DE GRUYTER Journal of the Mechanical Behavior of Materials 2022; 31: 177–185

Research Article

Nguyen Huu Phan, Santosh R. Patil*, Sanjaykumar Gawade, Shailesh S. Shirguppikar, Nguyen Trong Ly, Nguyen Chi Tam, and Le Thi Phuong Thanh

Design of ER damper for recoil length minimization: A case study on gun recoil system

https://doi.org/10.1515/jmbm-2022-0017
received September 22, 2021; accepted April 07, 2022

Abstract

Purpose – Recoil length minimization is always been a top concern in various structural, industrial, and defence applications. The traditional passive recoil system is not able to respond quickly to the changes in the impact force. These limitations can be overcome by introducing a semi-active recoil system, which comprises an intelligent fluid damper.

Design, methodology, and approach – The electrorheological (ER) fluid, which responds to the applied electric field and exhibits high yield strength, will be a proper damper for the recoil system. An ER damper can bring back the vibrating system to its equilibrium in a brief period. In this article, the general design procedure of an ER fluid semi-active damper has been developed, which is suitable for a recoil system. The equations of damping forces (viscous, quadratic, and ER) are modified in terms of geometric parameter, namely the piston diameter (D_P), which can be selected to obtain the desired dynamic range (greater than two) and optimum damping force. With the MATLAB software, the damper and spring specifications are fixed to meet the required conditions, viz. the dynamic range should be greater than two, and the total damping and spring force should counter-recoil the system.

Findings – Results obtained for a case study of gunfire conclude that the recoil system developed using design specifications exhibits desired performance with maximum recoil of 75.59 mm at 80°, angle of elevation, which is within the allowable range of 150 mm. It is shown that the rate of firing can be increased by decreasing the recoil length.

Originality and value – The novel procedure for the design of ER damper outlined in this work will be helpful for any recoil length minimization problem.

Keywords: design parameters, ER damper design, dynamic range, recoil system, total damping force, dynamic systems, active damping

1 Introduction

The selection of appropriate electrorheological (ER) fluid with desired yield strength and post-yield viscosity is a prerequisite of the design process of an ER damper. As the response time of ER fluids (in milliseconds) is greater than magnetorheological (MR) fluids, ER fluids have been selected for the application of gun recoil system. The barium titanyl oxalate (BTO)-based ER fluid is selected to exhibit a yield strength of 20 kPa at 3 kV/mm. The design of ER fluid damper within certain constraints of dynamic range and damping force is carried out. Also, calculations of recoil parameters, damper specifications, and the development of modified damper force formulae in terms of piston diameter for the general design process of ER damper have been carried out. The recoil mechanism considered here is of hydro-spring type in which a damper is incorporated with ER fluid with a recuperator as a spring. In the design of ER damper, the maximum impact force for the respective charge (firing a projectile) is taken as an input. The recoil force and recoil velocity are calculated and are used in the design process. Though the spring stiffness is calculated by using impact force equations, the stiffness of this system is adjusted to avoid vibration chattering. The damper rod size is fixed by using buckling theory, and by using empirical relation, damper diameter is fixed to meet criteria like dynamic range and total damping force. For this purpose, simple MATLAB codes are generated and run to fix these criteria'

*** Corresponding author: Santosh R. Patil,** Department of Mechanical Engineering, Rajarambapu Institute of Technology, Rajaramnagar, India, e-mail: santosh.patil@ritindia.edu
Nguyen Huu Phan, Nguyen Trong Ly, Nguyen Chi Tam, Le Thi Phuong Thanh: Faculty of Mechanical Engineering, Hanoi University of Industry, Hanoi, Vietnam
Sanjaykumar Gawade, Shailesh S. Shirguppikar: Department of Mechanical Engineering, Rajarambapu Institute of Technology, Rajaramnagar, India

piston diameter. Other parameters of the damper are determined using empirical relations. In many applications where recoil length minimization is paramount, this general design procedure of the recoil system will be helpful. In the present study, the recoil system designed has been applied as a case study to recoil minimization of battlefield gun.

