International Journal of Engineering

Journal Homepage: www.ije.ir

Performance Evaluation of Magnetorheological Damper Valve Configurations Using Finite Element Method

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PAPER INFO

Paper history:
Received 16 November 2016
Received in revised form 29 November 2016
Accepted 05 January 2017

Keywords:
Magneto-rheological (MR) Damper
Damping Force
Dynamic Range
Valve Ratio
Inductive Time Constant
Pressure Drop

ABSTRACT

The main purpose of this paper is to study various configurations of a magnetorheological (MR) damper valve and to evaluate their performance indices typically dynamic range, valve ratio, inductive time constant and pressure drop. It is known that these performance indices (PI) of the damper depend upon the magnetic circuit design of the valve. Hence, nine valve configurations are considered for which mathematical models are developed. A finite element model is built to analyze and investigate the PI of a 2-D axi-symmetric MR damper valve. All configurations of the damper valve are simulated within a given range of input current and number of turns of coil, and within this range, damping force, dynamic range, valve ratio, inductive time constant and achieved pressure drop have been evaluated. The simulation results show that the PI of the MR damper are highly dependent on the shapes of valves and hence the valve shape should be selected based on the intended application. The results obtained in this work provide an insight for designers to create application-specific MR dampers.

doi: *10.5829/idosi.ije.2017.30.02b.18*

NOMENCLATURE

H_{MR}	Magnetic field intensity of MR fluid along the pole length	m_{cr}	Mass of core
B_{MR}	Magnetic flux density of MR fluid along the pole length	m_h	Mass of inner house
Q	Volumetric rate of flow	m_{co}	Mass of coil
c	Coefficient of the flow velocity profile	m_{MR}	Mass of MR fluid
I	Electric current applied to the valve	G	Complex shear modulus
P_o	Initial pressure of the gas chamber	**Greek Symbols**	
V_o	Initial volume of the gas chamber	τ	Fluid shear stress
$\dot{x}_p$	Velocity of the piston	τ_y	Yield stress developed as result of applied field
x_p	Displacement of the piston	$\dot{\gamma}$	Fluid shear strain rate
A_p	Piston effective cross sectional area	γ	Fluid shear strain
A_s	Piston shaft effective cross sectional area	ΔP_τ	Applied field dependent pressure drop
A_w	Cross sectional area of the coil wire	ΔP_η	Applied field independent pressure drop
m_v	Total mass of vlave	η	Plastic viscosity of MR fluid without applied magnetic field

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Please cite this article as: S. Seid, S. Sujatha, S. Chandramohan, Performance Evaluation of Magnetorheological Damper Valve Configurations Using Finite Element Method, International Journal of Engineering (IJE), TRANSACTIONSB: Applications Vol. 30, No. 2, (February 2017) 303-310

1. INTRODUCTION

A magnetorheological (MR) damper is a device used in a semi-active control system to mitigate unwanted vibration. The damping performance indices in this device can be controlled by changing the viscosity of the MR fluid used. MR fluid is a suspension of micrometer-sized magnetic particles in a carrier fluid, which is usually a type of oil. In the absence of an applied field, the particles are distributed randomly and the fluid exhibits quasi-newtonian behavior. When the MR fluid is subjected to a magnetic field, the particles become magnetized and start to behave like tiny magnets. The interaction between the resulting induced dipoles causes the particles to aggregate and form fibrous structures within the carrier liquid, changing the rheology of the MR fluid to a near solid state. These chain-like structures restrict the flow of the MR fluid, thereby increasing the viscous characteristics of the suspension. The mechanical energy needed to yield these chain-like structures increases nonlinearly with an increase of the applied magnetic field, resulting in a field-dependent yield stress. The process is fully variable and reversible. By controlling the strength of the magnetic field, the shear strength of the MR fluid can be altered, so that resistance to the MR flow can be varied.

