings. By placing a Load cell gauge just below the wheel of the vehicle, the measurement of the vertical dynamic load of the vehicle successfully transformed to the road surface can be obtained through a direct reading on the LCD screen.



(a)

(b)

Figure-2. (a) Loading experiments with pneumatic actuators (b) Block diagram of vehicle suspension loading process.

Optimization of physical models, using pneumatic cylinders as dynamic vertical drive test simulators on wheel drive suspension work has been discussed by [6].

The type of pneumatic cylinder selected is, DNU-100-300-PVA and sourced from [13] with specifications; Diameter of piston D = 100 mm, stroke L = 300 mm, Weight piston W = 3.864 kg = 38.64 N, and Weight piston rod / 10 mm w = 0.090 kg . The forward and reverse thrust force of cylinder piston at a working pressure of 6 bar are respectively F_k = 4496 N and F_m = 4221 N.

Mathematical model optimization is done by making the weight data of wheel axis, m_1 (kg) and body weight, m_2 (kg) as INPUT as shown in Figure-2 (b). Similarly, the constant values of k_1, k_2, and c_2 are the input data to create the program in Mat Lab.

The test spring used in the suspension system mechanism is a kind of "Helical" with the following specifications: inside and outside coil diameters are Di = 12.985 cm and Do = 15.815 cm, diameter of spring wire, d = 1.415 cm, and a number of coils, n = 5 pieces. The spring material used is from Chrome Vanadium, ASTM A231 with Stiffness Modulus, G = 7.929E + 10 Pa = 7.929 x 10^6 N / cm^2.

The description of the implementation of planning activities, data collection, experiments, mathematical studies, and the results to be obtained is shown in the research flow diagram as follows:

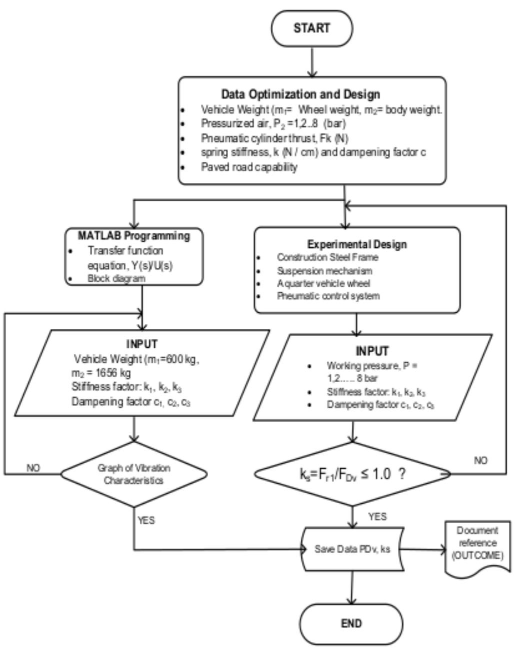


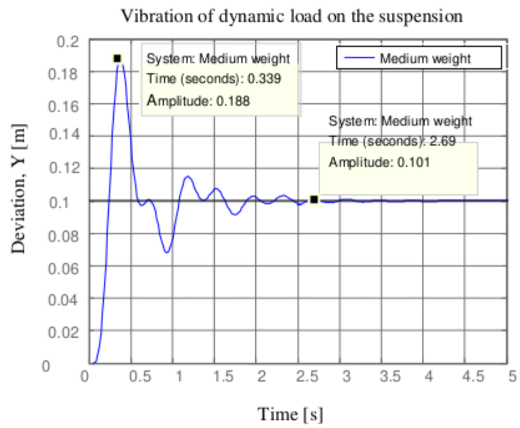
Figure-3. The research flow diagram.

3. RESULTS AND DISCUSSIONS

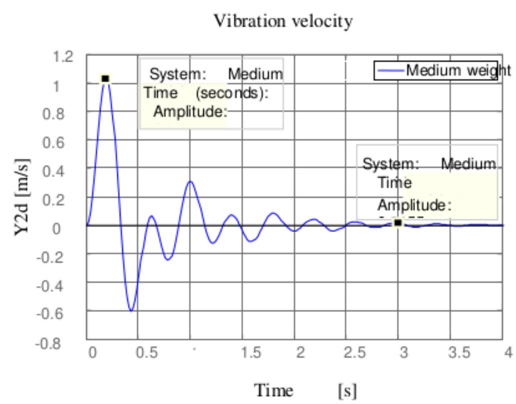
The actual vehicle vertical load calculation which overloads the road structure is by multiplying the dynamic factor of 80% with the force according to equation (4). If the weight of the vertical load, after getting suspense resistance on the wheels of the vehicle is Fr1 = 22197 (N) then the load of the vehicle that weighs the road is Ft1 = 0.8 x 22197 (N) = 17758 (N). The magnitude of the reduced vertical load by the spring and the shock absorber is 4439 (N) or equivalent to 25%.

