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EVALUATION OF TRACKED VEHICLES PERFORMANCE CHARACTERISTICS

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ABSTRACT

Performance characteristics of tracked vehicles are the principal measure of their dynamics and it is disturbed by any change in vehicle technical specifications. This paper evaluates the performance characteristics of Armour Personnel Carrier M113 and its variation due to additional forces and masses resulting from increasing of vehicle capabilities such as fire power, degree of armor protection and mobility. Performance of vehicles is determined through number of characteristics like dynamic, acceleration, steering and suspension characteristics. These characteristics are evaluated theoretically using vehicle technical specifications as input data for the vehicle. For suspension characteristics the parameter of damping coefficient of APC M113 shock absorber is measured in a special test rig and is introduced in the equation of motion to find the vehicle response. Accuracy of results is depending on the accuracy of input data and method of calculations. The evaluated characteristics are used for comparison between performance of different vehicles and also for comparison between performances of the same vehicle after increasing its capabilities.

KEY WORDS

Dynamics, Tracked, Vehicle, and Vibration

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NOMENCLATURE

c_i	Reduced rigidity of the i^{th} torsion bar
D	Torque converter diameter
f	Coefficient of rolling resistance.
f_1, f_2	Needed specific force on inner and outer track respectively.
f_c	Coefficient of total road resistance.
f_T	Needed specific driving force for turning.
g	Gravity acceleration (m/s2).
G	Tank weight (kp).
G_s	Weight of suspended part of the vehicle
i_o	Internal gear ratio of double differential steering mechanism.
J_y	Moment of inertia to transverse axis passing through vehicle center of gravity
l_i	Location of the i^{th} road wheel center
L/B	Ratio of track length and track gauge
M_p	Pump torque
n	Damping constant
n_t, n_p	Turbine and pump speed
q	Generalized coordinate
R	Radius of turning.
r_{ds}	Radius of drive sprocket.
R_f	Free radius of turning.
V_o	Vehicle velocity in straight motion (km/h)
W_k	Vehicle kinetic energy
W_p	Vehicle potential energy
$z, \dot{z}, \ddot{z}$	linear displacement, velocity and acceleration of vertical vibration
ω_e	Frequency of excitation force
ρ	Fluid density of torque converter
$\psi, \dot{\psi}, \ddot{\psi}$	angular displacement, velocity and acceleration of lateral angular vibration
$\varphi_o, \dot{\varphi}_o$	Initial displacement and velocity of longitudinal vibration
$\varphi, \dot{\varphi}, \ddot{\varphi}$,	angular displacement, velocity and acceleration of longitudinal angular vibration
ω_z, ω_φ	Natural frequencies in vertical and longitudinal vibration
μ, μ_{max}	Coefficient of resistance to turning, maximum value.
ζ	Shock absorber damping coefficient
λ_p	Characteristics coefficient of torque converter
ρ_y	Radius of gyration about y axis

1. INTRODUCTION

High mobility tracked vehicles such as armored personnel carriers are designed for mobility over rough off road terrain surfaces. The mobility performance of these vehicles is often limited by the operator's endurance to withstand the transmitted shocks and vibrations, and his ability to maintain control. The maximum allowable vehicle speed varies with the roughness of the terrain, and is primarily influenced by the suspension system design. High mobility tracked vehicles are generally fitted with passive suspension systems utilizing torsion bars and shock absorbers to attenuate the terrain induced shocks and vibrations [2]. Tracked vehicles upgrade results in change in vehicle technical specifications like weight, speed, and other main characteristics. This change in vehicle specifications affects its performance and limits vehicle mobility and maneuverability. So it was very important for tracked vehicles to evaluate a number of characteristics describing its performance. The principal and most important characteristics that describe vehicle performance are traction or dynamic characteristics, acceleration characteristics, steering characteristics and suspension characteristics [4]. The traction characteristics illustrate the relationship between specific tractive effort developed by the engine and the vehicle velocity for each speed gear. The acceleration characteristics are defined by the variation in vehicle velocity related to time when moving from standstill on subsequent speed gears up to the maximum possible velocity for a certain road, so it measures the time to reach vehicle maximum speed. The steering characteristics are defined by the variation of the specific tractive effort needed to be developed by the engine with the radius of turning for a specific road. Finally the suspension characteristic describes vehicle vibration as a function of time including the effect of suspension elements parameters such as torsion bar stiffness and shock absorber damping factor. In this paper a detail explanation for dynamic, steering and suspension characteristics for APC M113 will be carried out using the collected and measured data of the vehicle. Comparison of these characteristics will be performed for different cases to indicate the effect of added masses. The suspension system parameters like torsion bar stiffness and shock absorber damping factor are characterized and measured using a special test rig and introduced in the equation of motion of the vehicle to find its response [1].

