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Identification of Characteristics of Hydraulic Shock Absorbers Used in Light Weight Tracked Vehicles

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Abstract: Shock absorbers have become the most widely used term in automotive suspensions. They are used for the optimization of driving comfort and driving safety. This paper is concerning with identification of characteristics and damping coefficient of the hydraulic shock absorbers of a light weight tracked vehicle under real conditions. Dynamic behaviors of the absorber are studied by both computer simulation and experimental testing. In the experimental testing the force and velocity of the shock absorber are measured on a standard test bench which allow sinusoidal excitation with different frequencies and amplitudes corresponding to road conditions. The simulated model is carried out for the same excitation conditions using ADAMS program. The road roughness is generated in the model using a rotational joint motion applied to revolute joint which is connected to the movable end of the shock absorber. The damping coefficient of the shock absorber is also determined according to the required rate of damping of the vehicle. The predicted characteristic is compared with the experimental results and with those obtained from the computer simulated model.

Keywords: shock absorber, hydraulic, viscous damping, vibration

Introduction

During tracked vehicle motion, shocks are generated due to road roughness and transmitted through tracks to the road wheels causing deformation of the elastic elements which results in hull vibration. This vibration is damped by the action of shock absorbers connected to such wheels. In order to design shock absorbers for resisting impact and attenuating vibration, the accurate characterization of the shock absorber has to be studied. The effectiveness of damping depends on the damping coefficient and number and location of shock absorbers used in the suspension system of vehicles.

In modeling tracked vehicle dynamics, the equation of motion of its hull has to be derived; this modeling can be carried out analytically with a considered number of assumptions or using software package such as Automotive Dynamic Analysis of Mechanical System (ADAMS). Some physical parameters in the equation of motion can be calculated directly while other parameters must be identified from experiments [1]. One of these parameters is that of damping coefficient of the hydraulic shock absorbers used in vehicle suspension system.

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In modeling vehicle systems, many researches are related to the characterization of shock absorbers [2, 3]. The principle of damping in hydraulic shock absorbers depends on transforming the kinetic energy of the oscillating hull into heat energy throughout the damping orifices. In tracked vehicles, the hull vibration has a bad effect on crew comfortability, accuracy of firing on move, and vehicle and undercarriage systems. When shock absorbers get old and lose fluid, the damping coefficient decreases and the suspension system lost its ability to reduce vehicle vibration and after the vehicle passes over a bump, it will oscillate up and down many times rather than just once.

The paper is concerning with identification of dynamic characteristic and damping coefficient of the hydraulic shock absorber used in Armor Personnel Carrier (APC) M113 through three approaches. The first one estimates theoretically the damping coefficient of the shock absorber on the basis of the required rate of damping and the vehicle specifications. The vehicle specifications that affect the needed damping are those of mass moment of inertia, distribution of road wheels, number of shock absorbers that the vehicle equipped, and the stiffness of vehicle suspension. The second approach evaluates the damping characteristics and the damping coefficient experimentally using a standard test bench by measuring the damping force and the velocity of the piston of the shock absorber under different excitation frequencies and amplitudes. The third approach estimates the damping characteristics of the shock absorber by a developed simulated model using ADAMS under real operating conditions. The operating conditions are derived from the nature of excitation of the vehicle hull. This excitation is depending on configuration of the road that the vehicle is moving on and on the vehicle velocity. In the following investigations the three approaches will be discussed and the final conclusion will be obtained.

Theoretical Investigation

For limiting the amplitude and other parameters of vibration, the vehicle suspension is equipped with shock absorbers connected to extreme road wheels. The shock absorber applies damping resistance force resulting in absorption of vibration energy. The damping force can be function of displacement, velocity or frequency and may be linear or nonlinear with respect to these parameters. The shock absorber used in APC M113 is of viscous type. This type of shock absorber is the basic type used in suspension system of tracked vehicles and automobiles. The ideal viscous damper is such one in which the damping force is proportional to the first power of velocity (linear damping) [4]. In APC M113 the dampers are connected to the first, second and last road wheels in order to provide an effective damping moment.

The acting moments on vehicle hull when it is subjected to angular oscillation φ, are indicated in the following Fig. 1.

