Dynamics of a Cannon Barrel-Recoil Mechanism with a Nonlinear Hydraulic Damper

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Abstract

Different types of dampers are available for use in the recoil mechanism of military cannons. The objective of this paper is to investigate the dynamics of the barrel assembly-recoil mechanism of military cannons when using a hydraulic damper and a constant stiffness helical spring in their recoil mechanisms. The damping characteristic of the hydraulic damper 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 analysis shows that the least minimum displacement, maximum displacement and minimum settling time of the barrel assembly are -30 mm, 226 mm and 7.5 seconds respectively.

Keywords: *Cannon recoil mechanism, Barrel assembly dynamics, Hydraulic damper, Nonlinear dynamic model, Barrel response upon firing, Recoil mechanism performance.*

1. INTRODUCTION:

Proper design of the cannon recoil mechanism is of vital importance to increase the performance of operation of the cannon and maintain high degree of safety of the operating soldiers. This is why 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].

Hydraulic dampers find wide application in a lot of engineering industries. They are used to cushion against dynamic loading to control resulting dynamic motions. Cheng-guo, Lixin, Jin-zhao and Wen-zhang (2003) obtained a compact empirical model for the hydraulic damper relating the damping force to its piston velocity. Their model depends mainly on the trigonometric functions sin and arctan [10]. Kurino, Matsunago, Yamada and Tagami (2004) presented an ingenious passive hydraulic damper for structural control with high performance equivalent to that of a semi-active damper. Their damper controls the damping coefficient by regulating the opening of a flow control valve housed in the damper without any outer power source [11]. Wei (2006) studied the characteristics of hydraulic dampers used in automotive suspension systems. The dampers characteristics are nonlinear and covered velocity range up to 1 m/s [12]. Guzzomi, O'Neill and Tavner (2007) studied the dynamics of a Tenneco automotive hydraulic damper to predict the damper performance [13]. Salem and Galal (2009) identified the characteristics and damping coefficient of a hydraulic shock absorber used in a light weight tracked vehicle under real conditions. They showed that the hydraulic damper characteristics are nonlinear in the velocity range -0.22 to 0.22 m/s with remarkable hysteresis in the negative range of damper velocity [14]. Stawik, Czop, Krol and Wszotek (2010) identified the root cause of the temporary decrease in the damping force occurring during the early stage of the stroking cycle's compression phase of hydraulic dampers. They presented the damper characteristics as a damping force against damper displacement in an extremely nonlinear fashion [15]. Hou et. al. (2011) established a detailed model of a shock absorber using the modelica language in the form of mathematical equations and object-oriented constructions [16]. Ferdek and Luczko (2012) created a physical and mathematical model for a twin-tube hydraulic damper incorporating numerical integration. They examined the effect of the amplitude and frequency range and the parameters describing the oil flow rate through the damper valves. They inestigated the damper characteristics for damper-piston velocity in the range -0.6 to 0.6 m/s [17]. Kate and Jadhav (2013) presented a mathematical model for the damping force of a hydraulic shock absorber. They showed that the damping characteristics of the hydraulic shock absorber are nonlinear for the damper-rod velocity range -3 to 3 m/s [18]. Sun, Jioo, Huang and Hua (2014) filled a hydraulic damper with $5x10^5$ cSt silicone oil as a non-Newtonian fluid with low velocity exponent. They presented the variation of the damper damping force with time for different impact drop height both experimentally and by simulation [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.

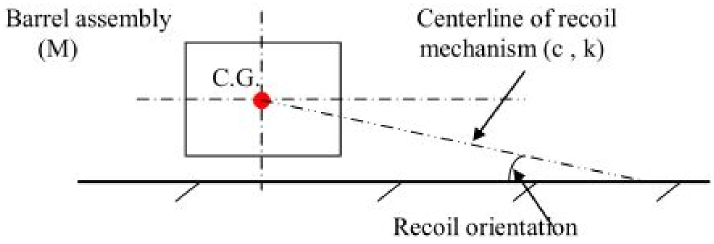

Fig.2 Cannon equivalent dynamic system.

The barrel assembly has a center of mass, G that is 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_o 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.