2. THEORETICAL INVESTIGATION

The theoretical investigation of performance characteristics includes that of dynamic, steering, and suspension characteristics of the vehicle. Hereafter each characteristic will be discussed.

2.1. Dynamic Characteristics

As mentioned before dynamic characteristic represents the relationship between the specific tractive effort developed by the engine f_d and the vehicle velocity V. So it was necessary to find for each value of engine speed n_e and for individual speed gears, the driving force f_d and vehicle velocity V. These parameters are expressed by the following relations:

$$f_d = \frac{270 N_e \eta_T}{GV}, \qquad V_i = \frac{\pi\, n_e}{30\, i_{Ti}} r_{d.s}$$

The suffix i in the velocity equation, represents the i^{th} speed gear while the total transmission ratio i_{Ti} represents the transmission ratio between the engine and driving sprocket including that of hydraulic torque converter, gear box, steering mechanism, and final drive.

Calculations of dynamic characteristics are based on engine speed characteristic ($N_e, M_e = f(n_e)$) obtained by laboratory testing and it can be given in a graphical form as shown in Fig. 1.

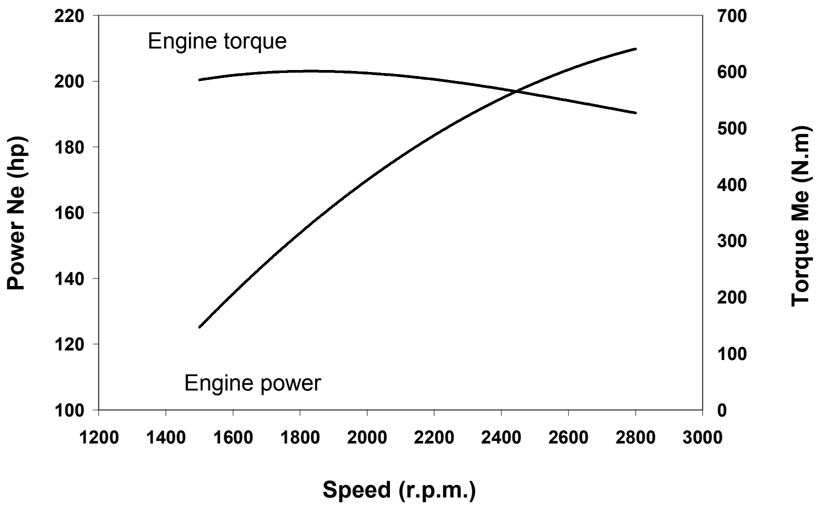


Fig. 1: M113 Engine characteristics

Dynamic characteristics of vehicles equipped with hydromechanical transmission mechanism, like that of M113, differ from that of mechanical transmission due to existence of hydraulic torque converter. So its characteristics are determined on the basis of cooperation characteristics between the engine and torque converter. To find this characteristic the torque converter characteristic is evaluated according to the following relation:

$$M_p = \lambda_p \rho D^5 n_p^2$$

λ_p represents the characteristic coefficient of torque converter and it depends on converter slip value so that for each value of converter speed ratio (n_t / n_p) there is a corresponding value of λ_p as shown in Table 1.

