

Dynamics of a Cannon Barrel-Recoil Mechanism with Nonlinear Air-Springs

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ABSTRACT: The objective of this paper is to investigate the dynamics of the barrel assembly-recoil mechanism of military cannons when using air springs and a constant damping coefficient hydraulic damper in their recoil mechanisms. The elastic characteristics of the air spring is nonlinear and the recoil mechanism orientation introduces extra nonlinearity to the dynamic model of the system. An extremely nonlinear model of the barrel assembly is derived and solved using Runge-Kutta 4 method to provide the dynamic response of the barrel assembly upon firing. The simulation results using the data of a Howitzer M114 cannon are presented for recoil mechanism orientation ≤ 50 degrees. The performance of the recoil mechanism is evaluated through the minimum and maximum displacements of the barrel assembly and the settling time of its response upon firing. The effect of the number of air springs on the performance of recoil mechanism is investigated. The analysis shows that it is possible with air springs to obtain barrel assembly response similar to that of a critically damped second-order system. It is possible with proper selection of the recoil assembly parameters to decrease the maximum barrel displacement to 54 mm and the settling time to less than 2 seconds.

KEYWORDS: Cannon recoil mechanism, Barrel assembly dynamics, Air springs, Nonlinear dynamic model, Barrel response upon firing, Recoil mechanism performance.

1 INTRODUCTION

Researchers pay deep attention to the analysis and design of artillery recoil mechanisms. Hogg (2000) described about 300 artillery pieces from 1900 to 2000 with full dimensions, mass, ammunitions and range details, country of origin and muzzle velocity [1]. Ahmadian, Appleton and Norris (2002) used a MR damper to control the recoil dynamics. The suggested technique for using MR dampers for free out of battery. They used a recoil demonstrator including a 0.5 caliber gun and a MR damper [2]. Slizys (2005) studied the dynamic characteristics of the plane motion of the recoil of the automatic rifle AK-4. He formulated the mathematical models of the recoil plane motion and obtained its dynamic characteristics [3]. Choi, Hoo and Wereley (2005) examined the use of a double adjustable MR damper to produce high damping force over a high speed piston range. They proposed an on-off control algorithm to improve the shock mitigation of the passive MR gun recoil system [4]. Bao-lin (2006) studied the use of a gun recoil MR damper for a gun test application. He constructed a one-dimensional parallel-plate laminar flow model for the damper based on Herchel-Bulkley shear model and obtained the damping characteristic curves for the damper and evaluated its performance [5]. Lin et.al. (2009) derived a mathematical model for the recoil force during firing. They claimed that their results provide a clear understanding for designing the recoil mechanism and improve its performance [6]. Xue-zheng, Jiong and Hong-sheng (2010) designed a large-stroke MR impact damper which can work effectively at the large velocities occurring in artillery recoil. They showed that the MR damper is able to effectively control the recoil dynamics in terms of recoil force and stroke [7]. Gimm, Cha and Cho (2012) investigated experimentally the behavior of shock vibration for a medium caliber gun barrel. They applied the numerical modal analysis, signal processing and shock response analysis techniques in their analysis [8]. Ting, Lu and Rui (2013) designed a deflection system for the breech block and anti-recoil mechanism. They realized the technical indicators detection under the recoil process using advanced hydraulic control technology [9].

Air springs find great attention from a lot of researchers because of its simple design and control. Presthus (2002) developed a mathematical model for air springs. He determined five of the model parameters using thermodynamics and

fluid dynamic constitutions. He presented the air spring stiffness as function of the vibration frequency [10]. Wei-min, Can-hui, Ya-ling and Yan-sha (2004) studied the deformation and static characteristics of an air spring used in automotive suspension. They compared the characteristic obtained by finite elements to that obtained experimentally [11]. Deo and Sah (2006) proposed a design for a pneumatic automotive suspension system. They showed that the air spring stiffness is function of pressure, volume and temperature [12]. Silva and Costa (2008) developed empirical models relating force acting on the air spring to its deflection and internal pressure for 4 different springs [13]. Koizumi et. al. (2009) proposed an air suspension system coupled with a rotary damper to attain low dynamic stiffness and high damping coefficient. They attained an extremely low dynamic stiffness at a specific frequency [14]. Zhang and Yang (2010) established a nonlinear model for a pneumatic vibration isolator considering the volumetric compressibility of air. They showed that the system has a strong nonlinear characteristics [15]. Todkar and Joshi (2011) discussed the effect of the variation of mass ratio, air damping ratio and air spring stiffness on the motion transmissibility at the resonant frequency of the main system. They designed a control system to vary the air pressure in the damper of the absorber system [16]. Robinson (2012) investigated the use of a pneumatic suspension system containing an air spring, air flow valve and an accumulator where the air spring and accumulator provide the spring characteristics and the valve provides the damping characteristics [17]. Chen et. al. (2013) formulated a nonlinear model of a multi-axle semi-trailer with longitudinal-connected air suspension based on fluid mechanics and thermodynamics. They showed that the influence of air line diameter on load-sharing is more significant than that of the connector [18]. Razdan, Bhawe and Awasare (2014) used an active pneumatic suspension with control strategy based on mass flow control for a commercially manufactured small car. They concluded that the active suspension using velocity feedback as the control strategy has better performance at resonance [19].