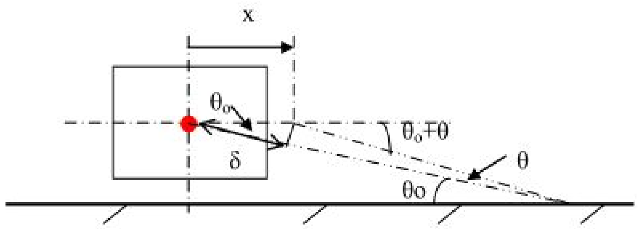

Fig.3 Dynamic system motion.

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.
$\theta_o+\theta$: 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):

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

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

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

where L_o = 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.

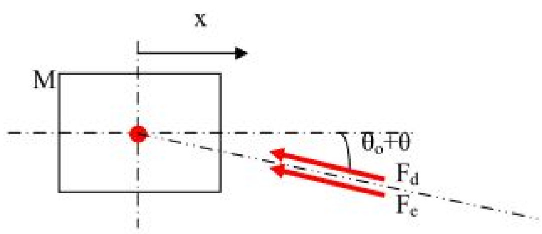

Fig.4 Free body diagram of the barrel assembly.

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:

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

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. Fig.5 shows a typical damping characteristics of a typical hydraulic damper [21].

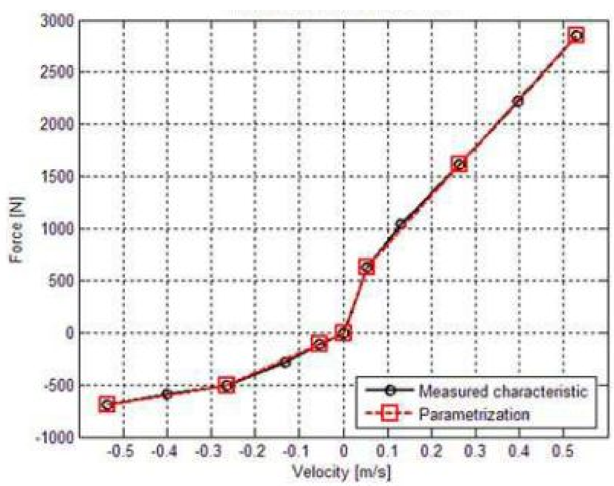

Fig.5 Typical damping characteristics of a hydraulic damper.

The damping characteristics of this damper depends on its velocity in magnitude and direction for a velocity range: -0.54 ≤ damper velocity ≤ 0.5 m/s. 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 third-order polynomials with parameters depending on the direction of the damper velocity . The model takes the form:

$$F_d = a_1 v^3 + a_2 v^2 + a_3 v + a_4 \quad (4)$$

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: Hydraulic damper damping parameters

Parameter	Positive velocity	Negative velocity
a_1	23615.02883	1267.72048024
a_2	-19968.573887	2963.66361350
a_3	9490.302535	2532.47600920
a_4	58.0478254	6.42175050
Correlation coefficient	0.99627	0.99754

The spring which is the elastic element of the buffer is assumed a helical spring. The selection of the spring stiffness is not straight forward. Its selection depends on:

- The damping characteristics of the hydraulic damper.
- The mass of the barrel assembly.
- The accepted characteristics of the barrel-recoil mechanism upon firing.

Following this procedure for a Howitzer M114 cannon of 5600 kg barrel assembly mass with a projectile having 6.86 kg mass and 564 m/s muzzle velocity, and for a hydraulic damper of the damping characteristics defined by Eq.4 and Table 1, the recommended equivalent stiffness of the recoil mechanism is:

$$k = 27500 \text{ N/m} \qquad (5)$$

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

$$F_d = a_1\, \delta'^3 + a_2\, \delta'^2 + a_3\, \delta' + a_4 \qquad (6)$$

Combining Eqs.1 and 6 gives:

$$F_d = a_1\, (\cos\theta_o)^3\, x'^3 + a_2\, (\cos\theta_o)^2\, x'^2 + a_3\, (\cos\theta_o)\, x' + a_4 \qquad (7)$$

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

$$F_e = kx \cos\theta_o \qquad (8)$$

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

$$Mx'' + [a_1\, (\cos\theta_o)^3\, x'^3 + a_2\, (\cos\theta_o)^2\, x'^2 + a_3\, (\cos\theta_o)\, x' + a_4] \cos(\theta_o+\theta) + kx \cos\theta_o \cos(\theta_o+\theta) = 0 \quad (9)$$

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.