Table 1. Characteristics of M113 torque converter

n_t / n_p	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	0.95
$\lambda_p * 10^5$	1.99	2.14	2.28	2.4	2.49	2.54	2.54	2.46	2.25	1.55	1.02

Fig.2 represents the cooperation characteristic between the engine and torque converter for different values of speed ratio n_t / n_p.

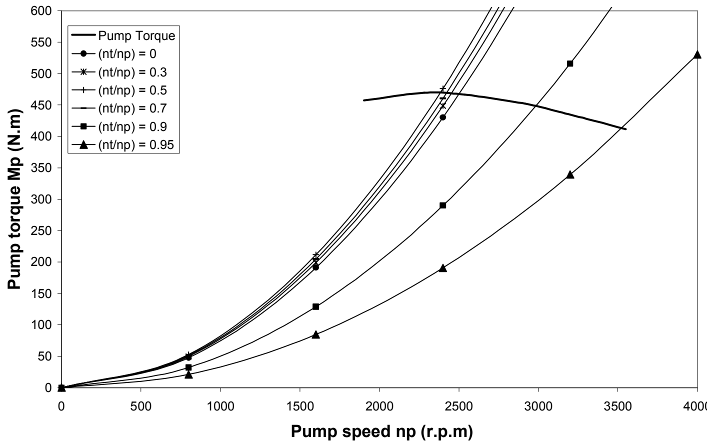


Fig.2: Engine –torque converter cooperation characteristics

The dynamic characteristic of APC M113 is represented graphically in Fig.3. The Figure shows the effect of increased vehicle weight by about 10% of the original weight.
The collected data of APC M113 transmission mechanism from its technical manuals and from the actual transmission helps in estimating this characteristic.

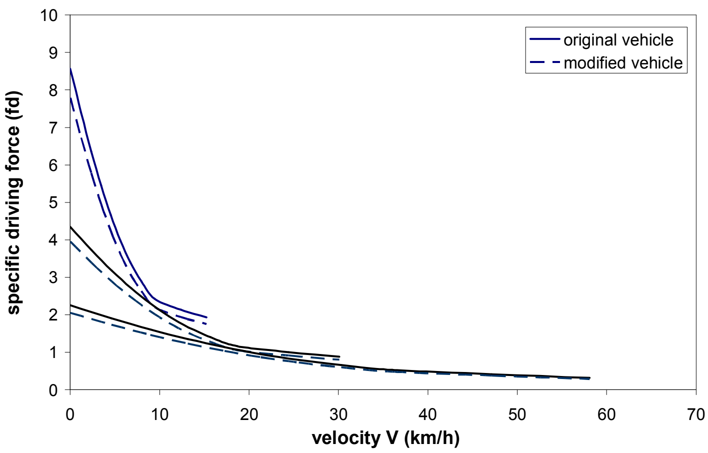


Fig.3: Dynamic characteristics M113

It can be seen that the specific driving force for the modified vehicle is lower than that of the original one. If two vehicles with the same type but with different weights are driven with the same velocity, the driving force of the vehicle with less weight will be higher than that of the vehicle with higher weight and if the two vehicles are driven with the same specific driving force, the velocity of the lower weight vehicle will be higher than that of the higher weight. So the additional masses have an adverse effect on the dynamic characteristics and must be limited to a certain value to reduce its effect.

2.2 Steering Characteristics

Steering characteristics are defined by the variation of the specific tractive effort f_T ,needed by the engine to perform turning, with the radius of turning R, so that $f_T = f(R)$. The needed specific driving force for turning f_T depends on the type of steering mechanism, vehicle geometric configuration (L/B) and type of terrain and it is calculated according to the following relation:

$$N_e = f_T \frac{G V_o}{270 \eta_T}$$

Where, N_e is the power consumption by the steering mechanism during turning.
The steering mechanism of M113 is a double differential steering mechanism with internal gear ratio i_o. So the coefficient f_T can be calculated according to the following formula [4]:

$$f_T = f_2(1+i_o) - f_1(1-i_o)\eta_R$$

Where:

$$f_2 = 0.25\mu\frac{L}{B} + 0.5f \qquad f_1 = 0.25\mu\frac{L}{B} - 0.5f \qquad \mu = \frac{\mu_{max}}{0.85 + 0.15\frac{R}{B}}$$

Fig. 4 demonstrates steering characteristics of APC M113.