2 ANALYSIS

The dynamic system of any cannon-recoil mechanism dynamic system is considered as a mass-damper-spring system. Fig.1 shows a typical Howitzer M114 155 mm cannon model [20].



Fig.1 The Howitzer M114 155 mm cannon [20].

The equivalent dynamic model is shown in Fig.2 for the barrel assembly and the recoil mechanism.

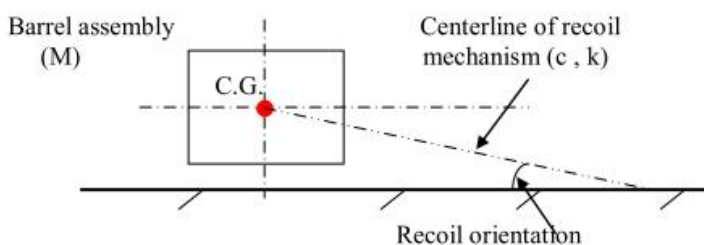


Fig.2 Cannon equivalent dynamic system.

The barrel assembly has a center of mass, G that translates horizontally by a dynamic motion x . The recoil elements are joined to the barrel assembly and the main cannon chassis secured to the ground and has an original orientation θ_0 with the ground. As the gun fires, the barrel assembly moves in the opposite direction of the projectile horizontal motion component with initial velocity depending on the projectile momentum and barrel assembly mass.

The recoil mechanism takes this barrel assembly momentum and tries to return the barrel assembly to the original position before firing to start a new firing cycle. The motion of the equivalent dynamic system upon firing is illustrated in Fig.3.

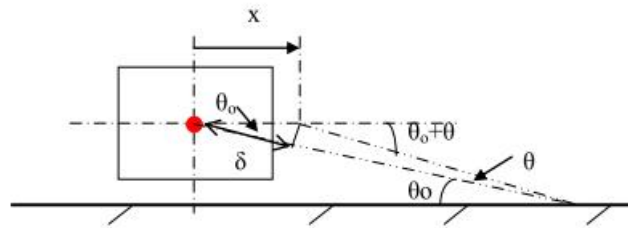


Fig.3 Dynamic system motion.

The dynamic system has the motions and orientation:

x : dynamic motion of the barrel-assembly center of mass (horizontal).

δ : dynamic deflection of the recoil elements.

θ : dynamic orientation of the recoil elements.

$\theta_0 + \theta$: new orientation of the recoil mechanism centerline.

The recoil elements dynamic deflection δ is related to the barrel assembly motion x through the geometrical relation (see Fig.3):

$$\delta = x \cos \theta_0 \quad (1)$$

The orientation change θ is related to the motion x through (see Fig.3):

$$\theta = \sin^{-1} \{x \sin \theta_0 / (L_0 - x)\} \quad (2)$$

where L_0 = initial length of the buffer elements (dampers & springs).

The dynamics of the barrel assembly-recoil mechanism depend on its differential equation. The differential equation depends on the characteristic nature of the buffer elements and its orientation angle. Fig.4 shows the free body diagram of the barrel assembly.

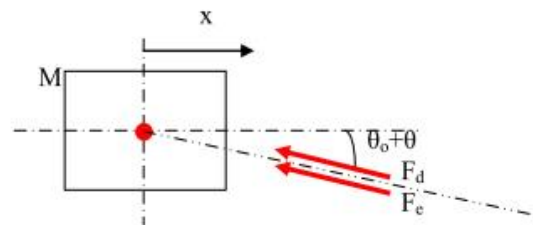


Fig.4 Free body diagram of the barrel assembly.

Using the free body diagram of Fig.4 and the second-law of motion, the differential equation of the barrel assembly of dynamic motion x is:

$$Mx'' = -F_d \cos(\theta_0 + \theta) - F_e \cos(\theta_0 + \theta) \quad (3)$$

Where: M = barrel assembly mass in kg,
 $x'' = d^2x/dt^2$ = barrel assembly acceleration (m/s^2)
 F_d = damping force of the hydraulic damper (N).
 F_e = elastic force of the recoil mechanism spring.

