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Effects of Work on Shock Absorber and Spiral Springs Against Vertical Loads of Vehicles Burdening the Road Structure

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Abstract. This research aims to examine the effect of vertical dynamic load of vehicles and changes in dimensional barriers on the road surface in its path. Experimentally this fluctuating load is replaced by a pneumatic force change based on the regulation of air pressure on the regulator. The deviations generated by the varying load work are measured by placing a proximity sensor along the spring movement. The amount of vertical load transformation reaches the road surface is measured by using Load cell. Characteristics of vertical dynamic vibration occurring due to several dimensional barriers, U (cm) obtained using mathematical modelling method with 2 DOF suspension system transfer function. The results showed a condition on the body and wheels of vehicles experienced a brief overshot for 0.14 seconds with 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 the spring reaction force and shock absorber can absorb 45% vibration against the sprung and un-sprung vertical load of the vehicle.

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, 2] 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 [2-4]. 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 [4].

The shape and mechanism of the suspension system for quarter vehicles is 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), [5, 6].
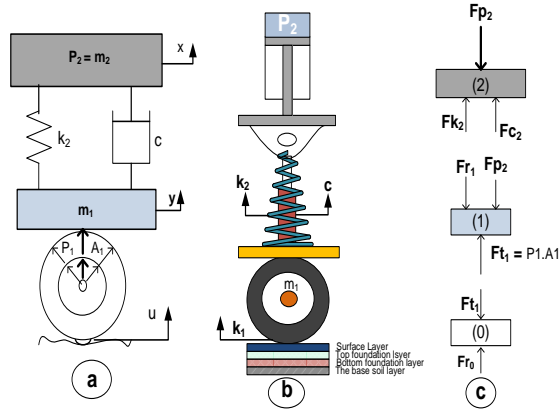

Figure 1. Pneumatic system experimental loading mechanism

The magnitude of the pneumatic cylinder thrust force generated by the air pressure P_2 (bar) according to [5, 7] is formulated as;

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

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:

$$F_k = 785 \times P_2 \ \text{and} \ F_{p2} = 706.5 P_2 = 707 P_2 \ (N) \tag{2}$$

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

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

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:

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

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 k_1 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:

$$\frac{Y(s)}{U(s)} = \frac{k_1 (cs + k_2)}{m_1 m_2 s_4 + (m_1 + m_2) cs^3 + \left[k_1 m_2 + (m_1 + m_2) k_2 \right] s2 + k_1 cs + k_1 k_2} \tag{5}$$

Calculation of spring constant value, k_2 by [8, 9] 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, $n_a = 80\% \times n$ and the mean coil diameter, $D = D_0 - d$ then the value of the spring constant, k$_2$ can be obtained by using equation (6). The energy analysis and efficiency of shock absorber can produced by the suspension work on the wheels of the vehicle uses a linear dynamic equation [10, 11].

2. Research Method and Material

2.1. Pneumatic cylinder and direction control valve
Experimental tests conducted on the mechanical work of suspension on the wheels of the vehicle is by adjusting the working pressure P$_2$ (bar) on the regulator ranging from 1 to 8 bar [5]. 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.

| (a) | (b) |

Figure 2. (a) Pneumatic actuator (b) Memory valve with 5/2-way

The dimensions of the Pneumatic actuator and the 5/2 directional control valve used at the working pressure, P = 6 (bar) are as follows; The type of pneumatic cylinder selected is, DNU-100-300-PVA and sourced from [12] 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, piston weight, W = 3.864 kg = 38.64 N Weight / 10 mm and piston rod, w = 0.090 kg / 10mm.

2.2. Experimental set up
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.

Optimization of physical models, using pneumatic cylinders as dynamic vertical drive test simulators on wheel drive suspension work has been discussed by [5, 13]. 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 3 (b). Similarly, the constant values of k_1, k_2, and c_2 are the input data to create the program in Mat Lab.