Tian [1] derived a powerful yield strength equation, which can be used for selecting the ER fluid for industrial applications. Xianxiang *et al.* [2] proposed mechanisms of the ER effect. Laszlo *et al.* [3] investigated different properties of two ER fluids to check their suitability in gun recoil application. Tao [4], a pioneer in introducing the ER fluid with a yield stress of more than 100 kPa, studied the behaviour of physical phenomena of ER mechanism. Li *et al.* [5] synthesized giant electrorheological (GER) effect fluids. The fluid exhibits anti-sedimentation property with a yield strength of 194 kPa. Singh and Wereley [6] developed a control mechanism for the minimum transfer of gun recoil load to the structure and optimized the gunfire rate by introducing an MR damper in the semi-active control system. Wen *et al.* [7] carried out synthesis and characterization of GER fluid with the yield strength of 130 kPa at 5 kV/mm. Li and Wang [8] developed a recoil system with an MR absorber and have shown an improvement in the capacity of the absorber. Carlos and Lopez [9] have proved that semi-active suspension systems with ER damper can dissipate more energy. Xiangcong *et al.* [10] designed and analysed an intelligent gun recoil system with a damper. Mehdi *et al.* [11] studied the application of MR damper for controlling recoil dynamics. Zekeriya *et al.* [12] carried out a performance analysis of the flow mode MR damper having the annular gap in the piston. Kathe *et al.* [13] determined venting time, impulse reduction estimate, and bore heating reduction for 120 mm bore diameter gun. Dixon [14] developed a mathematical model for a damping system considering the quadratic damping coefficient. Shams *et al.* [15] have carried out the coupled computational: fluid dynamics (CFD) and finite element analysis (FEA) analysis of hydraulic shock absorber valve behaviour.

Because of the scope and depth of this research, there is no study in the literature on improving the performance by minimizing recoil length using an intelligent recoil mechanism. The major contribution of the present work is a novel design process of ER damper. The proposed design process can be helpful in recoil system applications. From the literature, it has been observed that there is no such systematic design process readily available. The main challenge is to design a recoil system within the constraints of maintaining a dynamic range of more than two and producing the required damping force. The better controllability over the structure, such as battlefield gun, has been achieved; it is necessary to develop an ER damping force more significantly than the force created by a viscous damper. The dynamic range is defined considering these forces, which is essential for better controllability. The MATLAB codes are generated and run to fix the piston diameter. The remarkable achievement of the proposed ER damper-based recoil system is that it showed percentage reduction in recoil mass displacement as 92% with respect to technical report by US ARDE and 43% compared to recoil system incorporated with MR damper at 63° of firing angle.

2 Design of an ER damper subjected to transient excitation

The general procedure of the design of ER damper subjected to external transient force excitation is presented in this section. The suitable ER damper has been designed by the iterative method, considering various parameters, which typically affect the performance of ER fluid damper. The dynamic range is the most crucial parameter concerned with the damper, which controls such a fluid damper. The dynamic range strongly depends on damper parameters like piston diameter, gap thickness, and length of recoil. During the designing of the damper, emphasis has been given to these parameters to maintain the required controllability and force. The design process is carried out by considering the maximum transient input to the recoil system. Design details are given in the following sub-sections.

2.1 Non-linear damping force equations to design ER damper

The equation of motion of a recoil system using a semi-active ER damper is given as:

$$m\ddot{x} + F_{\mathrm{D}} + kx = F_{\mathrm{F}}(t), \tag{1}$$

where F_{D} is the non-linear damping force provided by the ER damper and is given as:

$$F_{\mathrm{D}} = C_{\mathrm{D}}V_d + C_{\mathrm{Q}}V_d^2 + F_{\mathrm{ER}}. \tag{2}$$

The first term on the right-hand side of equation (2) represents viscous damping force, the second term represents quadratic damping force, and the third term is the ER damping force [14]. Here, C_D is the coefficient of viscous damping and C_Q is the coefficient of quadratic damping.

With this value of F_D, Eq. (1) becomes:

$$m\ddot{x} + C_D V_d + C_Q V_d^2 + F_{ER} + kx = F_F(t). \tag{3}$$

Eq. (3) is used when the firing angle, θ is zero. When θ is not equal to zero, the equation of motion (3) is modified to take into account the effect of the component of gravitational force in the direction of the motion. The modified Eq. (4) is as follows:

$$\begin{aligned} m\ddot{x} &+ (C_{D1}x + C_{D2}x^2 + F_{ER})\text{sign}\dot{x} + kx \\ &= F_F(t) + mg\text{Sin}\theta. \end{aligned} \tag{4}$$

2.1.1 Problem of the requirement of *a priori* knowledge of ER damper piston diameter

In the development of the recoil system, the design of an ER damper is the most important one, as shown in Figure 1. From a review of the literature on the design and development of an ER damper, it is seen that the main problem encountered is the requirement of *a priori* knowledge of the damper piston diameter. To overcome this problem, in this article, the formulae for viscous damping, quadratic damping, and ER damping are expressed in terms of piston diameter.

2.1.2 Modification of equations for F_{DV}, F_{DQ}, and F_{ER}

The annular flow design approach is employed while designing ER damper, as shown in Figure 1, and the controlling force is achieved by varying the electric field, as the orifices are not provided in the main piston. The ER phenomenon is achieved by maintaining an air gap less than 1 mm between two-cylinder electrodes. High DC voltage is applied across these electrodes.