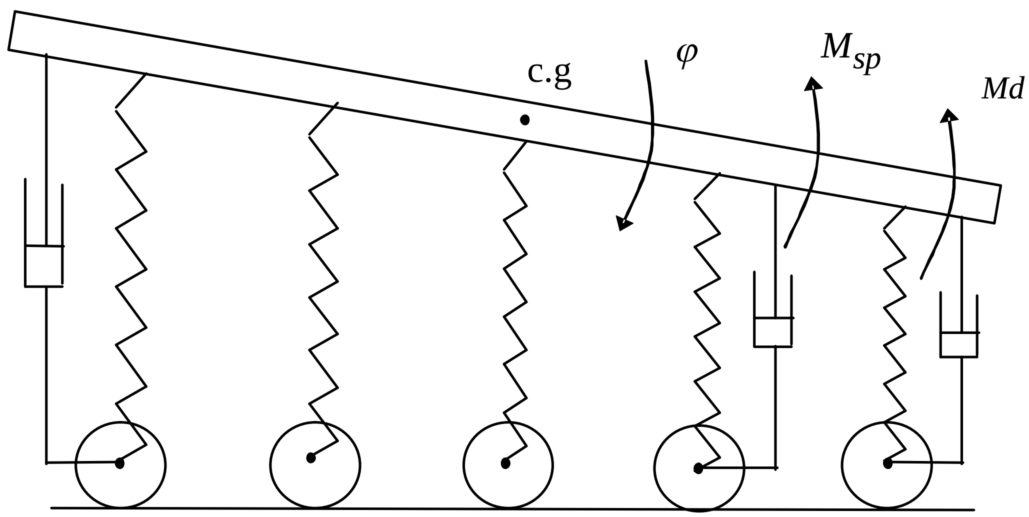


Fig. 1 Acting moments on vehicle hull in damped vibration

The equation of motion of hull in this case will be as follows:

$$J_y\ddot{\varphi} = -M_{sp} - M_d \tag{1}$$

J_y: mass moment of inertia of vehicle about its lateral axes
M_{sp} : moment of springs forces
M_d : moment of damping forces

$$M_{sp} = 2\sum_{1}^{k} F_i l_i \quad , \quad F_i = C_i f_i \quad , \quad f_i = \varphi l_i \tag{2}$$

Fi: force of the ith spring
fi : deflection of the ith spring
C_i: stiffness of suspension system at the ith road wheel
l_i: distance of the ith road wheel from vehicle center of gravity
k : number of road wheels on one side of vehicle (for M113 k=5)
and hence:

$$M_{sp} = 2\varphi\sum_{1}^{k} C_i l_i^2 \tag{3}$$

$$M_d = 2\sum_{1}^{k_d} M_{d\,i} \tag{4}$$

M_{di} : moment of damping force of the ith damper
K_d : number of dampers on one side of vehicle (for M113 k_d = 3)

$$M_{di} = R_i l_i \quad , \quad R_i = \mu V_i \quad , \quad V_i = \dot{\varphi} l_i \tag{5}$$

R_i : damping force of the ith damper

μ : coefficient of shock absorber
V_i : vertical velocity of damper movable part
Hence,

$$M_d = 2\mu\dot{\varphi}\sum_{1}^{k_d} l_i^2 \tag{6}$$

Substituting for M_{sp} and M_d in equation (1), we get:

$$J_y\ddot{\varphi} = -2\varphi\sum_{1}^{k} C_i l_i^2 - 2\mu\dot{\varphi}\sum_{1}^{k_d} l_i^2 \tag{7}$$

Introducing:

$$\mu\frac{\sum_{1}^{k_d} l_i^2}{J_y} = n \qquad \text{damping constant} \tag{8}$$

$$\frac{\sum_{1}^{k} 2C_i l_i^2}{J_y} = \omega_\varphi^2 \qquad \text{square of natural frequency} \tag{9}$$

Then the equation of motion will be:

$$\ddot{\varphi} + 2n\dot{\varphi} + \omega_\varphi^2\varphi = 0 \tag{10}$$

It is a homogenous second order linear differential equation in which its general solution is:

$$\varphi = Be^{-nt}\cos(\omega_d t + \alpha) \tag{11}$$

where:

$\omega_d = \sqrt{\omega_\varphi^2 - n^2}$ frequency of damped oscillation

$T_d = \dfrac{2\pi}{\omega_d}$ period of damped oscillation

The complete solution is obtained by determining the constants B and α from the initial conditions, so that:

$$\varphi = \frac{\varphi_o}{\cos\alpha_o} e^{-nt}\cos(\omega_d t + \alpha_o) \tag{12}$$

φ_o: initial angular displacement of vehicle hull

$$\alpha_o = tan^{-1}\left(\frac{-n}{\omega_d}\right)$$

The graph of complete solution is interpreted on the following figure:

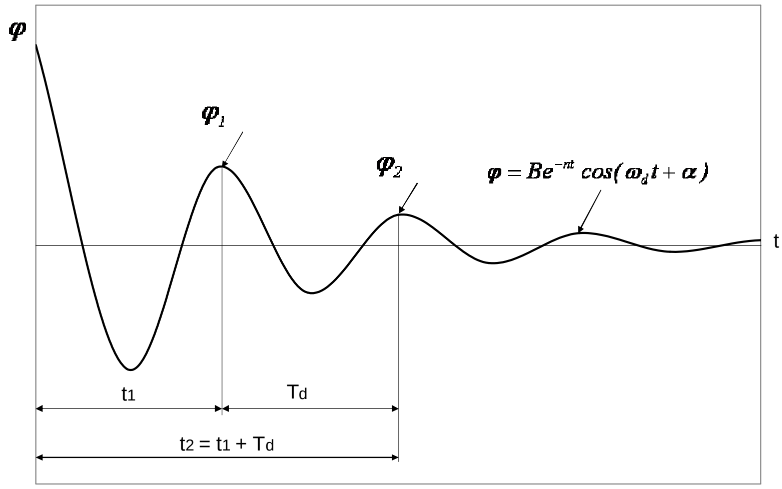


Fig. 2 Response of vehicle hull of free damped vibration

The mean value of shock absorber coefficient μ is determined from the required rate of amplitude of damping represented by the ratio of amplitudes of two successive cycles $\left(\delta = \frac{\varphi_1}{\varphi_2}\right)$. By dividing the response φ at two different periods (t_1, t_2), the following relation of damping constant is obtained:

$$n = \frac{\omega_\varphi \ln\delta}{\sqrt{4\pi^2 + (\ln\delta)^2}} \tag{13}$$

By equalizing the above equation by equation (7), the mean value of shock absorber coefficient is found:

$$\mu = \frac{1.41\ln\delta}{\sum_1^{kd} l_i^2}\sqrt{\frac{J_y\sum_1^k C_i l_i^2}{4\pi^2 + (\ln\delta)^2}} \tag{14}$$

In tracked vehicles and automobiles, the damper coefficient is not constant when the road wheel moves upwards towards the hull and during wheel motion downwards away from hull. The coefficient is smaller when the wheel moves upwards (damper compression) than when wheel moves downwards (damper rebound). Therefore the damper has two coefficients:
μ_1 for compression and μ_2 for rebound, where $\mu_1 < \mu_2$ so,

$$n = \frac{\mu \sum_{1}^{kd} l_i^2}{J_y} = \frac{\mu_1 \sum_{1}^{kd1} l_i^2 + \mu_2 \sum_{1}^{kd2} l_i^2}{J_y} \tag{15}$$

k_{d1} : number of shock absorbers subjected to compression (front absorbers)
k_{d2} : number of shock absorbers subjected to rebound (rear absorbers)

The ratio of coefficients μ_1 and μ_2 are interrelated as $\frac{\mu_1}{\mu_2} = r$, the ratio r has a range between 0.4 and 0.5 as mentioned in [5], hence:

$$\mu_2 = \frac{nJ_y}{r\sum_{1}^{kd1} l_i^2 + \sum_{1}^{kd2} l_i^2} \tag{16}$$

Knowing μ_2 , μ_1 is calculated as $\mu_1 = r\mu_2$

For tracked vehicles the rate of damping δ is considered to be in the range between 12 and 15 [5]. For M113, the following parameters are considered [6]:

$J_y = 24$ kN.m.s^2 , $k_{d1} = 2$, $k_{d2} = 1$, $C_i = 127$ kN/m

So by using equation (14), the mean value of damping coefficient μ , is found to be in the range between 15.5 and 16.5 kN.s/m.

For an average value of $r=0.5$, the damping coefficient in compression μ_1 = 11.5 kN.s/m and in rebound μ_2 = 23 kN.s/m

Experimental work

The shock absorber of APC M113 is tested on a standard test rig (MTS850 damper test system). The MTS850 damper test system is used to measure the performance characteristics of shock absorbers by obtaining the relation between the developed force in the shock absorber with the piston displacement and velocity at different amplitudes and different frequencies.

The loading unit is the main part of the MTS850. It consists of load frame, crosshead lift and lock, force transducer, actuator, three stage servo valve, hydraulic manifold, and accumulators as shown in Fig. 3**.**

The tests performed with the hydraulic test rig were conducted by exciting the shock absorber by a sinusoidal input frequencies equal to (0.5 and 0.7 Hz) with an amplitude equal to 50 mm. Data representing force, displacement, temperature and velocity are collected from several installed sensors on the test rig. This information is fed into instrumentation software to generate graphs that reveal damper performance. The force of the shock absorber and the velocity of the piston versus time were obtained and recorded. The damping coefficient of the tested shock absorber is obtained from the relationship between the applied force and the velocity of the shock absorber piston.