<table>
<tr><td>(a)</td><td>(b)</td></tr>
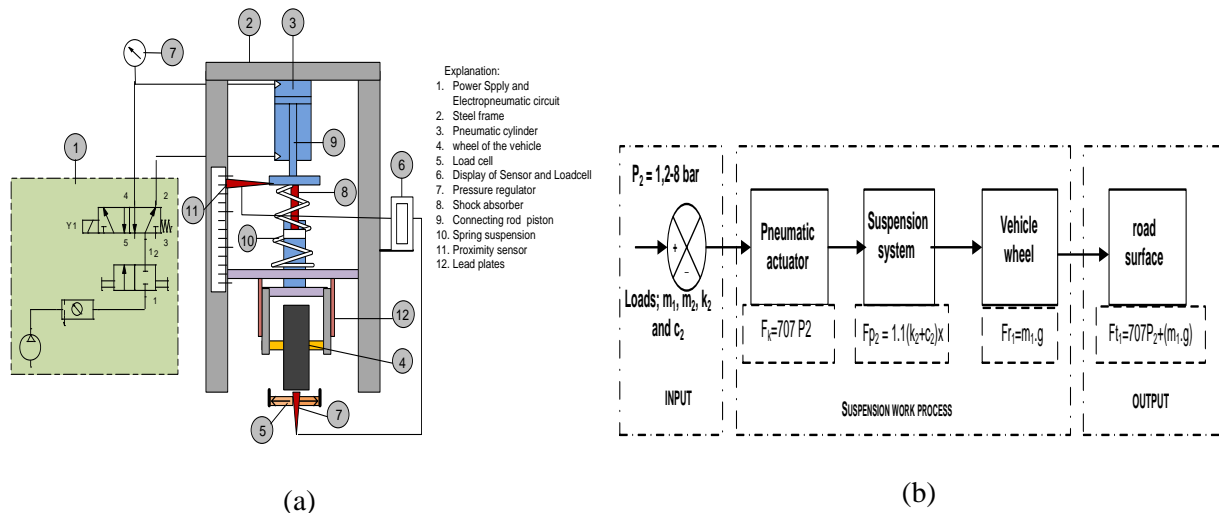</table>

Figure 3. (a) Loading experiments with pneumatic actuators (b) Block diagram of vehicle suspension loading process

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 number of coils, $n = 5$ pieces. The spring material used is from Chrome Vanadium, ASTM A231 with Stiffness Modulus, $G = 7.929E + 10$ Pa $= 7.929 \times 10^6$ N / cm^2.

2.3. Population and sample

The population of the targeted vehicle is the medium category vehicles including Terios, New Rush, Suzuki R3, Grand Max, Suzuki APV, Isuzu D-Max pickup until Fortuner. The sample selection of medium weight vehicles is shown in Table 1, each of which is randomly performed.

A number of vehicles belonging to the medium weight category, Terios is one medium-weighted vehicle with a net weight of $m_2 = 1665$ kg and axis weight, $m_1 = 600$ kg, then selected as research sampling, representing the population of other vehicle types.

Table 1. Vehicle data with medium weight categories

| No. | Vehicles | Vehicle Weight (kg) *) | | | |
| | | Body | Total/Net | Maximum | wheel axis |
		m_2	m_2	m_2	m_1
1	Terios	1235	1665		600
2	New Rush	1950	2510		600
3	PickUp, Grandmax	1950	2510		600
4	Suzuki R3	1185	1745		480
5	Suzuki APV	1950	2510		600
6	Isuzu D-max, Pick up	1990	2550	2550	600

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2.4. Data collection technique

Procedures and steps taken on primary data retrieval activities are: *First*; Collect medium-weighted vehicle data (M_2) which has axle weight, m_1 (kg) and body weight m_2 (kg) as in Table 1, relevant

journals, books, and other references. *Second*; Calculating the spring stiffness factor k_2, damping value, c_2 and wheel specifications in each vehicle type either obtained from reference journal or through calculation. *Third;* Conducting experimental and observation activities directly by operating the motion simulation test equipment as follows; Prepares the pressurized air on the air service unit while adjusting the working pressure on the regulator starting from P2 = 1 bar to 8 bar. *Fourth*; Operate the actuator lever on the 5/2 direction control valve by turning the lever towards the cylinder piston movement, up and down. When the cylinder piston moves down, the loading process will occur in the spring and shock absorber in the compression direction.

The drift, x (mm) that occurs due to loading, the magnitude can be read on the available digital display ranging from 0 to 60 mm by the proximity sensor. It should be noted that the magnitude of the deviation is highly correlated with the pressure setting on the Manometer.

Moving time, t (s) can be measured using a stopwatch when there is a drift from x_0 to x (mm). The amount of vertical load that reaches the road surface can be measured by placing "Load cell" just below the wheel of the vehicle.

3. Result and Discussion

3.1. Load and programming analysis

Programming with Mat Lab software referring to equation (2.3) will produce the suspension vibration characteristics with small and large deviations as shown in Figure 4 (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.