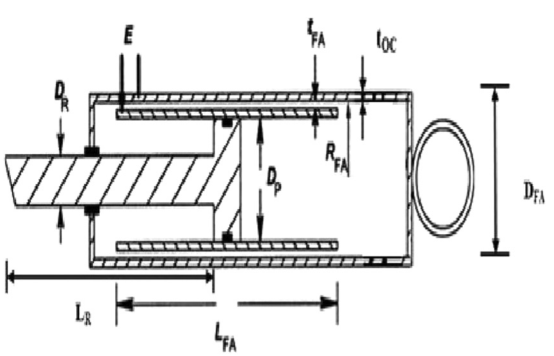

Figure 1: Cut section model of ER damper. E – applied voltage to the electrodes, L_{FA} – length of the fluid annulus, R_{FA} – radius of fluid annulus, D_{FA} – diameter of the fluid annulus, t_{FA} – thickness of fluid annulus, t_{OC} – thickness of outer cylinder, t_{IC} – thickness of inner cylinder, L_R – length of the piston rod, and the damping force expressions for F_{DV}, F_{DQ}, and F_{ER} are modified in terms of piston diameter (D_P).

The viscous damping force is given as follows:

$$F_{DV} = C_D V_d = \frac{6\,\mu L_{FA} A_{PA}^2}{\pi R_{FA} t_{FA}^3} V_d, \tag{5}$$

where μ is the viscosity of ER fluid and A_{PA} is the cross-sectional area of piston annulus.

$$A_{PA} = \frac{\pi}{4}(D_P^2 - D_R^2). \tag{6}$$

The cross-sectional area of the fluid annulus is obtained as:

$$A_{FA} = 2\pi R_{FA} t_{FA}. \tag{7}$$

Assuming geometric relations, diameter of rod, D_R = 0.6 D_P and the radius of fluid annulus, R_{FA} = 0.6 D_P.

Eq. (5) is modified by using Eqs. (6) and (7) with the aforementioned geometric relations as:

$$F_{DV} = \left(\frac{6 \times 0.409\,\pi}{16 \times 0.6}\right)\left(\frac{\mu\, L_{FA}\, V_d}{t_{FA}^3}\right) D_P^3. \tag{8}$$

Thus, Eq. (8) is deduced in terms of piston diameter. All other terms like the length of the fluid annulus, the viscosity of fluid, and gap maintained between two electrodes are fixed and assuming the maximum piston velocity as 2 m/s; the piston diameter can be adjusted to achieve the required damping force. Similarly, equation of quadratic damping force can be deduced in terms of piston diameter as:

$$F_{DQ} = C_D V_d^2 = \frac{1}{2}\rho\alpha f_A^2 A_{PA} V_d^2, \tag{9}$$

where ρ is the density of the fluid and α is the kinetic energy correction factor. f_A the area factor is given as:

$$f_A = \frac{A_{PA}}{A_{FA}} = \frac{\pi/4(0.64 \times D_P^2)}{2\pi R_{FA} t_{FA}} = 0.1333 \times \frac{D_P}{t_{FA}}. \tag{10}$$

Using Eqs. (9) and (10), equation of quadratic damping coefficient is simplified in terms of piston diameter as:

$$F_{DQ} = 0.0044\left(\frac{\rho\,\alpha\, V_d^2}{{t_{FA}}^2}\right) D_P^4. \tag{11}$$

Eq. (11) represents the quadratic damping force in terms of piston diameter.

The ER damping force equation is also deduced in terms of piston diameter. The ER force is subjected to the resulting ER effect, ultimately which depends upon the applied electric field between the gap of the electrodes. The ER effect produces yield shear stress (τ_{ER}) in ER fluid, which acts over the ER fluid trapped between the two electrodes. The ER shear force acting over the cross-sectional area of the fluid annulus is due to the pressure drop at the fluid annulus. Thus, resisting pressure acting over the cross-sectional area of piston annulus is given as:

$$F_{ER} = \frac{2\,L_{FA}\tau_{ER}}{t_{FA}}A_{PA}. \qquad (12)$$

Upon simplification F_{ER} is given as:

$$F_{ER} = 1.005\left(\frac{L_{FA}\tau_{ER}}{t_{FA}}\right)D_P^2. \qquad (13)$$

Eq. (13) represents the ER damping force in terms of piston diameter.

2.1.3 Development of a MATLAB problem to determine the piston diameter D_P

A MATLAB program is developed to calculate the total damping force (F_D), which comprises viscous, quadratic, and ER damping forces in terms of piston diameter (D_P). The diameter of the piston will be determined from a specific range of values with a fixed step size. The value of one such piston diameter from the range of value will be selected such that the total force obtained is approximately equal to the input firing force.