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:

- 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 are as follows:

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

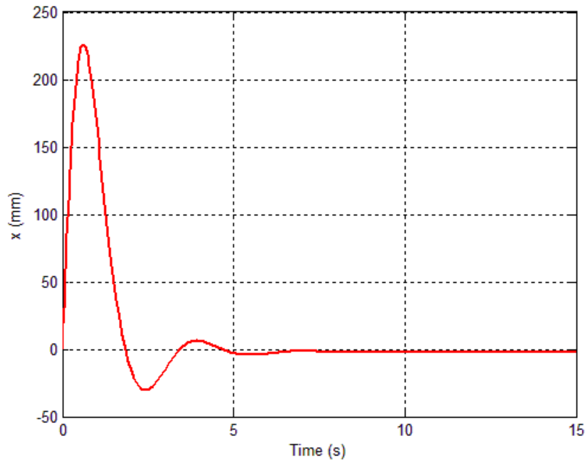

Fig.6 Dynamic response of the barrel assembly upon firing with $\theta_o = 0$.

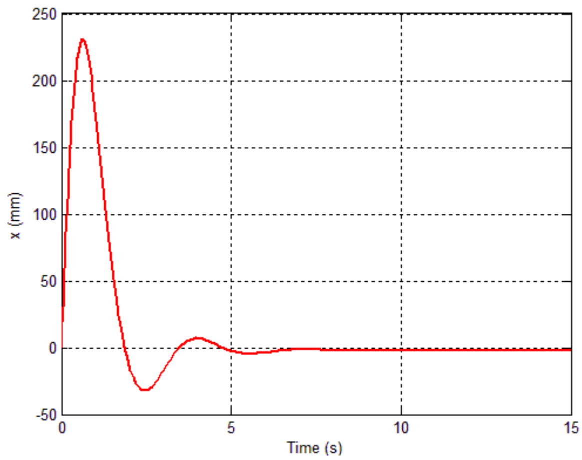

Fig.7 Dynamic response of the barrel assembly upon firing with $\theta_o = 10^o$.

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

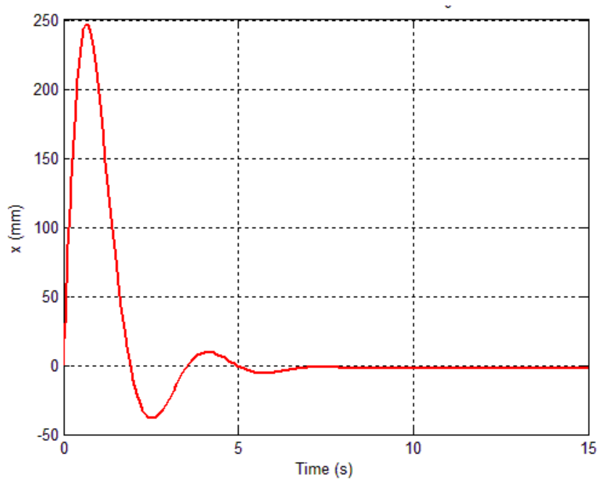

Fig.8 Dynamic response of the barrel assembly upon firing with $\theta_o = 20^o$.

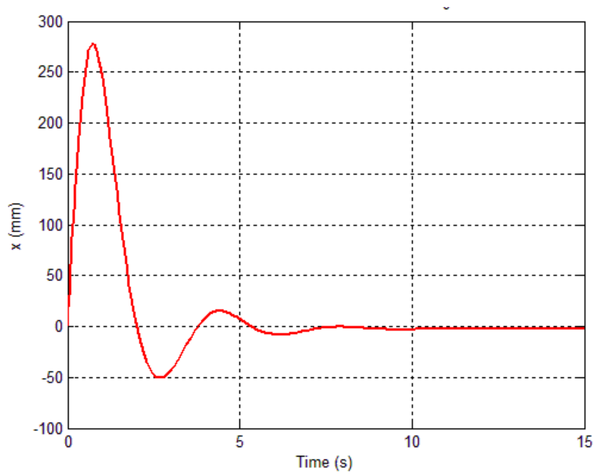

Fig.9 Dynamic response of the barrel assembly upon firing with $\theta_o = 30^o$.