Programming with Mat Lab software referring to equation (5) will produce the suspension vibration characteristics with small and large deviations as shown in Figure-3 (a), (b) and (c).

The execution result of the program has obtained the characteristic graph of vibration on the suspension as the impact of vertical dynamic load. In Figure-4 (a) if the INPUT resistance is U = 0.1 m, then the suspension on the wheels will experience vibration with maximum deviation, Y = 0.188 m with a vibration time of 0.339 s. then two seconds later, the minimum deviation occurs at Y = 0.101 m. Proportionally author [14] has obtained 0.12 m deviation on the measured vehicle wheel as a strain using the WIM sensor that placed below the tire.



(a)



(b)

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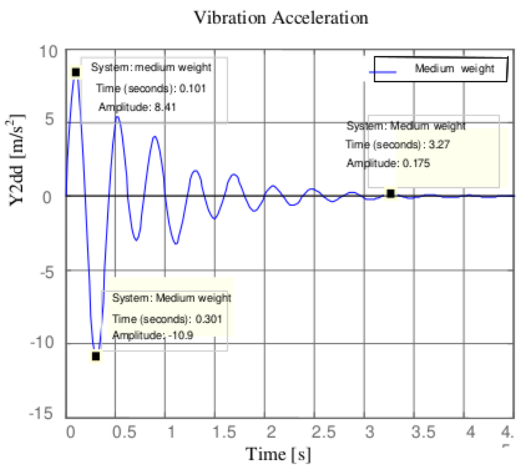


(c)

Figure-4. (a) Characteristics of vibration on the body and wheels with deviation Y (m), (b)The characteristics of vibration velocity on the body and wheels, V (m/s), (c) The characteristics of vibration acceleration on the body and wheels, a (m/s^2).

The characteristic shape of the vibration in the wheel suspension as in Figure-4(a) is caused by a mound as high as U = 0.1 (m) on the road surface. Similarly, the reaction to the total dynamic load of the vehicle can result in a vertical overshot deviation of Y = 0.188 (m) with a duration of 0.1 seconds. Furthermore, within a time interval of 2.34 seconds later, the vibrations begin to shrink and stabilize at 0.66 seconds.

The result of the research in Figure-4 (b) shows that the largest deviation rate occurs at Y2d = 1.03 m / s when t = 0.22 s, while the lowest deviation rate is Y2d = 0.02 m/s when the vibration time t = 2.65 s. From the graph shows that the rate of deviation that occurs largely enough caused by a sudden shock that occurs on the wheel. Such conditions can not last long because of the resistance of the spring and shock absorber.

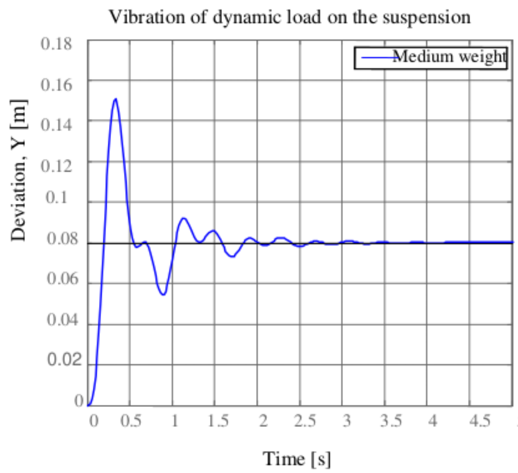
The change in the drift velocity shown in Figure-4(c) has a significant slowdown of Y2dd = (- 10.9 + 8.41) m / s = - 2.59 m / s^2 at t = 0.301 s.