2.2 Testing suitability of damper diameter

While obtaining a piston diameter of ER damper, it is necessary to check the dynamic range ratio and total damping force for maximum impact force. The dynamic range ratio is the ratio of damping force obtained during the off-state condition to applied electric field condition. It can be given as [16]:

$$D_R = 1 + \frac{F_{ER}}{F_{VD} + F_{DQ}}. \qquad (14)$$

In order to get a significant effect of ER force, it is required to maintain a dynamic range more than or equal to 2. At the same time, the ratio of viscous damping force to ER damping force should be less than 1. Thus, the piston diameter can be fixed to fulfil both the conditions of dynamic range and maximum recoil force.

The radius of the fluid annulus is given as:

$$R_{FA} = 0.6 \times D_P.$$

Furthermore, the thicknesses of the inner and outer cylinders are determined by using pressure vessel theory. For the inner cylinder, inner diameter (D_i) is taken as the diameter of the piston. The dimensions of the piston rod are obtained by using the theory of buckling.

3 A case study on gun recoil system

The general design process of a recoil system discussed in Section 2 is implemented to design the recoil system of a battlefield gun. The ER damper and mechanical spring specifications are obtained based on the dynamics of the gun recoil system and the general design approach.

3.1 Gun recoil system and its dynamics

A schematic model of the typical gun recoil system is shown in Figure 2. When the gun fires a projectile, it leaves a barrel due to high pressure created behind it by the propellant. As per Newton's third law, the same pressure forces back the gun barrel. For heavy-duty guns, it is necessary to design recoil system with spring force "k" and damper with high damping coefficient "c."

3.2 Parametric analysis of gun recoil system

The peak force generated by a projectile of gun is of the order of 460 kN [17]. Considering this as maximum input force to the recoil system, the design process is carried out. Calculations have been performed on separate excel sheets using empirical equations. The results obtained are as follows: spring stiffness k = 1,850 kN/m, critical damping coefficient c_c = 153.280 kN/s, and the damping coefficient, c = 91.96 kN/s. Assuming maximum damper piston velocity as 2 m/s, the total damping force, F_d is obtained as 184 kN.

Total recoil force = Damping force + spring force
= 184 + 277.5 = 461.5 kN.

The total recoil force is nearly equal to the maximum firing force (460 kN).

3.3 Design calculations of ER damper

As described in Section 2.1.3, the diameter of the piston is determined from the range of values: 50–225 mm with a step size of 1 mm. The value of one such piston diameter is selected such that the total damping force obtained is approximately equal to input firing force and dynamic

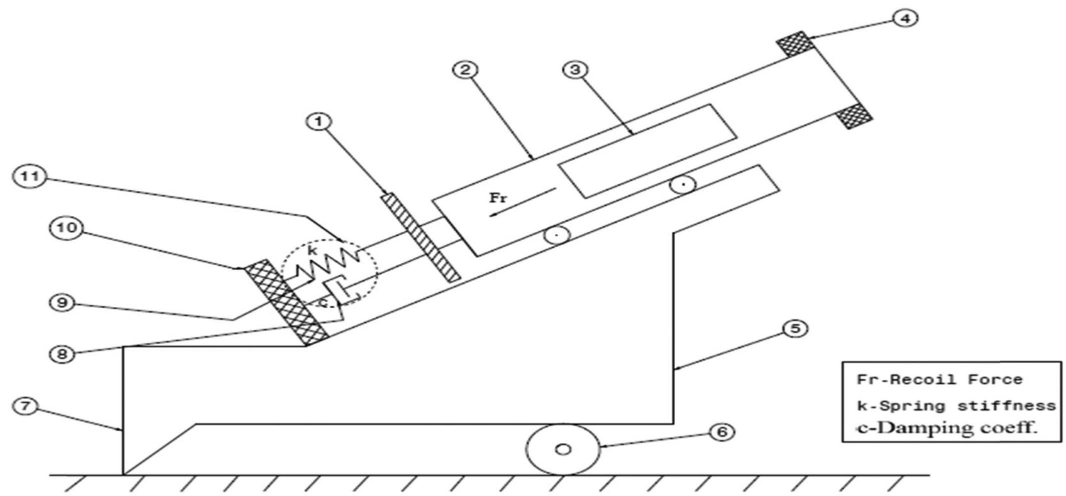

Figure 2: Schematic model of gun recoil system. 1 – Britch, 2 – barrel, 3 – caliber (projectile), 4 – muzzle brake, 5 – gun chasis, 6 – wheel, 7 – trail, 8 – ER damper, 9 – mechanical spring, 10 – gun carriage, and 11 – recoil system.

range is equal to 2. The dynamic range for the calculated diameter of the piston (140 mm) is 2, as shown in Figure 3. The total damping force for different values of piston diameter is shown in Figure 4.

Simultaneously,

i) Piston rod diameter is determined by using buckling theory.
ii) For calculated piston diameter, the dynamic range and damping force are checked.
iii) The buckling length for stroke length of 150 mm is determined as, $\text{Length}_{\text{buckling}} = 450$ mm.