The damping force F_d of the hydraulic damper used in the present study depends on the damper velocity ($d\delta/dt$). The damping characteristics of a hydraulic damper is usually nonlinear and depends on the direction of motion of the damper piston. Since this study focuses on the effect of using air springs on the dynamics of the cannon-recoil mechanism, the damping coefficient of the recoil mechanism hydraulic damper is assumed constant at an average value in the forward or reverse strokes. Using the data of Polach and Halzman [21], the average damping coefficient of the hydraulic damper is:

And
$$\left. \begin{aligned} C_{av1} &= 6239 \text{ Ns/m} && \text{(positive velocity)} \\ C_{av2} &= 1405 \text{ Ns/m} && \text{(negative velocity)} \end{aligned} \right\} \quad (4)$$

The elastic characteristics of the air springs suggested for this engineering application depend on the air pressure inside the spring and the spring deflection. A typical air spring characteristics are shown in Fig.5 for spring displacement up to 100 mm and air pressure in the range 1.2 to 3.5 bar [13].

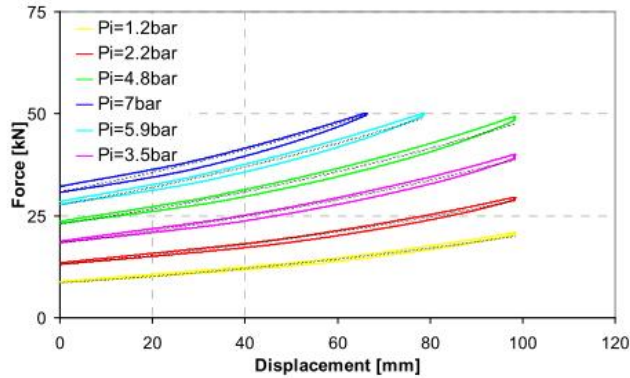


Fig.5 Air spring characteristics [13].

For the computer-aided analysis of the cannon barrel-recoil dynamics, the following mathematical models are fitted by the author to the data in Fig.5. The models are second-order polynomials with parameters depending on spring air pressure . The model takes the form:

$$F_e = a_1 \delta^2 + a_2 \delta + a_3 \quad (5)$$

The parameters and correlation coefficient of the model in Eq.4 are given in Table 1 as obtained by MATLAB using its command "polyfit" [22].

Table 1: Air spring parameters

Parameter	1.2 bar air pressure	4.8 bar air pressure	7 bar air pressure
a_1	0.000670163	0.0012237762	0.001904761905
a_2	0.0525291370	0.1367132860	0.144047619050
a_3	8.9825174825	23.42657342657	32.24999999
Correlation coefficient	0.9989	0.9980	0.9966

Now, the damping force acting on the barrel assembly is:

$$F_d = C_{av} \delta' \quad (6)$$

Combining Eqs.1 and 6 gives:

$$F_d = C_{av} (\cos\theta_o) x' \quad (7)$$

The elastic force now as function of the dynamic motion x is:

$$F_e = a_1 (\cos\theta_o)^2 x^2 + a_2 \cos\theta_o x + a_3 \quad (8)$$

Combining the Eqs. 3 through 8 gives the differential equation of the barrel-recoil system as:

$$Mx'' + F_d \cos(\theta_o + \theta) + F_e \cos(\theta_o + \theta) = 0 \quad (9)$$

3 BARREL RECOIL DYNAMICS

The dynamics of the barrel-recoil mechanism of the cannon are defined by solving Eq.9 which is extremely nonlinear. The procedure is as follows:

- Transfer the second-order homogeneous equation of Eq.9 to two first-order homogeneous equations.
- The mathematical tool for this is using state variables approach to build a new state model for the system.
- Use any numerical technique to solve the state model and get the dynamic system response for a specific initial conditions.
- MATLAB can be used to apply Runge-Kutta 4 technique to solve the state model using its command "ode45" [23].
- The dynamics can be evaluated for different recoil mechanism orientation θ_o since it has a great effect on the dynamic system characteristics.