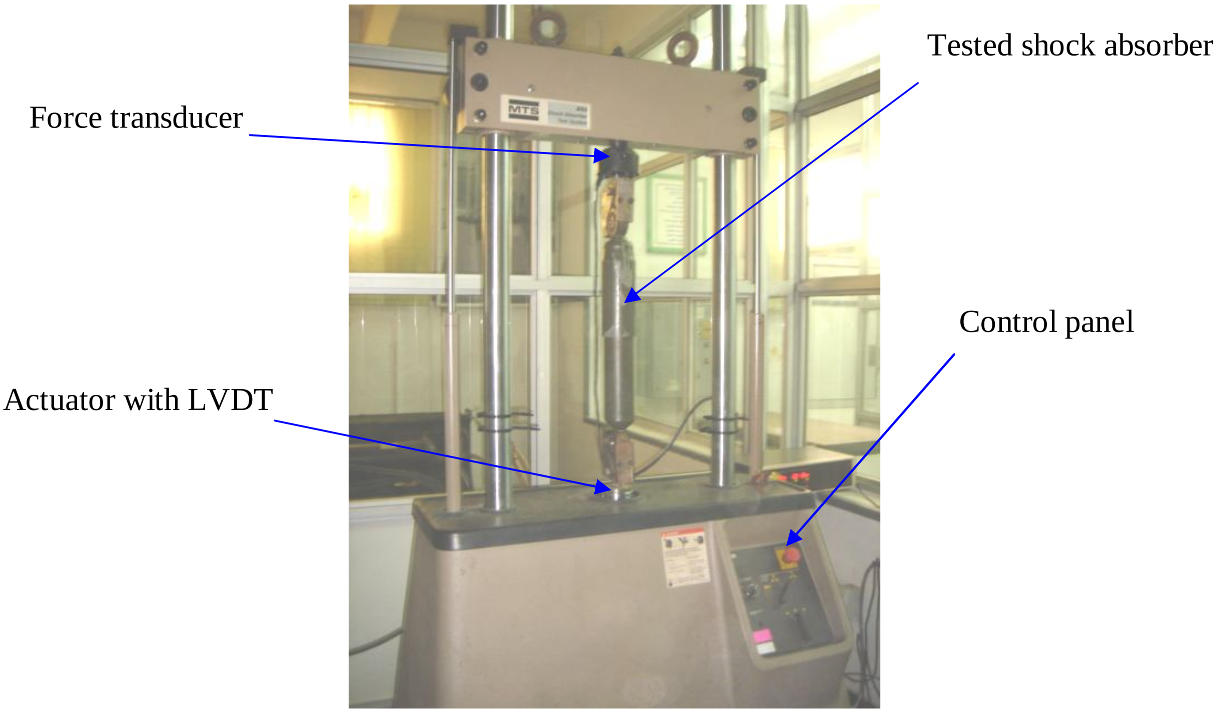


Fig. 3 Photography of damper test system

The rotary motion of the drive motor of the test rig is translated into linear motion of the tested shock absorber. With each compression/rebound cycle, the damper shaft is stroked through a preset displacement range. The Linear Variable differential Transducer (LVDT) follows this movement and yields a VDC signal that relates to speed. In simple linear modeling, the velocity readings are critical because the damping forces that are produced are directly proportional to the damper shaft speed. Fig. 4 shows the main elements of shock absorber of APC M113.

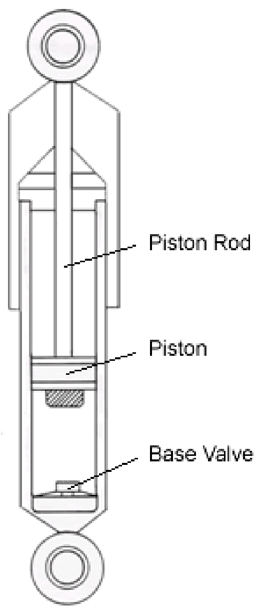
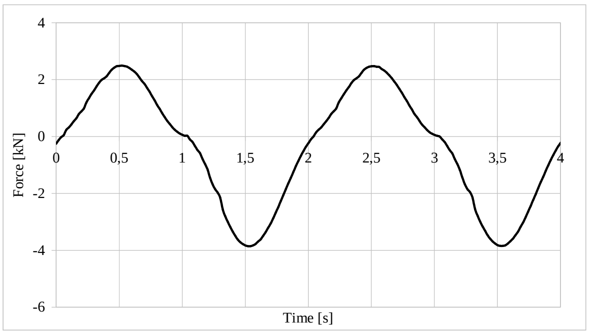


Fig. 4 Tested shock absorber of APC M113

Fig. 5,Fig. 6 represent the measured force and velocity versus the time at frequencies 0.5 and 0.7 Hz respectively with amplitude 50 mm. From these figures it is clear that by increasing the frequency of excitation; the applied force increases at the compression and rebound strokes. This increase in the force is due to increase of the elastic energy stored in the absorber.