Using this calculated buckling length and considering the factor of safety as 2, for the stroke of 150 mm the piston rod diameter is calculated as 82.25 mm. It will sustain maximum load of 460 kN without buckling.

By using geometric relation, $D_R = 0.6 \times D_P$; D_P is calculated as 137.08 mm.

Thus, the diameter of the piston is rounded off to the value of 140 mm. The dynamic range for 140 mm diameter of piston is nearly equal to 2 as shown in Figure 3 and the total damping force is 310 kN as shown in Figure 4.

The radius of fluid annulus is given as $R_{FA} = 0.6 \times D_P = 0.6 \times 140 = 84$ mm.

The inner diameter of inner cylinder is exactly equal to piston diameter, $D_{ii} = 140$ mm. The material selected for inner cylinder is mild steel.

The thickness of the cylinder can be determined as:

$$t = \frac{29.88 \times 140}{2 \times 140.16} = 14.92 \cong 15 \text{ mm}.$$

The outer diameter of inner cylinder, $D_{oi} = D_{ii} + 2t = 170$ mm.

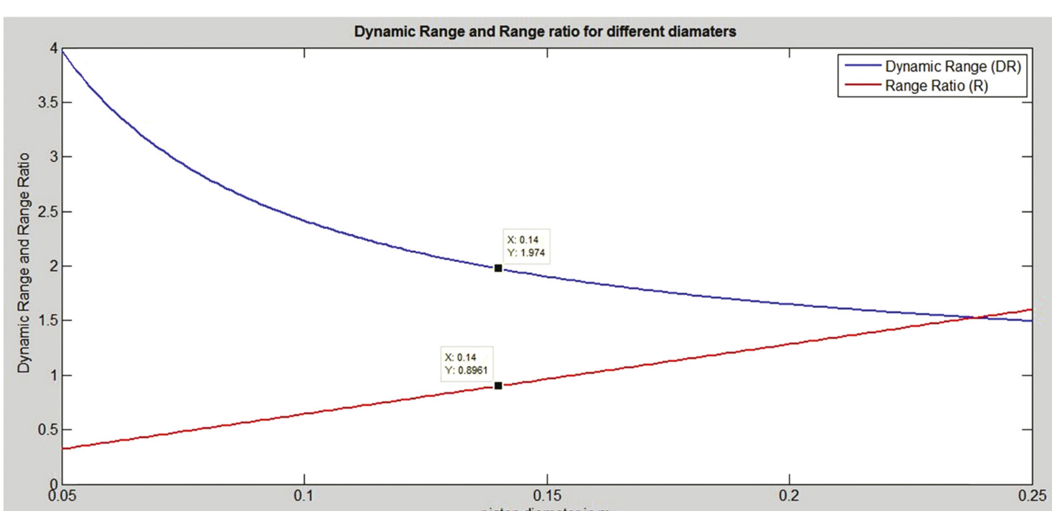

Figure 3: Dynamic range and range ratio vs different values of piston diameter (m).

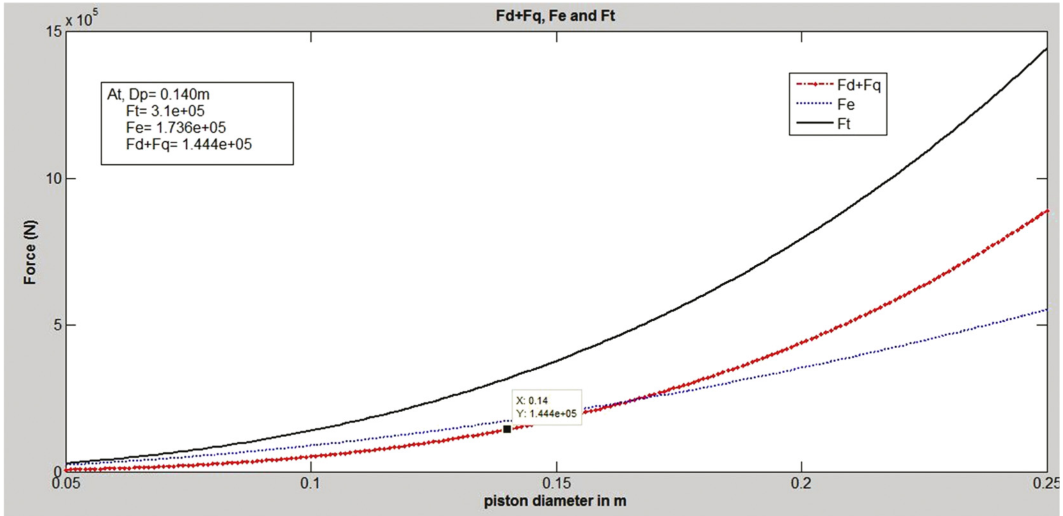

Figure 4: Total damping forces (F_t) vs different values of piston diameter (m).