The state model of the barrel assembly-recoil mechanism is derived by defining the state variables x_1 and x_2 as:

$$\left. \begin{aligned} x_1 &= x \\ x_2 &= dx/dt = x_1' \end{aligned} \right\} \quad (10)$$

Using all the above derived equations, the state model of the dynamic system is:

$$\text{And} \quad \left. \begin{aligned} x_1' &= x_2 \\ x_2' &= -(F_d/M) \cos(\theta_o+\theta) - (F_e/M) \cos(\theta_o+\theta) \end{aligned} \right\} \quad (11)$$

Where:

$$\begin{aligned} F_d &= C_{av} \cos\theta_o x_2 \\ F_e &= a_1(\cos\theta_o)^2 x_1^2 + a_2 \cos\theta_o x_1 + a_3 \\ \theta &= \sin^{-1} \{x_1 \sin\theta_o / (L_o - x_1)\} \end{aligned}$$

4 ANALYSIS RESULTS

A MATLAB code is written to apply the analysis and procedures suggested in this work to assign the dynamics of the cannon barrel assembly-recoil system upon firing. The other parameters of the dynamic system are:

- Barrel assembly mass, M: 5600 kg
- Projectile mass: 6.86 kg
- Muzzle velocity upon firing: 564 m/s
- Initial recoil elements length, L_o : 3.5 m.
- Initial horizontal velocity of the barrel assembly upon firing: 0.7 m/s based on momentum conservation of the barrel – projectile rigid bodies.

The results considering the recoil mechanism initial orientation and the spring internal pressure are as follows:

4.1 BAR SPRING AIR PRESSURE

- For zero and 10 degrees recoil mechanism orientation, Figs.6 and 7.

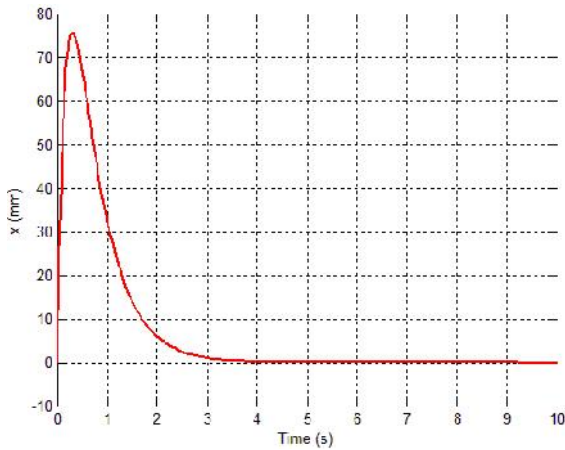


Fig.6 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 0$

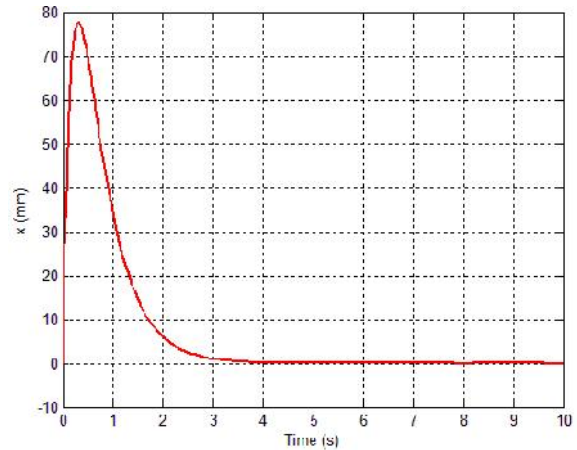


Fig.7 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 10^\circ$

- For 20 and 30 degrees recoil mechanism orientation: Figs.8 and 9.

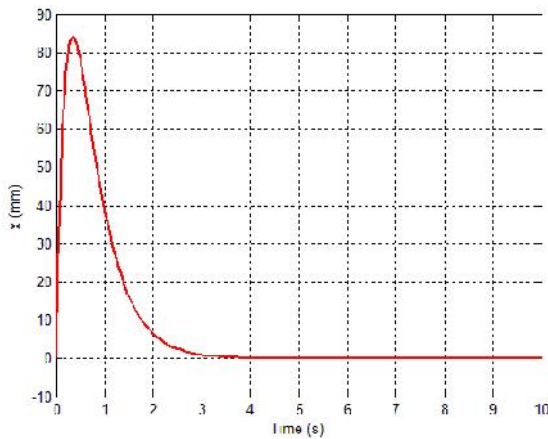


Fig.8 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 20^\circ$

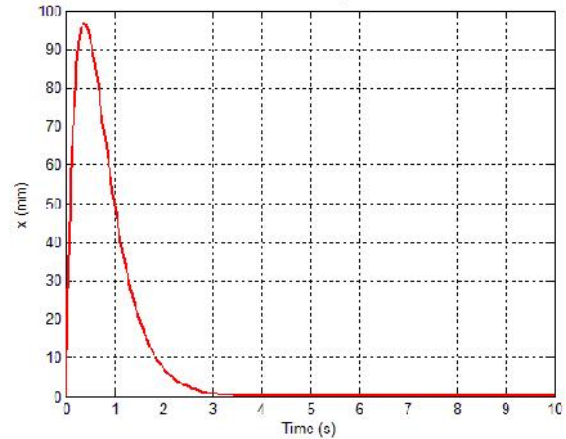


Fig.9 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 30^\circ$

- For 40 and 50 degrees recoil mechanism orientation: Figs.10 and 11.