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

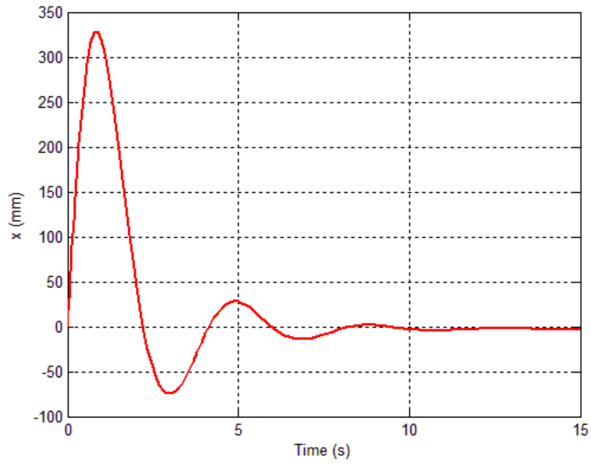

Fig.10 Dynamic response of the barrel assembly upon firing with $\theta_o = 40^o$.

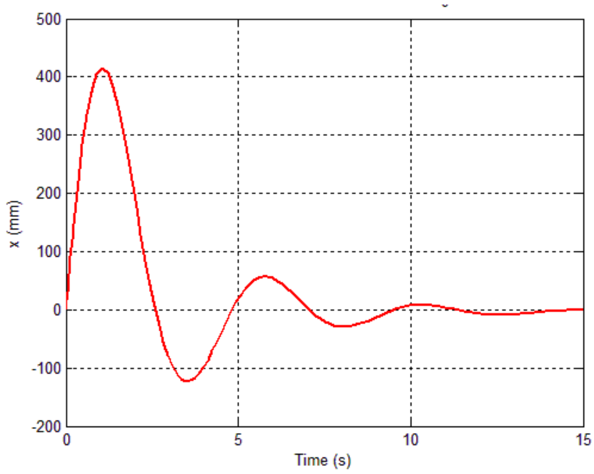

Fig.11 Dynamic response of the barrel assembly upon firing with $\theta_o = 50^o$.

5. CHARACTERISTICS OF THE BARREL-ASSEMBLY DYNAMIC MOTION:

- The system behaves as an underdamped second-order dynamic system.
- This is not an optimal situation because it is required to return the barrel to the original position before firing without any oscillation.
- This is possible only if the dynamic system is linear which is not the case here because of the nonlinear characteristics of the hydraulic damper and the orientation of the recoil elements.
- 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 as the recoil elements orientation increases as shown in Fig.12.
- Because the dynamic response is of an oscillating nature, it has a minimum value x_{min} which is the maximum motion in the forward direction after firing.
- x_{min} is function of the recoil elements orientation as shown in Fig.13.

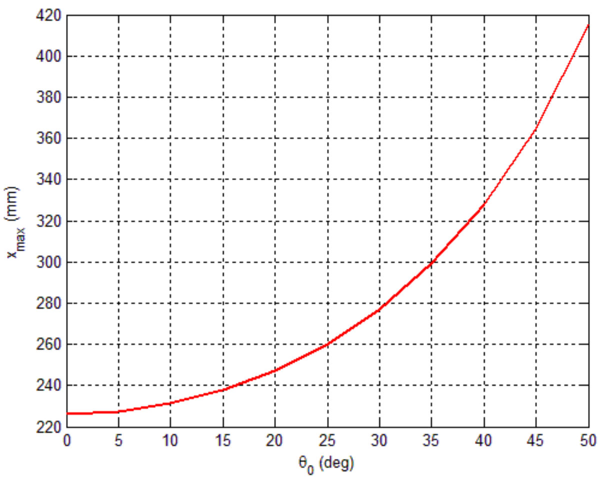

Fig.12. Maximum barrel assembly displacement.