Gap between two cylinders is already decided as 0.7 mm. The dimensions of outer cylinder are:

Inner diameter of outer cylinder (D_{io}) = 170 + 1.4 = 171.4 mm.

Outer diameter of outer cylinder (D_{oo}) = 171.4 + 2t = 201.4 mm.

All the specifications of the designed ER damper are summarized in Table 1.

The simulation based on designed ER damper is carried out for an application concerning gunfire.

4 The effect of angle of elevation on the gun recoil system performance: a simulation study

The 3D model of ER damper with designed specifications is developed. With this 3D model, performance evaluation of recoil system in MATLAB/Simulink environment has been carried out. The recoil displacement in terms of the desired value of stopping distance and total process time is recorded for different values of angles at which the gun is oriented.

For this purpose, Eq. (4) of recoil mass is rearranged as:

$$\ddot{x}(t) = \frac{1}{m}\{F_F(t) - kx + mg\mathrm{Sin}\theta - (C_{D1}x + C_{D2}x^2 + F_{ER})\mathrm{sign}(\dot{x})\}. \quad (15)$$

The Simulation Model is developed based on Eq. (15) using Simulink blocks. A Simulink code is developed using various inputs such as impact force, ER fluid characteristics, spring and damper specifications, and recoil mass. The resulting output is presented in graphical form, the velocity vs. displacement, displacement vs time, damping force vs time, and spring force vs time. The performance of the recoil system changes with firing elevation angle θ. The calculations of the projectile range and its height for different firing elevation angles θ (0–80°) have been carried out. The range for the projectile is maximum (32.315 km) at an elevation angle of 45°. Hence, the results are obtained for a firing elevation angle of 45°. The results of the analysis are given in

Table 1: Specifications of designed ER damper

Sr. no.	Design parameter	Notations	Dimension (mm)
1	Piston rod diameter	D_R	82.25
2	Piston diameter	D_P	140
3	Inner diameter of inner cylinder	D_{ii}	140
4	Outer diameter of inner cylinder	D_{oi}	170
5	Gap between cylinders/electrode gap	t_{FA}	0.7
6	Inner diameter of outer cylinder	D_{io}	171.4
7	Outer diameter of outer cylinder	D_{oo}	201.4

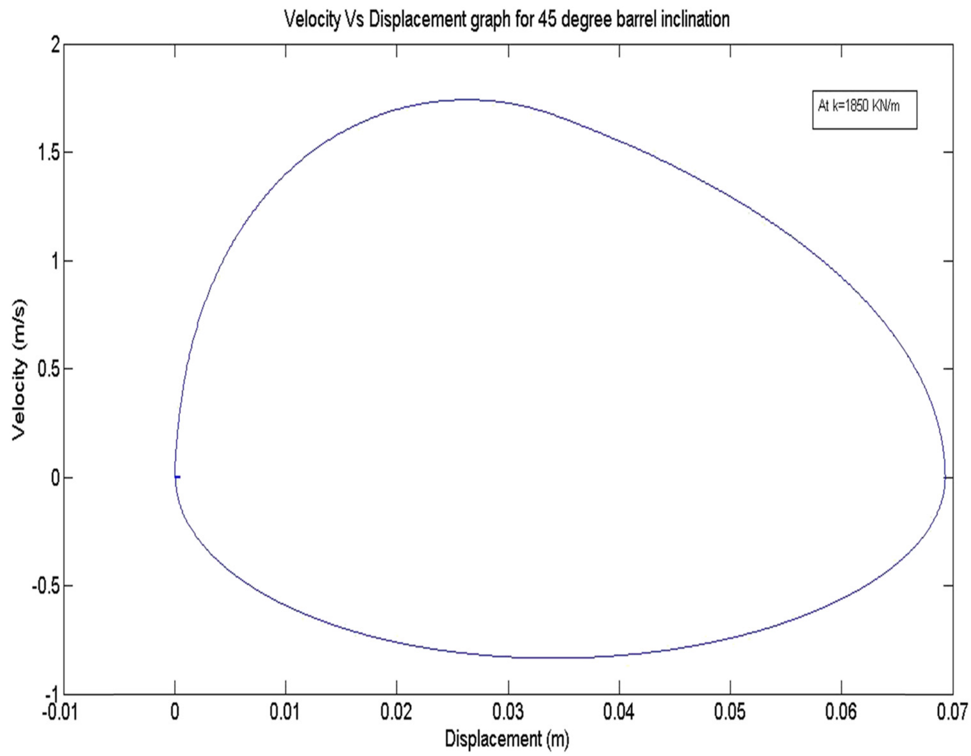

Figure 5: Velocity vs displacement at 45° firing angle.