It should be noted that the perceived comforts of the vehicle are often disrupted by the uneven, hollow, bumpy surface of the road surface and are filled with mound-shaped materials. Comfort can also be felt by passengers and drivers when the mechanism of shock absorber damper is able to reduce the vertical dynamic load of the vehicle well.

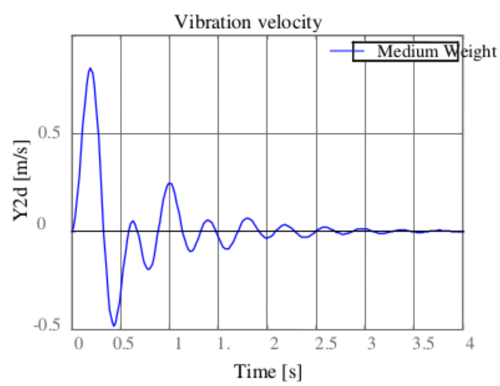
Given the circulation of liquid or gaseous fluid determined by the regulation of the flow control valve in the cylinder chamber, the performance of the absorbent shock is getting better. The amount of deviation experienced by the body and the vehicle's axle while experiencing vertical dynamic loads of experimental results and modeling using Mat Lab is shown in Figure-4 above.

If the road surface condition has a dimension of disturbance of U = 0.08 m, it will occur the deviation in the suspense system is slightly smaller than the deviation generated by the previous disturbance dimension of U = 0.1 m. It should be noted that the large and small deviations that occur in the spring work system and shock absorber will be directly correlated with the frequency of vibration felt by the driver and the passengers.

A description of the deviation and vibration conditions occurring during the disturbance of U = 0.08 m as shown in Figure-5. The deviation of Y = 0.15 m with the vibration time t = 0.35 (s) decreases the deviation to Y = 0.082 m during, t = 3.5 (s). The overshot condition occurs only within a short time of t = 0.15 (s) while the vibrations begin to disappear at t = 3 (s). The vibration velocity of Y2d = 0.75 m / s occurs when the vibration time t = 0.3 (s), while the vibration acceleration of Y2dd = 6.5 m / s^2 occurs with the duration of time, t = 0.15 (s).



(a)



(b)

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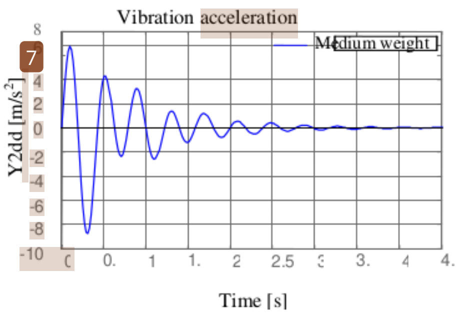

(c)

Figure-5. Characteristics of vibration on the body and wheels with (a) deviation Y (m), (b)vibration velocity on the body and wheels, V (m/s), (c) vibration acceleration on the body and wheels, a (m / s^2).

The comparison of deviation values between the experimental results and the programming with Mat-Lab is shown in Table-1.

Table-1, Characteristics of experimental results and theoretical deviation.

Pressure	Experimental		Theoretical	
P1(bar)	t(s)	Y(m)	t (s)	Y(m)
1	0.5	0.059	0.5	0.11
2	1.0	0.067	1	0.087
3	1.6	0.078	1.5	0.11
4	2.1	0.086	2	0.1
5	2.7	0.088	2.5	0.098
6	3.3	0.092	3	0.1
7	3.9	0.096	3.5	0.1
8	4.6	0.098	4	0.099

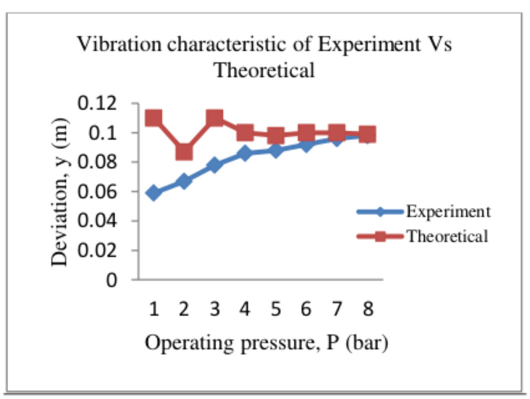

Figure-6. Relationship of vibration characteristics between experimental and theoretical studies.