In the last decade many researchers have carried out studies on semi-active control systems; a large number of academic publications have been presented. Researchers have shown that designs that make use of MR fluids and devices are potentially simpler, more reliable and consistent than conventional electromechanical devices. Chung et al. [1] compared the damping performances of MR dampers operating in flow and shear modes. Li et al. [2] designed an MR valve and studied the effects of magnetic field formulation mechanism on MR valve performance. Hu et al. [3] proposed a double-coiled MR damper valve where damping force and dynamic range were used in investigating the performance of the damper. Olabi and Grunwald [4] compared the performances of valve, shear and squeeze modes of operations of MR damper valves in terms of fast responses, simple interface with electric input and mechanical output and controllability. Nguyen et al. [5] developed an MR valve constrained in a specific volume considering valve ratio, pressure drop due to yield stress and power consumption. Nguyen et al. [6] designed and compared single-coiled and double-coiled annular MR valve structures based on the achieved pressure drop due to yield stress. Nguyen and Choi [7] designed a vehicle MR damper considering damping force and dynamic range. Gudmundsson et al. [8] developed an MR rotary valve for prosthetic knee application considering weight and torque produced. Parlak et al. [9] established an optimization method that was carried out for the objectives of target damper force and maximum magnetic flux density of an MR damper. Djavareshkian et al. [10] designed an MR damper where fluid gap, number of magnetic wire turns and active length of the MR damper valves were considered as design variables to optimize damper performances, particularly electrical power consumption, inductive time constant and damping force. Amiri et al. [11] studied the effect of the size of fluid-conveying magneto-electro-elastic (MEE) tubular nano-beam on its vibrational and instability behavior when it was subjected to magneto-electric potential and thermal field.

From the available literature, it is apparent that the study of MR damper valve configurations and their impact on the performance indices (PI) of the damper have been unexplored. In this study, the authors have considered nine configurations of an MR damper valve to study their impact on the performance indices of the damper, particularly on dynamic range, valve ratio, inductive time constant and pressure drop. From this study, one can improve particular aspects of the PI of the damper for intended applications. Therefore, the main objective of this paper is to evaluate the performance indices of different configurations of an MR damper valve using finite element analysis in ANSYS software environment.

2. MATHEMATICAL MODELS

The evaluation of the PI is done based on the quasi-static model of MR valves where the equations are derived based on the assumption that the MR fluid exhibits Bingham plastic behavior and the flow is fully developed in the ducts [5, 12, 13]. Schematic representation of single-coil valve with annular duct is shown in Figure 1 and the Bingham's plastic flow model is given by the following equation [13, 14]:

$$\tau=\eta\dot{\gamma}+\tau_y\left(H_{MR}\right)sgn\left(\dot{\gamma}\right), \left|\tau\right| \geq \tau_y \quad \tau = G\gamma, \left|\tau\right| < \tau_y \tag{1}$$

Equation (1) is used to design a device which works on the basis of MR fluid. The total pressure drop in the damper is evaluated by summing the viscous component and yield stress component which is approximated as [2, 5, 6, 14]:

$$\Delta P = \Delta P_\eta + \Delta P_\tau = 6\eta HQ / (\pi g^3 R_1) + 2cL_p \tau_y / g \tag{2}$$

in which ΔP is the pressure drop of the MR fluid flow through the orifice gap of the valve. The parameter c is a coefficient which depends on the flow velocity profile, and it has a value varying from 2.07 to 3.07 [7, 14]. The coefficient c can be approximately estimated as follows [6, 7, 14]:

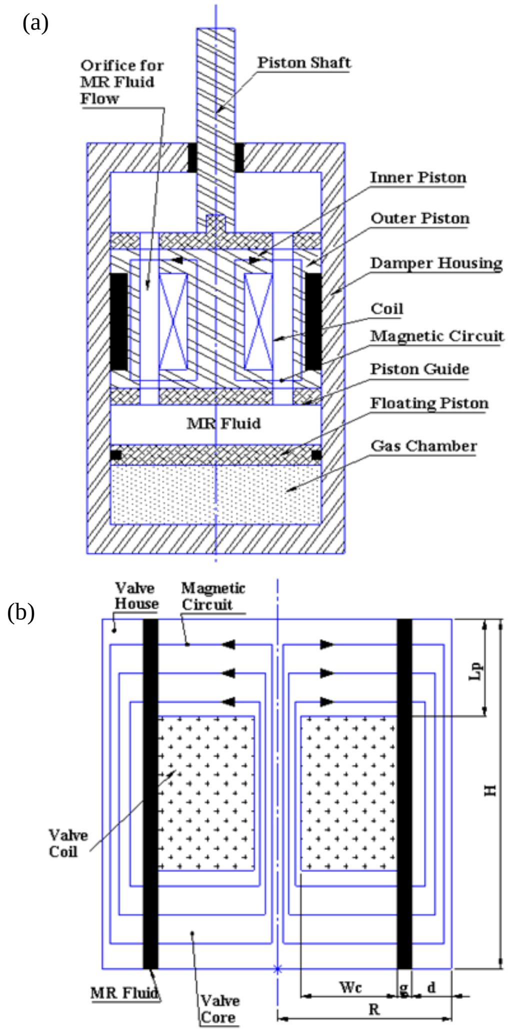

Figure 1. Schematic configuration of a single-coil MR valve and damper: (a) MR damper and (b) MR valve