By increasing the excitation frequency, the piston velocity of absorber is also increases at the compression and rebound strokes and hence the area of the cycle increases and the damping force obtained at maximum relative velocity decreases as sown in Fig. 7. The rates of increase of the applied force and the absorber velocity determine the amount of linearity of the damping coefficient of the tested shock absorber.
The ratio of damping coefficient in compression and rebound for two frequencies can be observed form the figures to be around 0.6. This value is correlated with ratio r mentioned in the theoretical investigation.



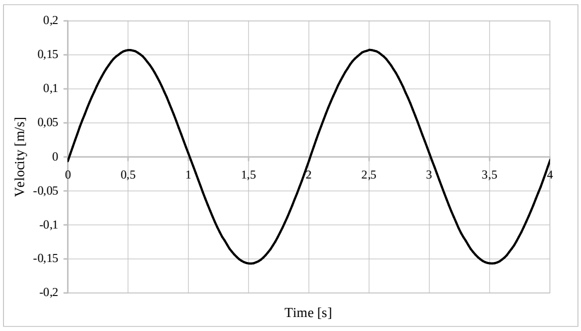
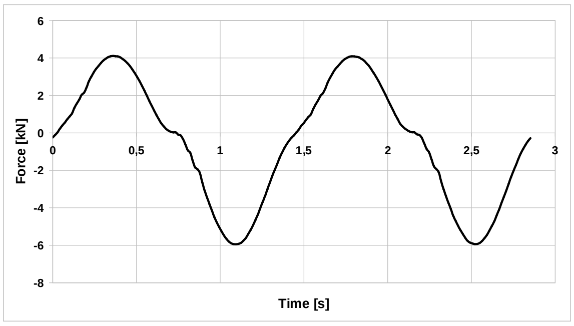


Fig. 5 Variation of force and velocity with time at 0.5 Hz and 50 mm amplitude



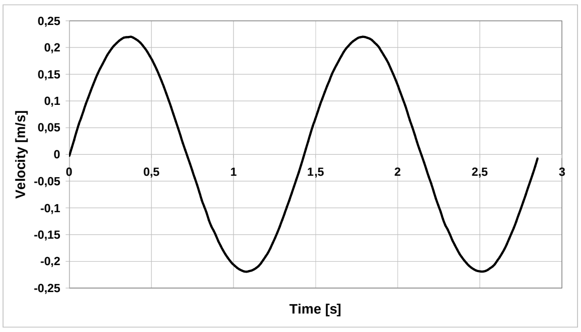


Fig. 6 Variation of force and velocity with time at 0.7 Hz and 50 mm amplitude

From the previous results of variation of force and velocity with time, the variation of force with velocity was conducted. Fig. 7 reveals the variation of the applied force on the shock absorber with time at frequencies equal to 0.5 and 0.7 Hz and with amplitude equal to 50 mm.