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 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 modelling using Mat Lab is shown in Figure 4.
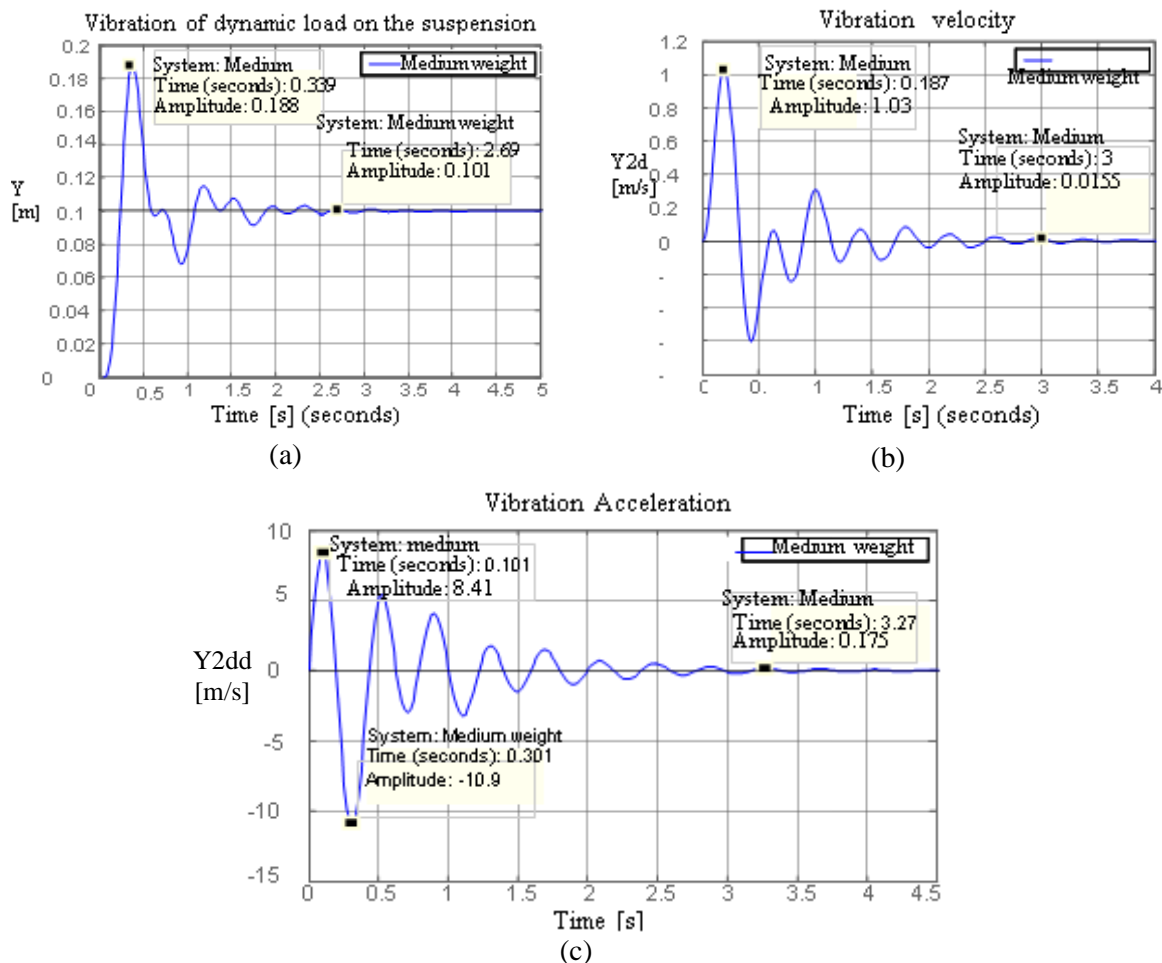
(a)

(b)

(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).

3.2. Pneumatic and road structure force

Work of loads in the vertical direction generated by pneumatic actuators based on equation (1.2) get reactions from the spring force, shock absorber, and air pressure in the tire [2, 5]. The magnitude of the vertical dynamic load of vehicles that overload the structure of the road according to [1, 7] is referring to the equation (1.3). Some variable and constant data such as pressure, P_2 (bar), m_1 axis load (kg), vehicle body load, m_2 (kg), spring constant, k2 (N / m), and shock absorber c2 (Ns / m) will be substituted into equations (1.2) and (1.3).