Figures 5 and 6. Figure 5 presents the recoil mass that comes back to its' equilibrium position or at battery position. Figure 6 displays the maximum displacement covered by recoil mass during its motion and is 69.32 mm, which is less than the limiting recoil-stopping distance of 150 mm.

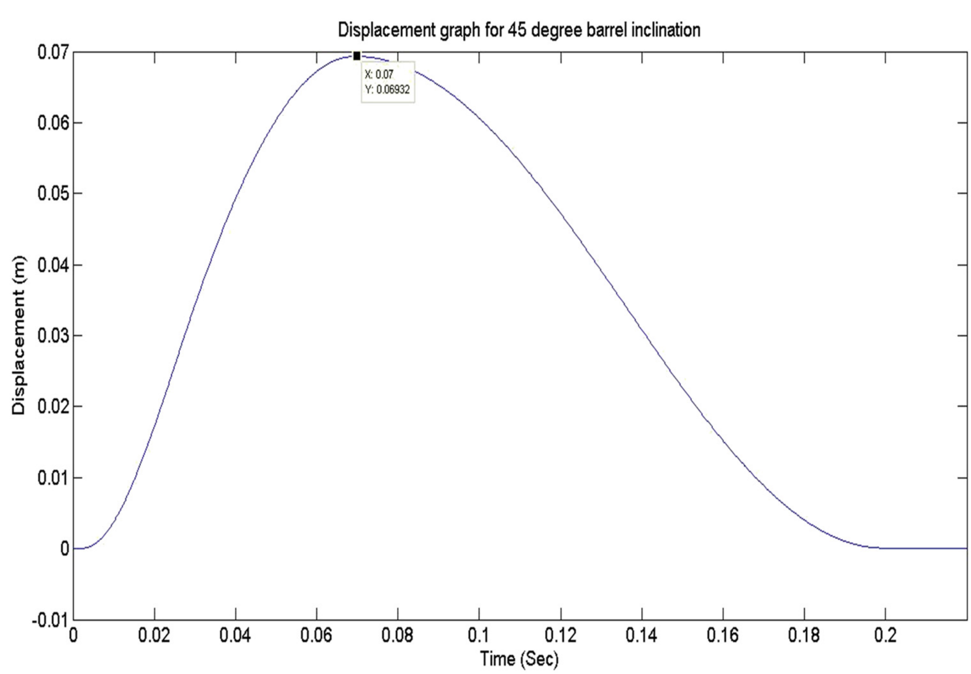

Figure 6: Displacement vs time at 45° firing angle.

Table 2: Displacement and yield strength at different firing angles

Firing angle in degrees	Yield strength of ER fluid (Pa)	Max. displacement by Simulink model (mm)
0	5,100	54.55
10	4,650	58.02
20	4,225	61.50
30	3,850	64.85
40	3,525	67.93
45	3,400	69.32
50	3,250	70.69
70	2,900	74.59
80	2,825	75.59

For the sake of completeness of the analysis, the values of maximum displacement at different firing elevation angles of 0°, 10°, 20°, 30°, 40°, 45°, 50°, 70°, and 80° have been obtained and are given in Table 2. Table 2 also gives yield strength of ER fluid required at respective firing angle to generate sufficient damping force so that the mass will recoil within the restricted distance of 150 mm and will come back to its equilibrium position.

It can be observed from Table 2 that the maximum displacement of recoil mass is achieved at 80°. The recoil mass displacement increases with the increase of angle of elevation. The maximum displacement even at an angle of 80° is less than the allowable displacement of 150 mm. Therefore, the specifications obtained for the developed ER damper are satisfactory to achieve the desired performance.

5 Comparative study

Based on this comparative study presented in Table 3, it is found that there is a large reduction in recoil displacement, hence the cycle duration will reduce and the number of fires per minute will increase significantly.

The proposed ER damper-based recoil system exhibited percentage reduction in recoil mass displacement as 95% with respect to technical report and 55% compared to recoil system incorporated with MR damper at 0° of firing angle and it is 92 and 43%, respectively, at 63° of firing angle.

Table 3: Comparison of recoil mass displacement

Firing inclination in degrees	Recoil mass displacement in (m)		
	Technical report	MR damper recoil system	Proposed ER damper recoil system
0	1.219	0.1225	0.05455
63	0.9525	0.1296	0.07342

6 Conclusion

In this research, we aimed to design the ER fluid damper for a recoil system. The equations of damping forces (viscous, quadratic, and ER) are modified in terms of geometric parameter, namely the piston diameter (D_P). The value of one such piston diameter is used such that the desired results like dynamic range greater than 2 and better controllability can be achieved. The design is also checked for piston rod buckling, and it is found safe. The great success of this study is to increase the firing rate by decreasing the recoil length, i.e. by restricting piston displacement of the innovative damper to 75 mm. The designed damper is developed for the semi-active gun recoil system to reduce the recoil length, cycle time, and firing rate. The experimental performance of the proposed recoil system reveals that the recoil displacement is varying from 40 to 60 mm with the recoil time of 0.25–0.35 s from 10° to 80° of firing inclination. Also, it showed significant reduction in recoil displacement as 92% compared to technical report of US ARDE and 43% to recoil system with MR damper at 63° firing angle. It is shown that the firing rate can be increased by decreasing the recoil length for this designed recoil system, and hence this is a viable alternative to counter the vibrations in the gun structure to improve the efficiency of the weapon.