$$c = 2.07 + \frac{12Q\eta}{12Q\eta + 0.8\pi R_1 d^2 \tau_y} \tag{3}$$

R_1 is the average radius of the duct given as a distance between the axis of the valve to the centroid of the duct. The duct has various shapes depending on the type of configuration of the piston valve (Figure 2). Therefore, the centroid of each duct has been computed and R_1 of the corresponding duct has been adopted in the computation. For the rectangular annular duct, R_1 is given by:

$$R_1 = R - d - 0.5g \tag{4}$$

Through curve fitting of the characteristic curves of the MR fluid (MRF132-DG) from Lord Corporation, the induced yield stress of the MR fluid as a function of the applied magnetic field intensity (H_{MR}) can be approximately expressed as [15]:

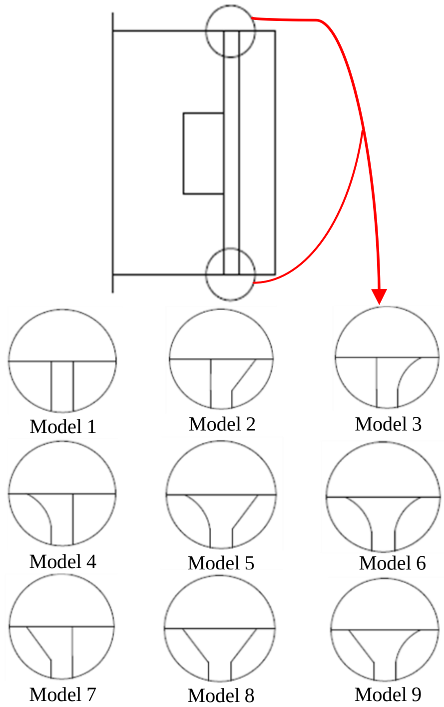

Figure 2. 2D axi-symmetric model and magnified views of the ends of the MR valve configurations

$$\tau_y = C_0 + C_1 H_{MR} + C_2 H_{MR}^2 + C_3 H_{MR}^3 \tag{5}$$

In Equation (5), the unit of the yield stress is in kPa, while that of the magnetic field intensity is in kA/m.The coefficients C_0, C_1, C_2, and C_3, determined from experimental results by applying the least square curve fitting method, are respectively identified as 0.3, 0.42, -0.00116 and 1.05×10^{-6}.

Damping force is given by [6, 7]:

$$F_d = P_a A_s + C_{vis}\dot{x}_p + F_{MR}\,\mathrm{sgn}\left(\dot{x}_p\right) \tag{6}$$

where:

$$C_{vis} = \frac{6\eta H}{\pi R_1 g^3}\left(A_p - A_s\right)^2,$$

$$F_{MR} = \left(A_p - A_s\right)\frac{2cL_p}{g}\tau_y \text{ and}$$

$$P_a = P_o\left(\frac{Vo}{Vo + A_s x_p}\right)^k,$$

The parameter k represents the coefficient of thermal expansion which varies between 1.4 and 1.7 for adiabetic expansion [7].

Dynamic range is given by [6]:

$$\lambda_d = \frac{P_a A_s + C_{vis}\dot{x}_p + F_{MR}\,\mathrm{sgn}\left(\dot{x}_p\right)}{P_a A_s + C_{vis}\dot{x}_p} \quad (7)$$

Valve ratio [6], $\lambda = \frac{\Delta P_\eta}{\Delta P_\tau} = \frac{3\eta HQ}{\pi g^2 R_1 c L_p \tau_y}$ (8)

On state pressure drop [2, 5, 6], $\Delta P_\tau = \frac{2cLp\tau_y}{g}$ (9)