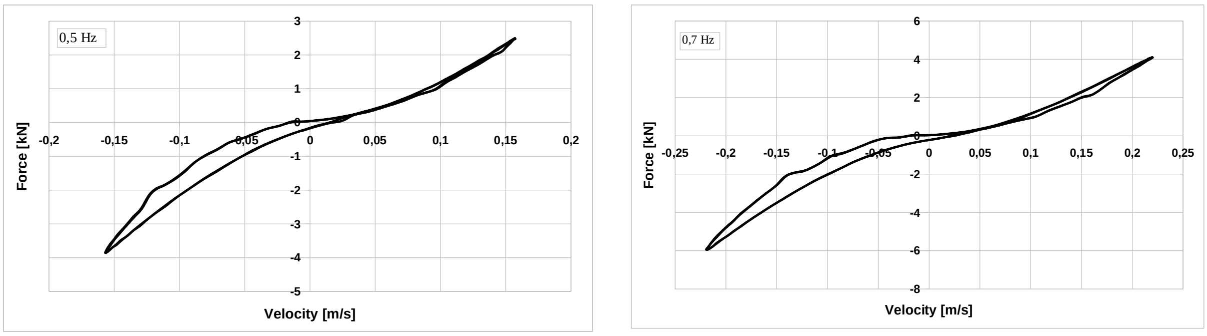


Fig. 7 Shock absorber characteristics at 0.5, 0.7 Hz and 50 mm amplitude

From the data measured for the force and velocity, the damping coefficient of the shock absorber can be obtained by dividing the values of the measured force (for compression and rebound) by the corresponding values of velocity. Table 1 represents the value of the damping coefficient of the shock absorber of APC M113 in compression and rebound stroke at excitation frequencies 0.5 and 0.7 Hz and at excitation amplitude 50 mm.

Table 1 Damping coefficient obtained by hydraulic test rig

Frequency (Hz)	Damping coefficient (kN.s/m)		
	Rebound (μ_2)	Compression (μ_1)	Average value (μ)
0.5	19.8	11.9	15.9
0.7	20.9	13.6	17.2

Simulation Investigation

In this part, the simulation of the dynamic behavior of shock absorber of APC M113 is accomplished through Automotive Dynamic Analysis of Mechanical System (ADAMS) program.

Fig. 8 shows the steps of modeling and simulation of a mechanical system. During tracked vehicle motion, it passes by road unevenness which acts on the hull through the tracks representing an external excitation causing the hull to undergo forced vibrations. As a result of this event, the shock absorber is subjected to an excitation force. Characteristics of such excitation depend on the shape of unevenness and the vehicle velocity. This excitation representing the case of forced damped vibration of vehicle hull.

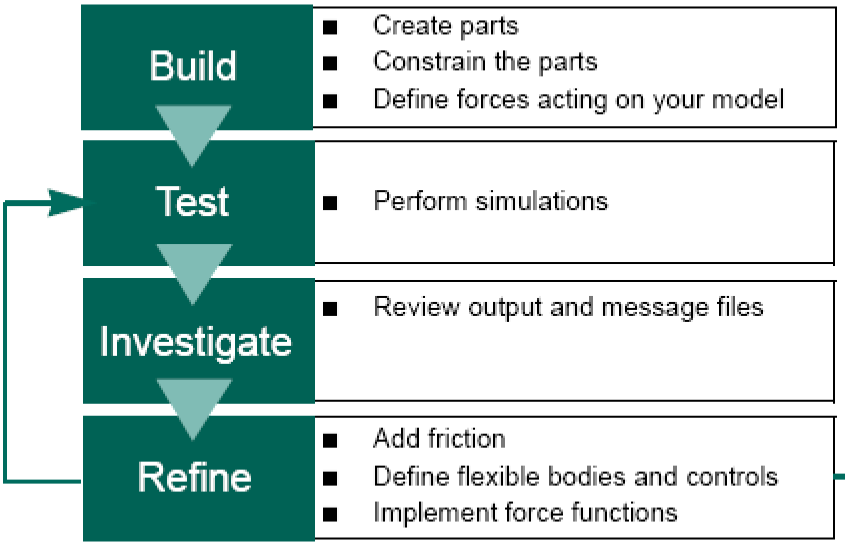


Fig. 8 Steps of modeling and simulation of mechanical systems

The road profile is ultimately changed and it is impossible to simulate exactly this profile. There exist several models for the road profile. One of the most widely applied road models for tracked vehicles, is the model in which the road unevenness is considered to be of sinusoidal shape of length a = 5 – 8 (m) and height h = 0.1 –0.2 (m).

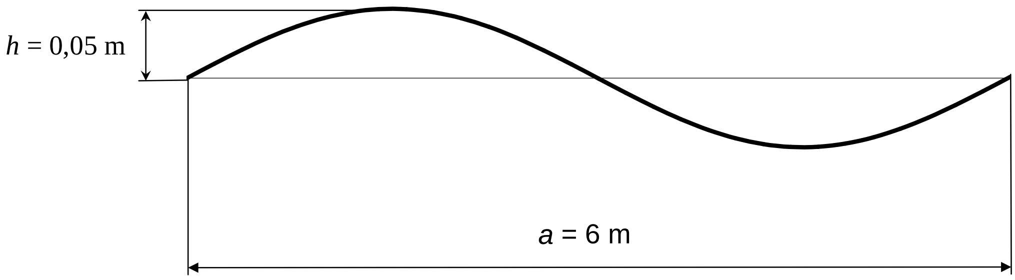


Fig. 9 characteristics of a typical road

Fig. 9 shows a typical type of road way with amplitude 50 mm and length 6 m. The amplitude of excitation is represented by the height of the road (*h*) while the frequency of excitation is depending on the shape of the road and vehicle velocity (*v*) so that it can be calculated as follows:

$$\omega_e = \frac{2\pi v}{a} \tag{17}$$

For a regular vehicle velocity of 12 km/hr and for the given road, the excitation frequency is found to be 0.55 Hz. Hence, the shock absorber is excited by two frequencies (0.5, 0.7 Hz) with amplitude 50 mm. In this model, the shock absorber is excited as it was tested in the experimental investigation as shown in Fig. 10.