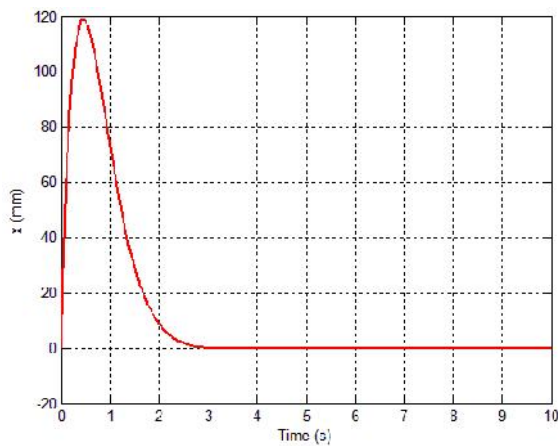


Fig.10 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 40^\circ$

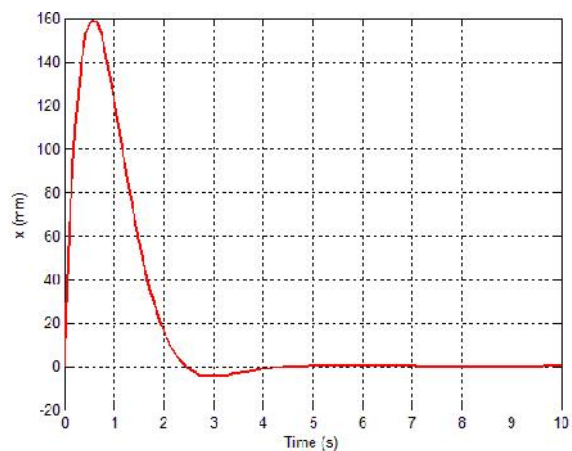


Fig.11 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 50^\circ$

4.2 BAR SPRING AIR PRESSURE

- For zero and 10 degrees recoil mechanism orientation , Figs.12 and 13.

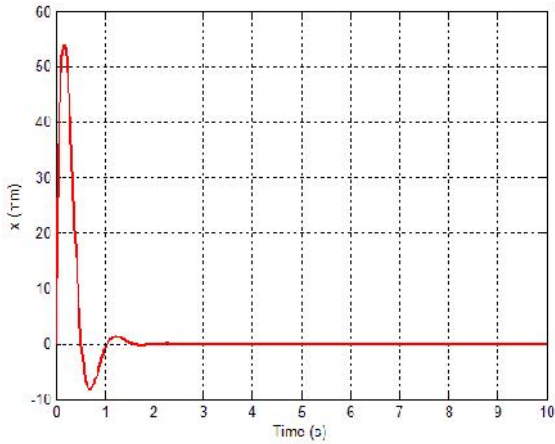


Fig.12 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 0^\circ$.

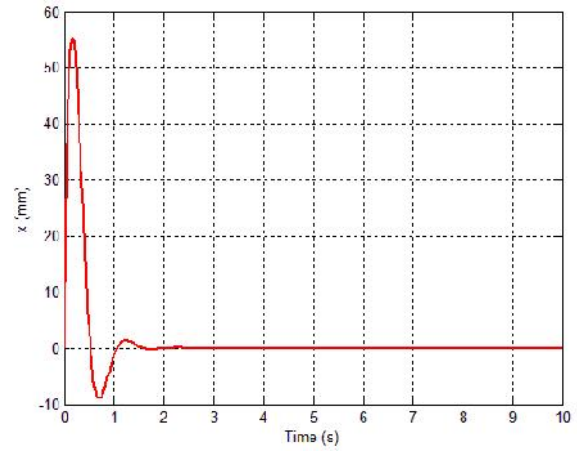


Fig.13 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 10^\circ$.

- For 20 and 30 degrees recoil mechanism orientation: Figs.14 and 15.