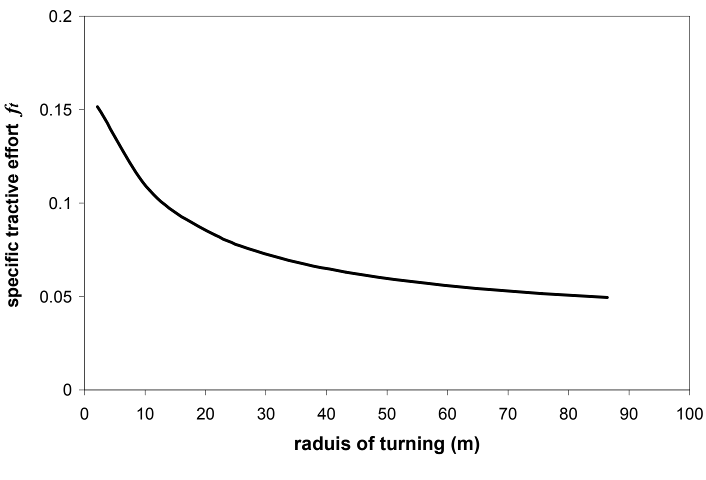


Fig. 4: Steering characteristics of APC M113

The associated utilization of steering and traction characteristics provides the possibility of evaluation of the turning capabilities. This can be defined by the vehicle velocity when turning at a specified radius. This velocity is determined from the equilibrium of f_T and f_d at such radius or by vehicle radius when moving with a specified velocity.

Utilization of dynamic and steering characteristics for the original and modified vehicle is shown in Fig. 5. This figure is utilized to identify the speed gear that the vehicle will move when it is driven at a specified radius of turning. It is seen that from the figure if the two vehicles (original and modified) are moving with a specified velocity we will get two radii of turning for each vehicle. The radius of turning of the original vehicle is lower than that of the modified vehicle. In other words, if the two vehicles are turning at a specified radius the modified vehicle will have a smaller velocity than the original one. So the increased mass decreases the vehicle performance from the point of view of steering characteristics.

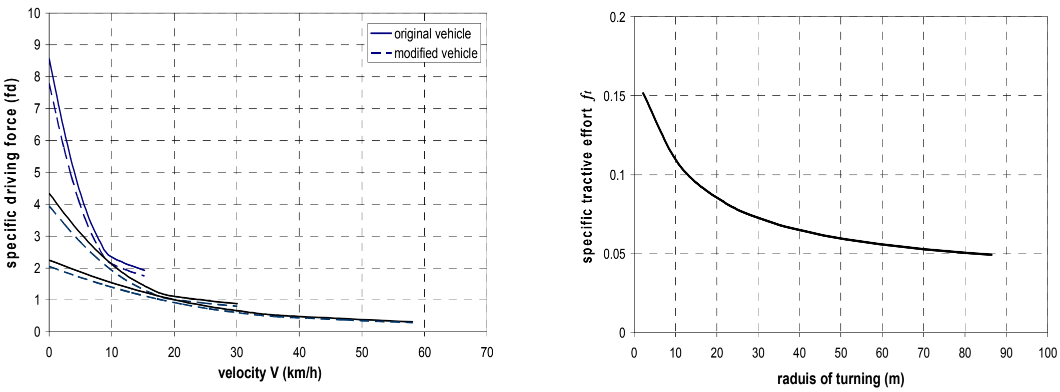


Fig. 5: utilization of dynamic and steering characteristics

2.3 Suspension Characteristics

Suspension characteristics is the principal measure of vehicle vibration, and it depends on the suspension elements parameters like torsion bar stiffness, shock absorber damping coefficient and any other elastic elements that connect the vehicle road wheels with the hull. When the vehicle is driving on an uneven road, the elastic elements (springs, torsion bars) are deformed resulting in hull vibration.