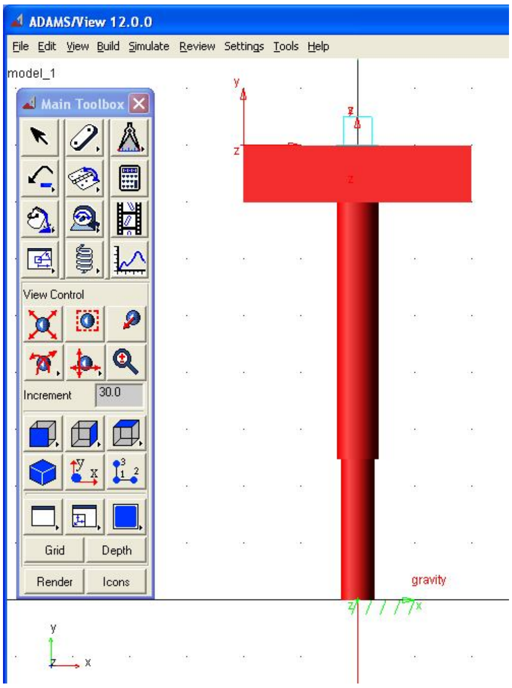


Fig. 10 Simulated model of shock absorber

The amplitude and frequency of excitation is represented by an imposed displacement expressed by the following function:

$$y = y_o \cos \omega t \tag{18}$$

y_o: amplitude of excitation displacement

ω: excitation frequency

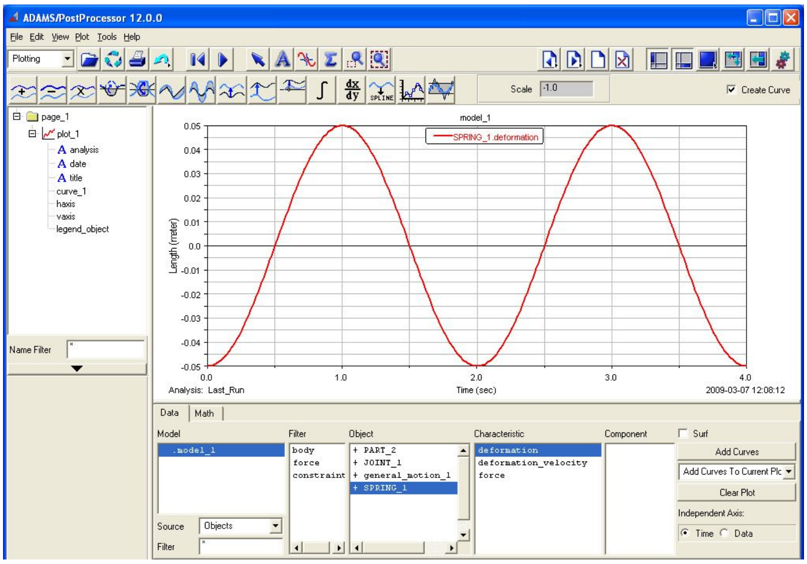


Fig. 11 Imposed displacement on the simulated shock absorber

The amplitude of excitation y_o is expressed by the road height (50 mm) while the frequency of excitation has two values corresponding to 0.5 and 0.7 Hz as mentioned in the experimental part such that ω has 3.14, 4.4 s^{-1} respectively.

The results obtained from the model are represented in Fig. 12,Fig. 13. Fig. 12 illustrates the variation of applied force and velocity with time at 0.5 Hz and 50 mm amplitude while Fig. 13 demonstrates these variations at 0.7 Hz and 50 mm amplitude. By comparing these results with those obtained from experimental investigation, it can be seen that variation of applied force and velocity with time has good agreements with the results obtained experimentally for both frequencies. Variation of the applied force with velocity is obtained by dividing the applied force by the velocity at each time interval. This variation represents the damping coefficient of the shock absorber in both compression and rebound. As mentioned in the theoretical part, the damping coefficient of shock absorbers used in tacked vehicles and automobiles is smaller in compression than in rebound. This difference can be seen from the figures. The positive values of the force and velocity in Fig. 12 and Fig. 13 shows the action of shock absorbers in compression while the negative values show its action in rebound. From both actions, the values of damping coefficient in both compression and rebound are found and compared with the values obtained from experimental results. The ratio of the damping coefficients of shock absorber in compression and rebound gives the predicted value mentioned in the theoretical part which is 0.6. This means that the damping coefficient of the hydraulic shock absorber can be predicted with high accuracy using the relations mentioned before.