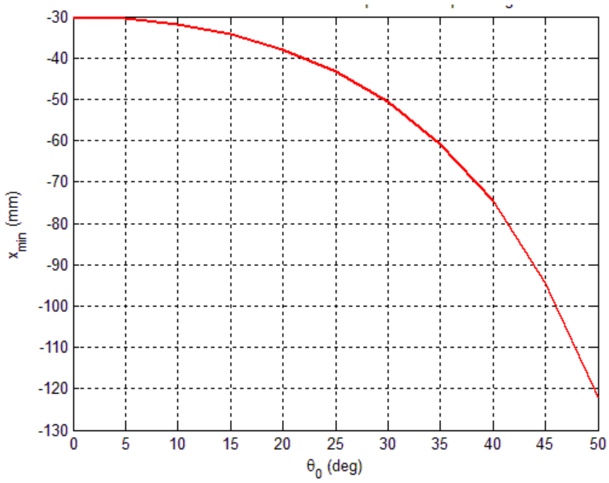

Fig.13 Minimum barrel assembly displacement.

- The variation of x_{max} is nonlinear having an exponential form.
- x_{min} decreases nonlinearly as the orientation increases.

- The third characteristic parameter is the response settling time. Functionally, it is required that the barrel settles at its original position before firing in a minimum time which is impossible because of the dynamics of the system.
- Again, the settling time depends on a number of parameters such as the recoil mechanism damping and stiffness, the barrel assembly mass, the recoil elements orientation and the initial velocity upon firing.
- Fig.14 shows the variation of the settling time with the orientation angle.

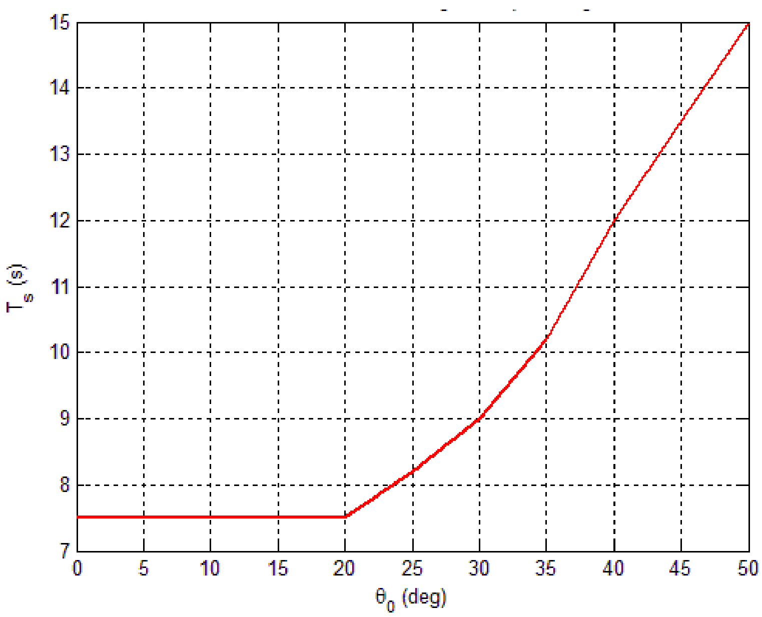

Fig.14. Settling time of the barrel assembly.

6. CONCLUSION:

- The dynamics of a cannon barrel assembly using recoil mechanism consisting of a hydraulic damper and a helical spring were studied.
- This dynamic problem was extremely nonlinear.
- The differential equation was derived in details in terms of the barrel assembly dynamic motion.
- The graphical nonlinear characteristics of the hydraulic damper were transformed to a third-order polynomial model for the forward and reverse motion of the damper piston rod.
- The dynamic model of the barrel assembly was an extremely nonlinear second-order differential equation.
- The dynamic motion of the barrel assembly was completely different than that of the linear dynamic system.
- It looked like that of an underdamped single degree of freedom system excited by an initial velocity.
- The dynamics of the barrel assembly were evaluated for different orientation of the recoil mechanism using Runge-Kutta 4 method through MATLAB.
- The maximum and minimum displacement of the barrel assembly upon firing changed significantly with the orientation of the recoil mechanism.
- The performance of the recoil mechanism was measured by the maximum displacement, minimum displacement and settling time of the barrel assembly upon firing.
- The optimum orientation of the recoil mechanism was zero (in the horizontal direction) where the maximum displacement, minimum displacement and settling time of the barrel assembly after firing were 226 mm, -30 mm and 7.5 seconds respectively.
- The minimum and maximum barrel assembly displacements increased exponentially with the recoil mechanism orientation.

- The settling time of the barrel assembly remained constant at 7.5 seconds for recoil mechanism orientation ≤ 20 degrees. Then increased in a semi-linear nature to reach 15 seconds at 50 degrees orientation.

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