In this approach the vehicle is considered as a system of six degrees of freedom which are $x,y,z,$ φ,ψ,ζ as shown in Fig. 6.

Practically the vibration along x,y,ζ are neglected due to their low effect, so the vehicle hull undergoes mainly the following types of vibration:

Longitudinal angular vibration with angular displacement φ

Lateral angular vibration with angular displacement ψ

Vertical vibration with linear displacement z

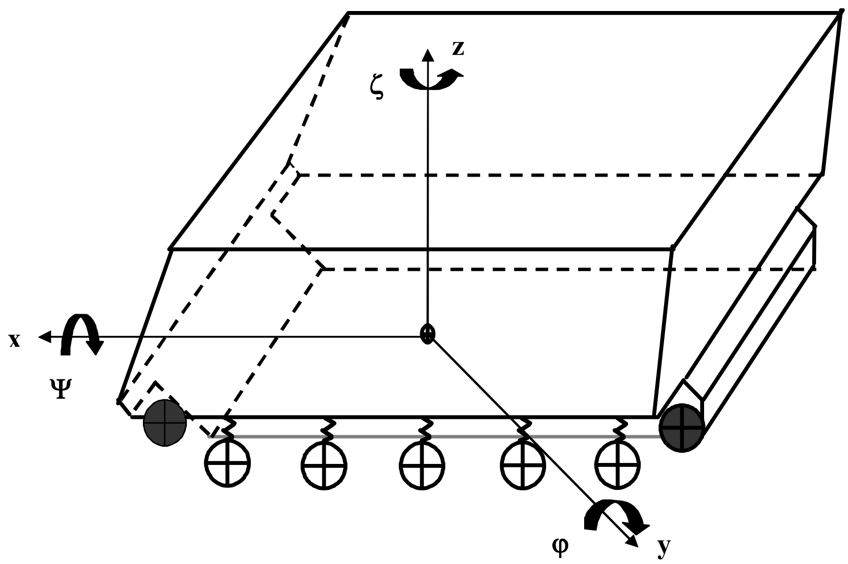


Fig. 6: Schematic diagram of APC M113

Lateral angular vibration ψ occurs less frequently with respect to the others two types of vibration mentioned above. Even if it is occurred it will be damped by the side shift of road wheels on tracks or by the tracks on the ground.

Equation of vehicle motion in the remaining two coordinates is derived from Lagrange equation [2,5]:

$$\frac{d}{dT}\left(\frac{\partial W_k}{\partial \dot{q}_i}\right) - \frac{\partial W_k}{\partial q_i} = -\frac{\partial W_p}{\partial q_i}$$

By evaluating the kinetic energy and the potential energy of the previous equation, the vehicle general equations of motion in z and φ coordinates can be expressed as follows:

$$\ddot{z} + a\dot{z} + b\varphi = 0$$

$$\ddot{\varphi} + \frac{e}{\rho_y^2}\dot{\varphi} + \frac{b}{\rho_y^2} z = 0$$

where:

$$a = \frac{1}{m_s}\sum_{1}^{k} 2c_i \qquad b = \frac{1}{m_s}\sum_{k_2}^{k_1} 2c_i l_i \qquad e = \frac{1}{m_s}\sum_{1}^{k} 2c_i l_i^2 \qquad \rho_y^2 = \frac{J_y}{m_s}$$