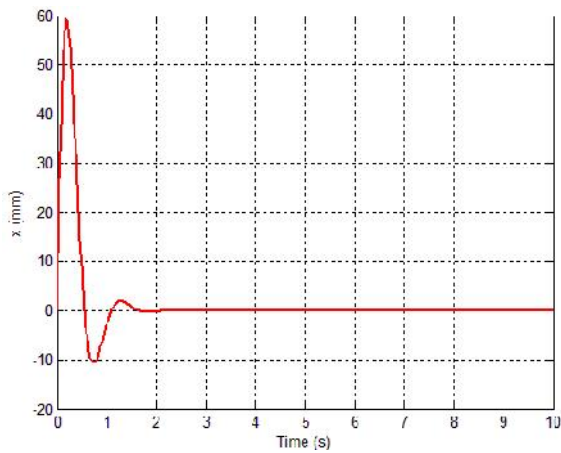


Fig.14 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 20^\circ$.

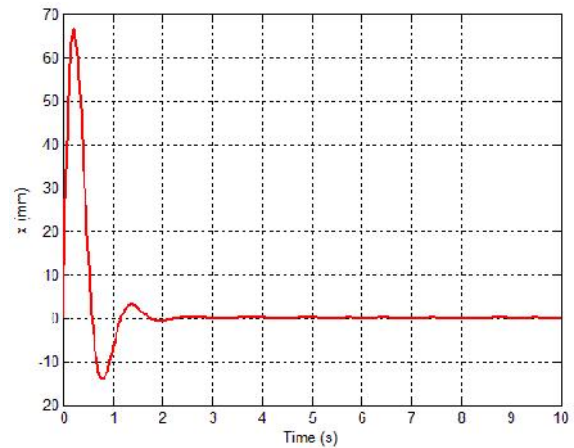


Fig.15 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 30^\circ$.

- For 40 and 50 degrees recoil mechanism orientation: Figs.16 and 17.

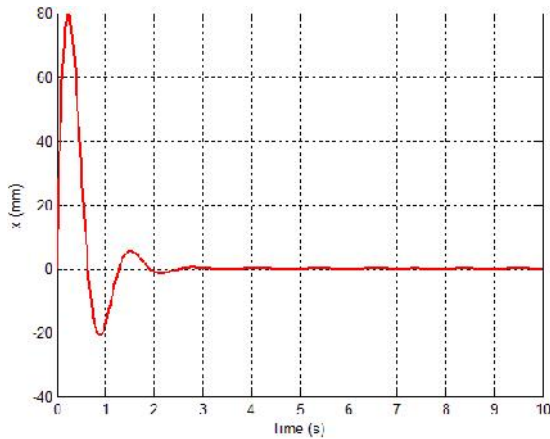


Fig.16 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 40^\circ$.

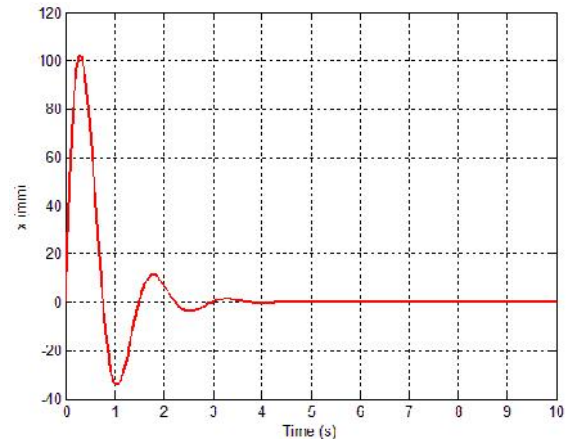


Fig.17 Dynamic response of the barrel assembly upon firing with $\vartheta_0 = 50^\circ$.

5 CHARACTERISTICS OF THE BARREL ASSEMBLY DYNAMIC MOTION

- For an 1.2 bar internal air spring pressure, the dynamic system behaves as a critically damped second-order one except at 50 degrees recoil mechanism orientation.
- For a 4.8 bar internal air spring pressure, the dynamic system behaves as an underdamped second-order one for all the recoil mechanism orientations.
- Upon firing, the dynamic motion of the barrel increases to a maximum value at the end of the return stroke of the barrel. This maximum value, x_{max} increases nonlinearly as the recoil elements orientation increases for the two air pressure levels of the air spring as shown in Fig.18.
- Because the dynamic response is of an oscillating nature, it has a minimum value x_{min} which is the maximum displacement in the forward direction after firing.
- x_{min} is function of the recoil elements orientation and the air spring initial pressure as shown in Fig.19.