The experimental vibration characteristics and theoretical studies show a trend that starting at the working pressure of 4 bars vibration deviation is increasingly blessing each other and finally at the working pressure of 8 bars the deviation is at Y = 0.1 m.

The mean deviation of the experimental results is Y = 0.083 m, with a mean duration of 2.5 s, while the average deviation of the programming results is Y = 0.1 (m) in the mean duration, t = 4.5 s.

The rate of deviation and the acceleration of movement of the vehicle body that weighs 300 kg by [4] are 0.03 m and 1.8 m/s^2 respectively and stabilized at t = 3.5 (s). The energy from suspense work is all transmitted to the un-sprung mass of the vehicle, [11]. The shape of the deviation graph shows that at the time of vibration, t = 0.5 s there is a deviation gap of Y = 0.051 m, when pressure P = 1 bar, but after reaching the vibration time t = 2.1 s when pressure P = 4 bar, then the deviation gap starts to decrease to Y = 0.014 m.

The number of vibrations generated by the ever-changing deviations will take place for a long time. To overcome this problem, the suspension system should follow the conditions between flexible and rigid [3].

4. CONCLUSIONS

Based on the results of experimental studies and analyzes that have been done on the vehicle suspension work performance, it can be concluded as follows:

a) Characteristics of the working mechanism of the suspension on the body and wheels of the vehicle for the type of medium weight vehicle if experiencing the disturbance U = 0.1 m it will happen the largest deviation Y = 0.18 m with a duration of 0.1 seconds. Within a span of 2.34 seconds later, the vibrations begin to shrink and eventually stabilize at 0.66 seconds. whereas for U = 0.08 m the maximum deviation is Y = 0.15 min the duration of time of 0.15 (s).

b) Characteristics of suspension vibration rate on the largest vehicle wheels for disturbance U = 0.1 m occur at Y2d = 1.03 m / s when t = 0.22 s, while the lowest deviation rate is Y2d = 0.02 m / s when the vibration time t = 2.65 s. The vibration velocity of Y2d = 0.75 m / s for the resistance U = 0.08 m occurs during, t = 0.3 (s), while the vibration acceleration of Y2dd = 6.5 m / s2 occurs when, t = 0.15 (s).

c) The occurrence of speed changes and acceleration of deviation on the work of the suspension will reduce the driving comfort felt directly by passengers and drivers.

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NOMENCLATURE

Symbol	Unit	Meaning
A, Ao	cm^2	The area of the tire contact on the road
A_1	m^2	The inside area of the tire
c_2, c	Ns/mm	Coefficient of shock absorber damping
C	-	Coefficient of vehicle distribution
COG	cm, cm	Center of gravity
D		Piston diameter
Di	mm	The inside diameter of the coil
DLC	cm, mm	Dynamic Load Coefficient
Do	-	The outer diameter of the coil
DOF	cm, mm	Degree Of Freedom
d	-	Diameter of spring wire
di		Diameter of the air channel
E	-	The vehicle's axis equivalent number
Ee	Pa	Elastic Modulus of the road
F_{Ao}	N	Road stability force
FBD	-	Free Body Diagram
Fc	N	Damping force of shock absorber
F_{Dv}	N	Vertical dynamic vehicle load force
F_{ef}	N	Effective force of piston forward
Fk	N	Theoretical force of piston cylinder
Fp_2	N	Effective force of piston forward
Fr_1	N	Force of the wheels
Ft_1	N	Force of the road surface
Fg	N	Friction force of tires
Fh	N	Horizontal force of tire
fg	-	Coefficient of the road surface
f	1/s	The frequency of vibration
g	m/s^2	Acceleration of gravity
G	Pa	Spring stiffness modulus
k	-	The standard axis correction factor
k_1, k_e	*N/m*	Tire elastic constants
k_2, k_s	*N/m*	The suspension spring constant
Ls	Ton	Axis loads of vehicles
L	cm	The length of the road surface contact area
m_1	Kg	Unsprung mass of vehicle
m_2	Kg	Sprung mass of vehicle
n	pieces	The number of spring coils
n_a	pieces	Number of active coils
P_1,p_1	bar	Air pressure in the tire
P_2	bar	Working air pressure
t	s	Vibration time
u	mm	Deviation shock absorber
W	N	Total vehicle weight
y	mm	deviation on the suspension spring.

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