Inductive time constant of the valve is [6, 7]:

$$T = \frac{2R_1 A_w \int_0^{L_p} B_{MR}(s)ds}{r\bar{d}IL_p} \quad (10)$$

where r represents the resistivity of the coil wire, 0.01726×10^{-6} Ωm for copper wire, and the average diameter of the coil is:

$$\bar{d} = R - d - g - \frac{W_c}{2}$$

L_p is the active length of the pole. It can be observed that the active length varies with the shape of the core ends in Figure 2. Accordingly, L_p of each configuration has been computed and used in the computation. Wherever the shape of the inner house is changing, only the change in active length on core end has been considered for simplifying the computation. g is the gap length of the duct in which the MR fluid is passing through. For the rectangular duct, the gap length is uniform throughout the active pole, whereas for the remaining ducts the gap length varies along the pole length, hence for each configuration the average gap length has been computed as follows:

$$g = \frac{\int_0^{L_p} g(s)ds}{L_p} \quad (11)$$

Total mass of MR valve consists of the core, inner house, coil and MR fluid. Material property of the valve used in this work is given in Table 1. Hence, it has been computed as:

$$m_v = m_{cr} + m_h + m_{co} + m_{MR} \quad (12)$$

3. FINITE ELEMENT MODELING OF MR DAMPER VALVES

The MR damper valves consist of a piston over which a coil is wound, and a gap is maintained between the inner and outer pistons.

TABLE 1. Values of parameters adopted for MR damper valve

Parameters	Values
Coil width, Wc	6 mm
Outer piston thickness, d	3 mm
Radius of the valve, R	16 mm
Height of the valve, H	20 mm
Copper wire diamete (for 24-guage). d_c	0.51 mm
Initial pressure of the gas chamber, P_o	5.45 N/mm^2
Initial volume of the gas chamber, V_o	6371.15 mm^3
Displacement of the piston, x_p	28.13 mm
Radius of piston shaft, R_s	5 mm
Range of applied current, I	0.1-1 Ampere
Number of turn of coil, N	294
Fillet radius, R_f	$L_p/3$
Chamfer distance, d_{ch}	$L_p/3$

The pistons are made up of low carbon cold rolled steel SAE 1020 due to its high relative permeability; the materials adopted are shown in Table 2. Nine configurations of damper valve with plain, chamfered and filleted ends are considered while keeping the dimensions of the piston, number of turns of the coil, and applied current constant. Details of 2 dimensional (2D) axi-symmetric configurations of the valve piston ends are shown in Figure 2 and the nine configurations are furthermore shown in Figure 3. The 2D axi-symmetric finite element models of the configurations of the damper valve are also shown in Figure 4. For computation of the PI for each configuration, the remaining dimensions of the valve and parameters are given in Table 1. These models are analyzed and the variation of magnetic field intensity and magnetic flux density with respect to the shapes along the active length of the pole are computed in ANSYS, 2014 with PLANE53 elements, as follows [7]:

$$B_{MR} = \frac{\int_0^{L_p} B_{MR}(s)ds}{L_p} \quad (13)$$

$$H_{MR} = \frac{\int_0^{L_p} H_{MR}(s)ds}{L_p} \quad (14)$$

where $B_{MR}(s)$ and $H_{MR}(s)$ are the magnetic flux density and magnetic field intensity at each of the nodal points located on the defined path, AA (Figure 5). The variation of magnetic flux density and magnetic field at 1 Ampere are shown in Figures 6 and 7 for the models, and the horizontal colorbars indicate corresponding values of B_{MR} (Tesla) and H_{MR} (A/m) across the models, respectively.

TABLE 2. Materials adopted

S.N	Valve components	Material and density	Relative permeability	Saturation flux (Tesla)
1	Valve Core	SAE1020 (7870 kg/m^3)	B-H Curve	2.390
2	Valve Housing	SAE1020 (7870 kg/m^3)	B-H Curve	2.390
3	MR-Fluid	MRF132 DG (2950 kg/m^3)	B-H Curve	1.65
4	Coil	Copper (24-Gauge) (8900 kg/m^3)	1	-