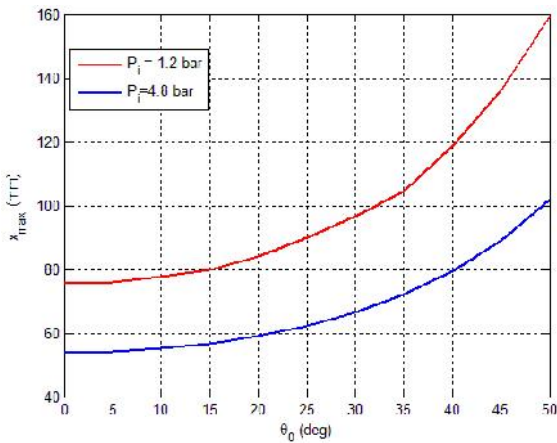


Fig.18. Maximum barrel assembly displacement.

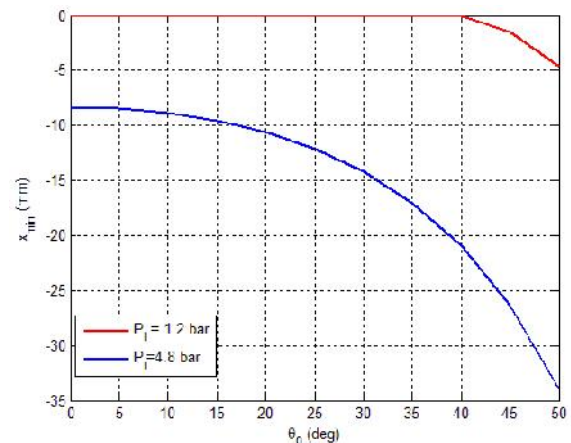


Fig.19 Minimum barrel assembly displacement.

- The minimum displacement of the barrel assembly remains at the zero level for recoil mechanism orientation < 40 degrees, then starts to decrease (increase in the negative direction) for the 1.2 bar internal air pressure of the air spring.
- For 4.8 bar air pressure, the minimum displacement x_{min} decreases as the orientation increases.
- The settling time of the barrel assembly remains constant at 4 seconds for 1.2 bar pressure and orientation ≤ 30 degrees (Fig.20). Then decreases to a minimum value at 40 degrees orientation, after which it increases to reach 4.5 degrees at 50 degrees orientation.

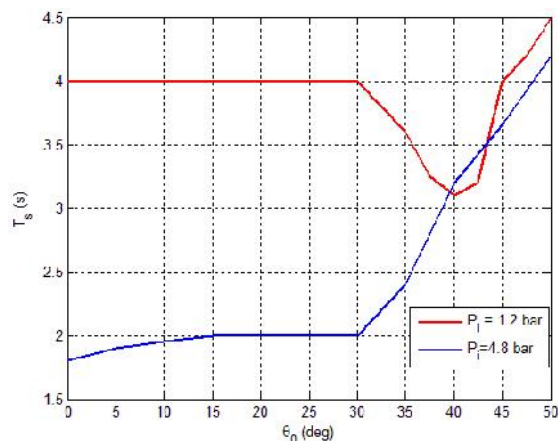


Fig.20 Settling time of the barrel assembly.

- At 4.8 bar air pressure, the settling time increases from 1.8 to 2 seconds gradually during 15 degree orientation, then remains fixed at 2 seconds for another 15 degrees orientation. Then increases in an increasing rate to reach 4.2 seconds at 50 degrees orientation.

6 EFFECT OF THE NUMBER OF AIR SPRINGS ON THE BARREL-RECOIL DYNAMICS

- The number of air springs encountered in the recoil mechanism affects the whole dynamics of the system.
- The air springs are connected to the barrel assembly in parallel.
- The effect of using 1, 2, 3 and 4 air springs for 1.2 bar air spring initial pressure and 30 degrees barrel assembly orientation is illustrated in Figs.21 through 24.