The modified model is helpful to design ER damper for any application where recoil minimization is the prime objective.

Nomenclature

BTO	barium titanyl oxalate
D_R	piston rod diameter
D_P	piston diameter
D_{ii}	inner diameter of inner cylinder
D_{oi}	outer diameter of inner cylinder
D_{io}	inner diameter of outer cylinder
D_{oo}	outer diameter of outer cylinder
E	applied voltage to the electrodes
ER	electrorheological
F_{DV}	viscous damping force

F_{DQ} quadratic damping force
F_D total damping force
F_{ER} ER force
GER giant electrorheological
L_{FA} length of the fluid annulus
L_R length of the piston rod
R_{FA} radius of fluid annulus
t_{OC} thickness of outer cylinder
t_{IC} thickness of inner cylinder
t_{FA} gap between cylinders/electrode gap
τ_{ER} yield shear stress

Acknowledgment: The authors would like to express their deep gratitude to Prof. S.G. Joshi Walchand, College of Engineering, Sangli, for his contributions and valuable critiques of this study.

Funding information: Authors state no funding involved.

Author contributions: All authors have accepted responsibility for the entire content of this manuscript and approved its submission.

Conflict of interest: Authors state no conflict of interest.

References

[1] Tian H. Electro-rheological suspensions. Adv Colloid Interface Sci. 2002;97(1–3):1–35.
[2] Xianxiang H, Weijia W, Shihe Y, Ping S. Mechanisms of the giant electro-rheological effect. Solid State Commun. 2006;139:581–8.
[3] Laszlo JK. Assessment of two electro-rheological fluids for use in recoil abatement applications. Army Research Laboratory; 2001. p. 1–53. ARL-TR-2483.
[4] Tao R. Super strong magneto-rheological fluids. J Phys Condens Matter. 2001;13:979–99.
[5] Li J, Xiuqing G, Shuyu C, Weijia W, Ping S. Giant electro-rheological fluid comprising nanoparticles: carbon nanotube composite. J Appl Phys. 2010;107:1–5.
[6] Singh HJ, Wereley NM. Optimal control of gun recoil using magnetorheological dampers. J Mech Behav Active Mater. 2012;2:441–50.
[7] Weijia W, Xianxiang H, Shihe Y, Kunquan L, Ping S. The giant electro-rheological effect in suspensions of nanoparticles. Nat Mater. 2003;2:727–30.
[8] Li ZC, Wang J. A gun recoil system employing a magneto-rheological fluid damper. J Smart Mater Struct. 2012;21:1–10.
[9] Carlos A, Lopez V. Modeling method for electro-rheological dampers. Cape Town: International Federation of Automatic Control; 2014. p. 24–9.
[10] Xiangcong Z, Xingjian J, Li C. Magneto-rheological fluid dampers: a review on structure design and analysis. J Intell Mater Syst Struct. 2012;23(8):839–73.
[11] Mehdi A, Randall JA, Norris JA. Designing magneto-rheological dampers in a fire out-of-battery recoil system. IEEE Trans Magn. 2003;39(1):480–5.
[12] Zekeriya P. Time-dependent CFD and quasi-static analysis of magnetorheological fluid dampers with experimental validation. Int J Mech Sci. 2012;64:22–31.
[13] Kathe E, Dillon R, Sopok S, Witherell M, Dunn S, Coats D. Rarefaction wave gun propulsion. US Army Armament Research, Development and Engineering; 2001. p. 1–23. ARCCB-TR-01008.
[14] Dixon JC. The shock absorber handbook. Co-publication between Professional Engineering Publishing Ltd and John Wiley and Sons. Ltd; 2007. p. 314–8.
[15] Shams M, Ebrahimi R, Raoufi A, Jafari BJ. CFD-FEA analysis of hydraulic shock absorber valve behavior. Int J Automot Technol. 2007;8(5):615–22.
[16] Akiwate D, Gawade SS. Design and performance analysis of smart fluid damper for gun recoil system. J Adv Mech Eng. 2014;4(5):543–50.
[17] Zepp W, Cirincione R. Modular Artillery charge system (MACS) compatibility with the 155-mm M114 towed howitzer. Technical report by Armament Engineering & Technology Centre; 2004. p. 1–16. ARAEW-TR-04009.