The relationship between the pneumatic action force and the force that overlies the road structure is shown in Figure 5. From the graph showing the damping characteristics by shock absorber gives a very big influence on the dynamic load of the vehicle which overload the road structure. Such a condition causes the dynamic load of F_{t1} (N) to overload the structure of the road according to Figure 5 to be around 10000 (N) or equivalent to 1000 kg. If the medium weighted vehicle being tested has a wheel axis weight of m_1 = 600 kg, body weight m_2 = 1665 kg and total weight, m_t = 2265 kg then obtained the dynamic load of the reduced vehicle average is m_r = m_t - 1000 kg = 2265-1000 = 1265 kg. The dynamic load drop presentation that weighs the structure of the path by the attenuation of the shock absorber is 55.85 % [5, 15].
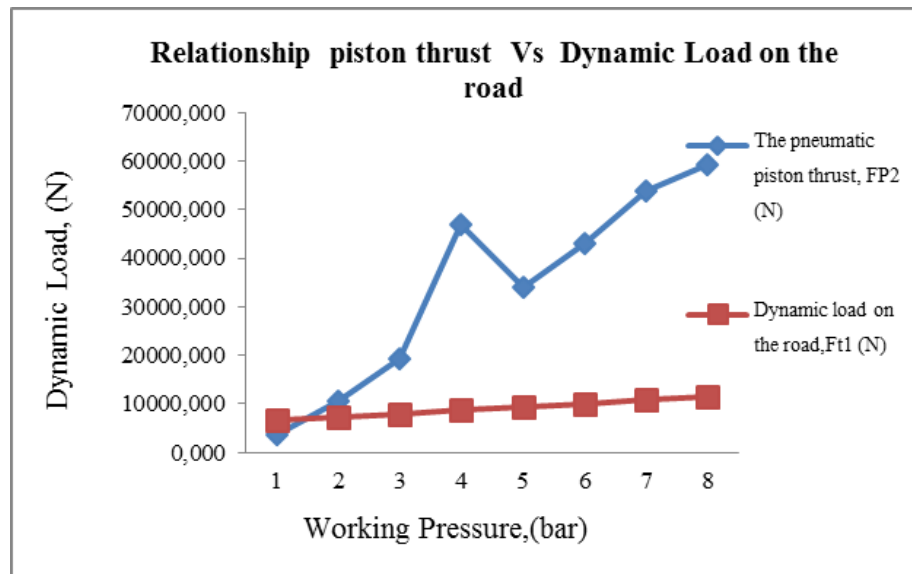
Figure 5. The relationship between piston force, F_{P2} and dynamic load, F_{t1}
which overloads the road structure

3.3. Comparison between experiment and theoretical
The comparison of deviation values between the experimental results and the programming with Mat-Lab is shown in Table 2.

Table 2. 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

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 [2] are 0.03 m and 1.8 m/s^2 respectively and stabilized at $t = 3.5$ (s).

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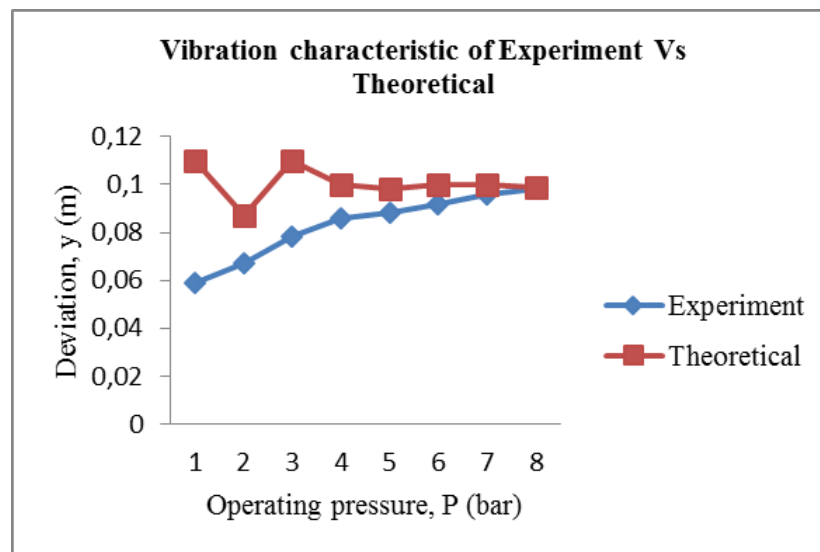
Figure 6. Relationship of vibration characteristics between
experimental and theoretical studies.

4. Conclusion

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:

1. 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.

2. Characteristics of suspension vibration rate on the largest vehicle wheels occur at Y2d = 1.03 m/s when t = 0.22 seconds, while the lowest deviation rate is Y2d = 0.02 m/s when the vibration time t = 2.65 seconds.

Acknowledgement

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