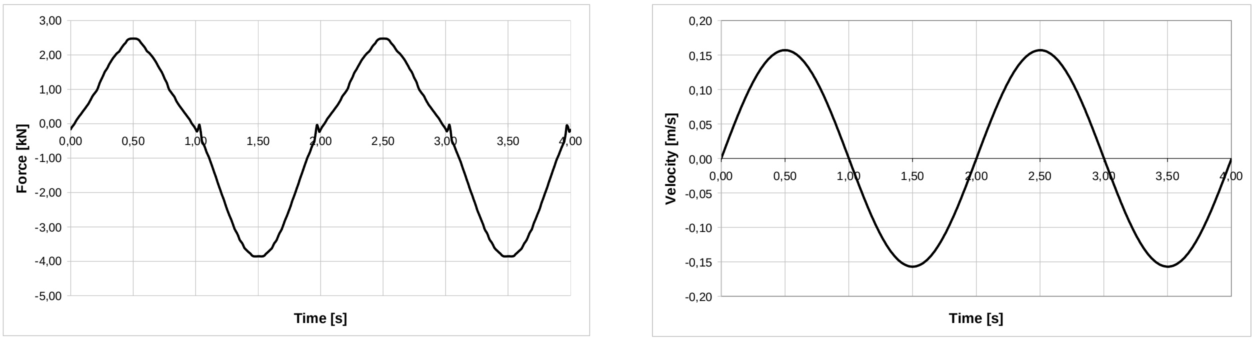


Fig. 12 Variation of force and velocity with time at 0.5 Hz and 50 mm amplitude

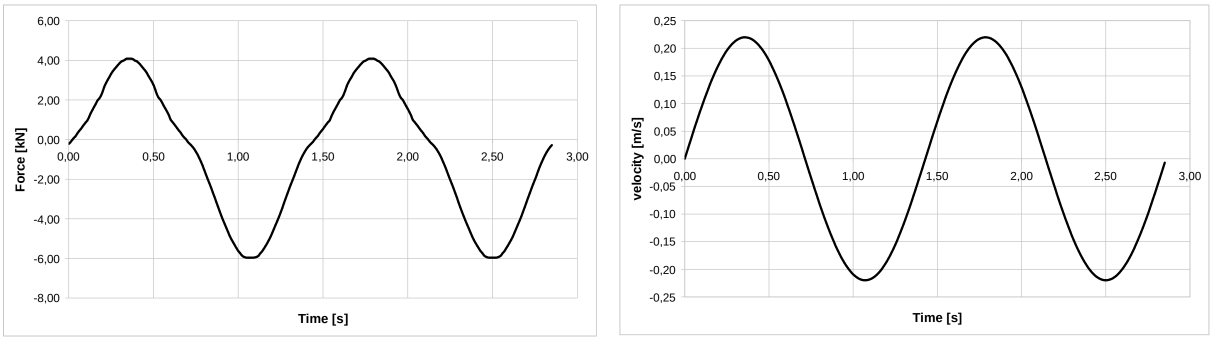


Fig. 13 Variation of force and velocity with time at 0.7 Hz and 50 mm amplitude

Conclusion

The theoretical investigation of the damping coefficient of shock absorber used in tracked vehicles provides a good approach to predict its value in both compression and rebound strokes.

The damping coefficient in both compression and rebound strokes can be estimated using a measured force and velocity obtained from the hydraulic test rig of model MTS 850
The average value of damping coefficient of hydraulic shock absorbers of APC M113 at low excitation frequencies is found to be 12.5 kN.s/m in compression stroke and 20.5 kN.s/m in rebound stroke.

A numerical model using ADAMS has been developed in order to simulate the dynamic behavior of the shock absorber and to describe and evaluate its damping coefficient in compression and rebound cycles.

Evaluation of some parameters like amplitude and frequency of excitation by using the simulated model is discussed. The study shows that these parameters beside the damping coefficient are the key factors that affect the performance characteristics of the shock absorber.

The developed numerical model can be used as a unit of suspension system of tracked vehicles during modeling of dynamic behavior of such vehicles.

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