From Fig. 6, the vehicle suspension system can be considered as a symmetrical suspension with respect to its center of gravity. In this case the coefficient b in the previous equations of motion is equal to zero and the equations can be rewritten as:

$$\ddot{z} + a\dot{z} = 0$$

$$\ddot{\varphi} + \frac{e}{\rho_y^2}\dot{\varphi} = 0$$

where $\sqrt{a}$, and $\sqrt{\dfrac{e}{\rho_y^2}}$ represent the natural frequencies, ω_z and ω_φ , of the vehicle in z and φ coordinates respectively so that:

$$\omega_z = \sqrt{\frac{\sum_1^k 2c_i}{m_s}} \qquad \text{and,} \qquad \omega_\varphi = \sqrt{\frac{\sum_1^k 2c_i l_i^2}{J_y}}$$

by solving the last two equations of motion, their response can be expressed as follows:

$$z = A\cos(\omega_z t + \alpha)$$
$$\varphi = B\cos(\omega_\varphi t + \alpha)$$

Where:
A and *B* are constants determined by the maximum stroke of road wheels, for M113 $A \cong 0.15$ m and $B \cong 0.052$ radians.

To find the previous response, the reduced rigidity of torsion bar c_i was evaluated according to the ratio between the vertical load acting on road wheels P_k to its vertical displacement f_k and it is shown in Fig. 7

$$c_i = \frac{P_k}{f_k}$$

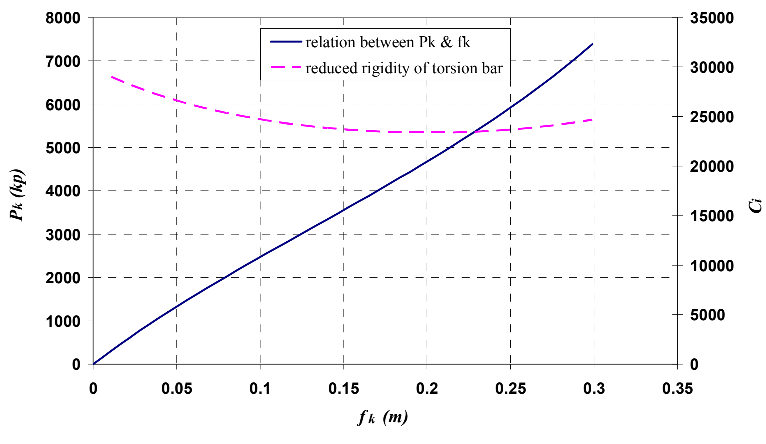


Fig. 7: Torsion bar characteristics of APC M113

From the figure, it can be seen that, the average value of the reduced rigidity of torsion bar is a quasi constant and its value is found to be 248 Kp/ cm. By substituting this value in the response equations, in z and φ coordinates, in case of free undamped vibration, the response can take the form shown in Fig. 8.

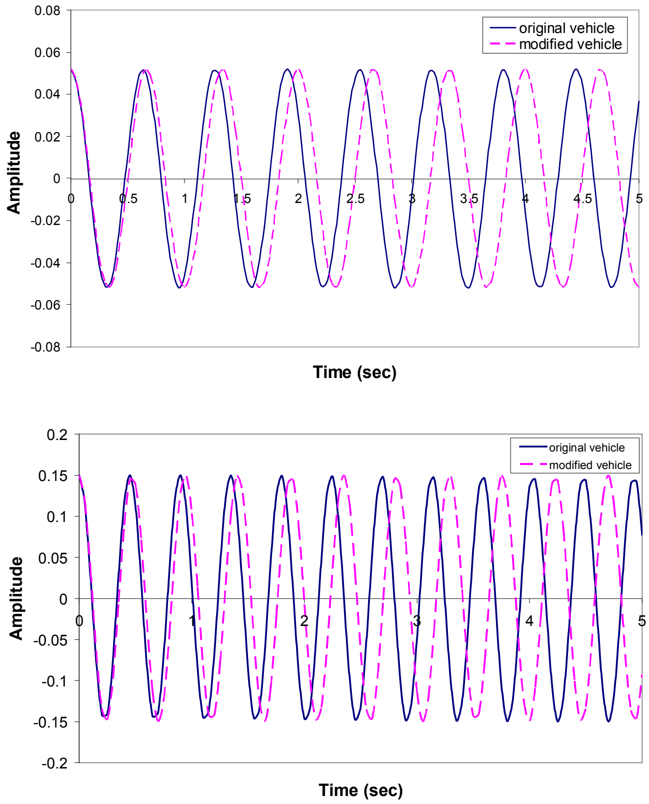
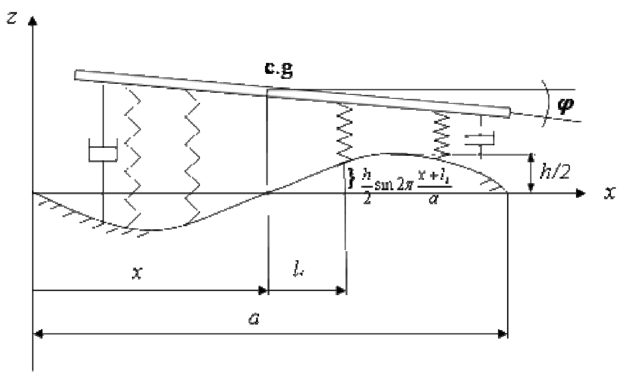