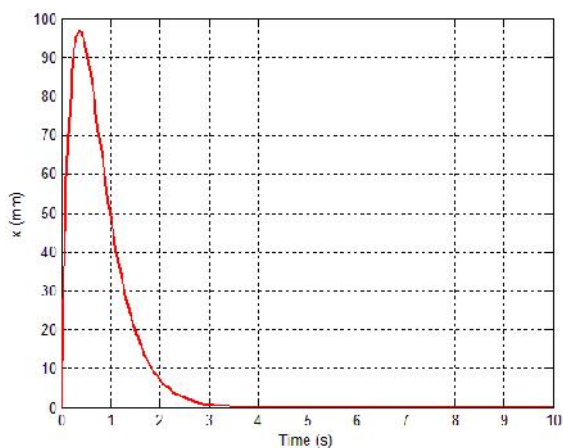


Fig.21 Barrel assembly dynamics with 1 air springs

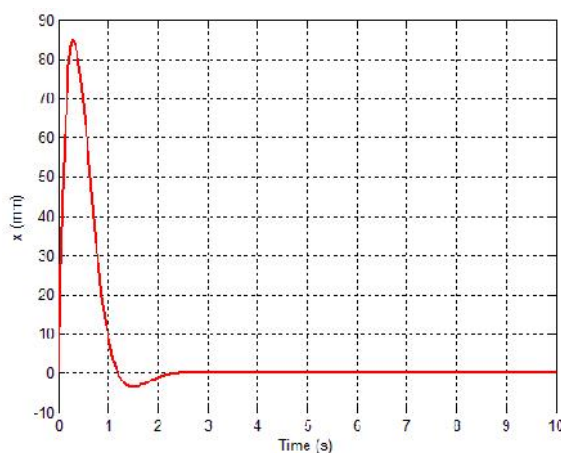


Fig.22 Barrel assembly dynamics with 2 air springs

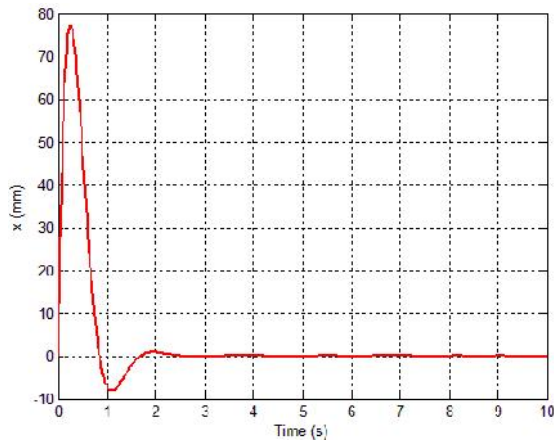


Fig.23 Barrel assembly dynamics with 3 air springs.

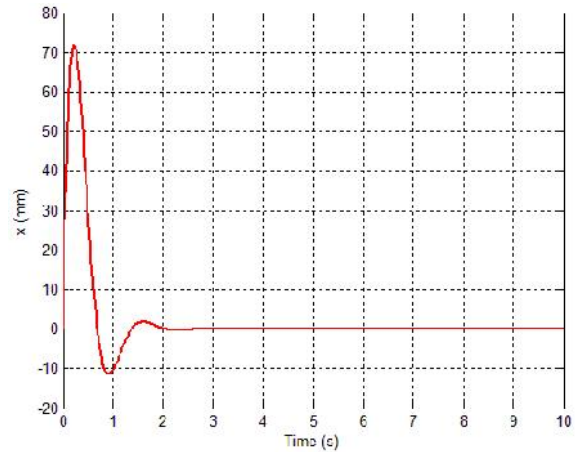


Fig.24 Barrel assembly dynamics with 4 air springs.

- The effect of the number of air springs on the parameters of the recoil mechanism performance is illustrated in Table 2.

Table 2 Effect of the number of air springs on the recoil mechanism performance.

Number of springs	x_{max} (mm)	x_{min} (mm)	T_s (s)
1	96.65	0	4
2	84.83	-3.46	2.5
3	77.23	-8.12	2.5
4	71.89	-11.44	2.1

7 CONCLUSIONS

- The characteristics of air springs depend on its displacement and internal pressure.
- Second order polynomial models are fitted to define the characteristics of the air spring used in this study.
- It was possible to obtain a barrel assembly-recoil mechanism characteristics similar to that of a critically damped second-order dynamic system at 1.2 bar spring initial pressure and recoil mechanism orientation ≤ 40 degrees.
- Increasing the recoil mechanism orientation increased the maximum barrel displacement upon firing and decreased the minimum displacement.
- The maximum barrel displacement increased nonlinearly with recoil mechanism orientation increase.
- The 1.2 bar spring initial pressure was optimum since it provided barrel assembly time response upon firing without any undershoot for recoil mechanism orientation ≤ 40 degrees.
- The settling time of the barrel assembly with 1.2 spring pressure was expected to be between 3.1 and 4.5 seconds for all the recoil orientation range.
- Increasing the spring initial pressure reduced the maximum barrel displacement, increased the minimum barrel displacement (undershoot) and decreased the settling time of the barrel assembly to as low as 1.8 seconds at zero orientation.
- The settling time remained constant at the level of 4 seconds for 1.2 bar spring pressure and recoil orientation ≤ 30 degrees.
- Increasing the number of air springs at 1.2 bar spring pressure and 30 degrees recoil orientation decreased the barrel maximum displacement and barrel settling time, while the minimum barrel displacement has increased (undershoot).

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