Fig. 8: Dynamic response of vehicle in free undamped vibration in z, φ coordinates

The general equation of motion of vehicle in case of forced damped vibration is derived from forces and moments acting on vehicle hull and it is expressed by:

$$\ddot{\varphi} + 2n\dot{\varphi} + \omega_{\varphi}^{2}\varphi = E\cos qt + D\sin qt$$

Where E and D are coefficients expressing the driving moments in dependence on the parameters of suspension system and profile of roadway.



The previous equation is a non- homogenous differential equation and its solution is equal to the sum of general integral φ_1 of homogenous equation and particular integral φ_2 of non homogenous equation so that:

$$\varphi = \varphi_1 + \varphi_2$$

The general integral φ_1 of homogenous equation is represented by:

$$\varphi_1 = e^{-nt}(c_1 \cos\sqrt{\omega_\varphi^2 - n^2}\ t + c_2 \sin\sqrt{\omega_\varphi^2 - n^2}\ t)$$

while the particular integral φ_2 of non homogenous equation is:

$$\varphi_2 = M\cos qt + N\sin qt$$

Where:

$$M = \frac{E(\omega_\varphi^2 - q^2) - 2nqD}{(\omega_\varphi^2 - q^2)^2 + 4n^2q^2}, \qquad N = \frac{D(\omega_\varphi^2 - q^2) + 2nqE}{(\omega_\varphi^2 - q^2)^2 + 4n^2q^2},$$

$$c_1 = \varphi_0 - M \text{ ,and} \qquad c_2 = \frac{\dot{\varphi}_0 + n\varphi_0 - nM - qN}{\sqrt{\omega_\varphi^2 - n^2}}$$

So the general solution can be expressed as follows:

$$\varphi = e^{-nt}(c_1 \cos\sqrt{\omega_\varphi^2 - n^2}\ t + c_2 \sin\sqrt{\omega_\varphi^2 - n^2}\ t) + M\cos qt + N\sin qt$$

To represent the previous response graphically, the damping constant n must be evaluated. The shock absorber of APC M113 is connected to a special test rig to measure its damping coefficient η. It is excited with different frequencies (0.2, 0.3, and 0.4 Hz) corresponding to road irregularities and the measured data are represented in the following Figure.

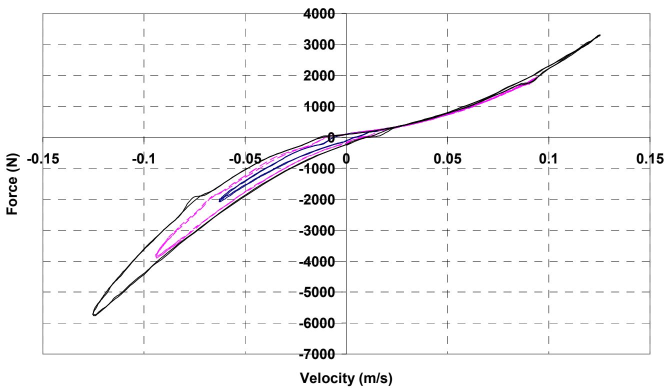


Fig. 9: Force - velocity dependent of M113 shock absorber

From the previous Figure, it is seen that the damping coefficient of the shock absorber has different values in compression and rebound stroke. The average value of damping coefficient η is found to be 14.7 KN.s/m so the damping constant n needed for the response equation is calculated according to the following relation [5]:

$$n = \frac{\eta \sum_{1}^{k_t} l_i^2}{Jy}$$

M113 contains two shock absorbers in each side, so by calculating the moment of inertia J_y for the vehicle, the value of damping constant n is found to be 2.32
the effect of the added mass is clear in the damping constant n so that as the vehicle mass increases the mass moment of inertia of the vehicle will also increase and this will affect the damping constant n so that to maintain the same level of vibration of the vehicle when its weight is increased, the damping constant is required to be increased which means that shock absorber with highest damping coefficient must be used. The effect of the increased mass is shown in the following Figures.
Fig. 10 represents the response of the vehicle in free damped vibration which represents the general integral solution φ_1 while Fig. 11 represents the particular integral solution φ_2 in forced excitation.

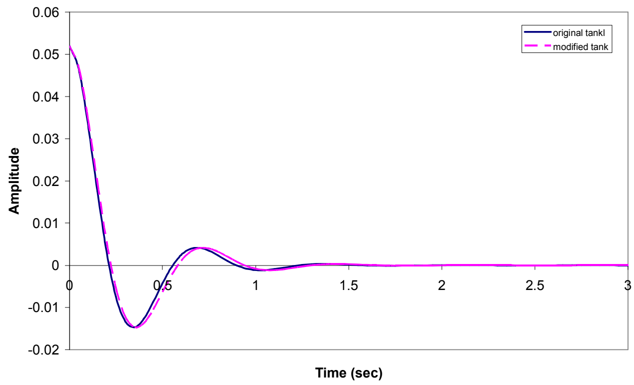


Fig. 10: particular integral response of free damped vibration

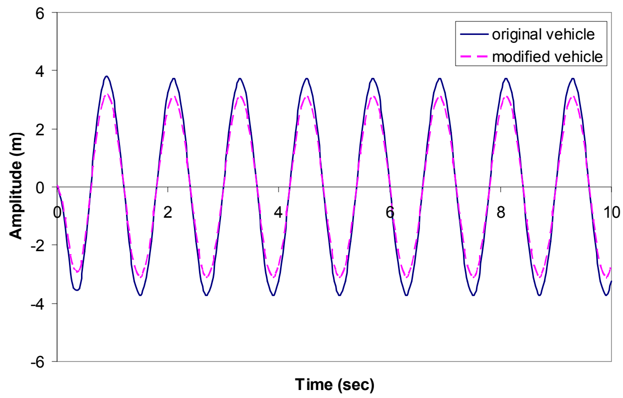


Fig. 11: Particular integral response of forced excitation

Due to the effect of the first component (general integral component) of the response equation the hard shocks exerting on the vehicle are reduced, and the particular integral solution is remained with time due to the existence of the excitation force.

3. CONCLUSION

1. In this paper a complete investigation for the performance characteristics of the Armoured personnel carrier M113 is carried out including that of dynamic, steering, and suspension characteristics. These characteristics are evaluated using the specifications of the vehicle collected and measured from the vehicle mechanisms and from its manuals.
2. Measuring of damping coefficient of vehicle shock absorber on an effective test rig is very useful in evaluating the suspension characteristic of the vehicle.
3. Effect of add on masses and changing of vehicle specifications is studied and it is found that these changes affect the vehicle performance and it must be limited to a certain limits. This effect is very clear on the dynamic characteristics which explain the relation between the specific driving force and the vehicle velocity.
4. Increasing the vehicle weight by about 10% of its original weight can disturb its performance characteristics but with an acceptable level depending on user requirements.

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