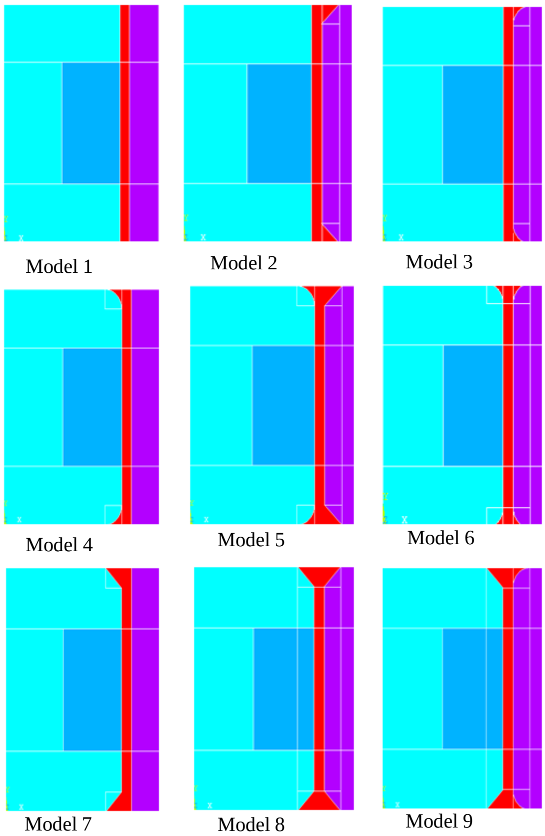

Figure 3. Simulation models with different piston valve configurations.

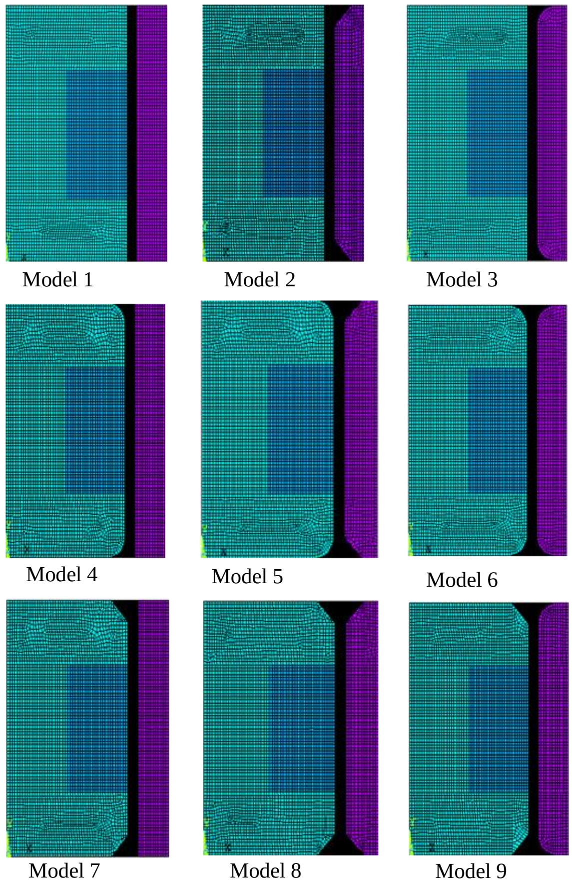

Figure 4. Finite element model of the valves

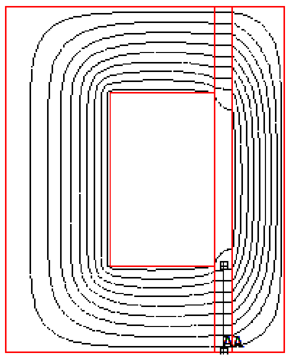

Figure 5. Flux lines around the electrical coil of Model 1and the path for computing H_{MR} and B_{MR}

4. SIMULATION RESULTS AND ANALYSIS

MR damper PI are computed for each configuration of the piston valve and the results are plotted for each performance index from Figures 8 to 12.

In designing an MR damper valve, light weight, high damping force, high dynamic range, low valve ratio, high on-state pressure drop due to yield stress and low inductive time constant are desired. Depending upon the desired application, one can choose the appropriate configuration of the damper valve.

The simulation results depicted in Figures 8 to 12 and in Table 2 show Model 1 to be the best performer in most of the PI except weight, inductive time constant and valve ratio. It can also be observed that Model 8 which has the lowest inductive time constant, which is desirable, is a moderate performer for weight and valve ratio but a poor one based on the other PI.

One can also observe that Model 7 has the lightest weight and shows relatively average performance indices.

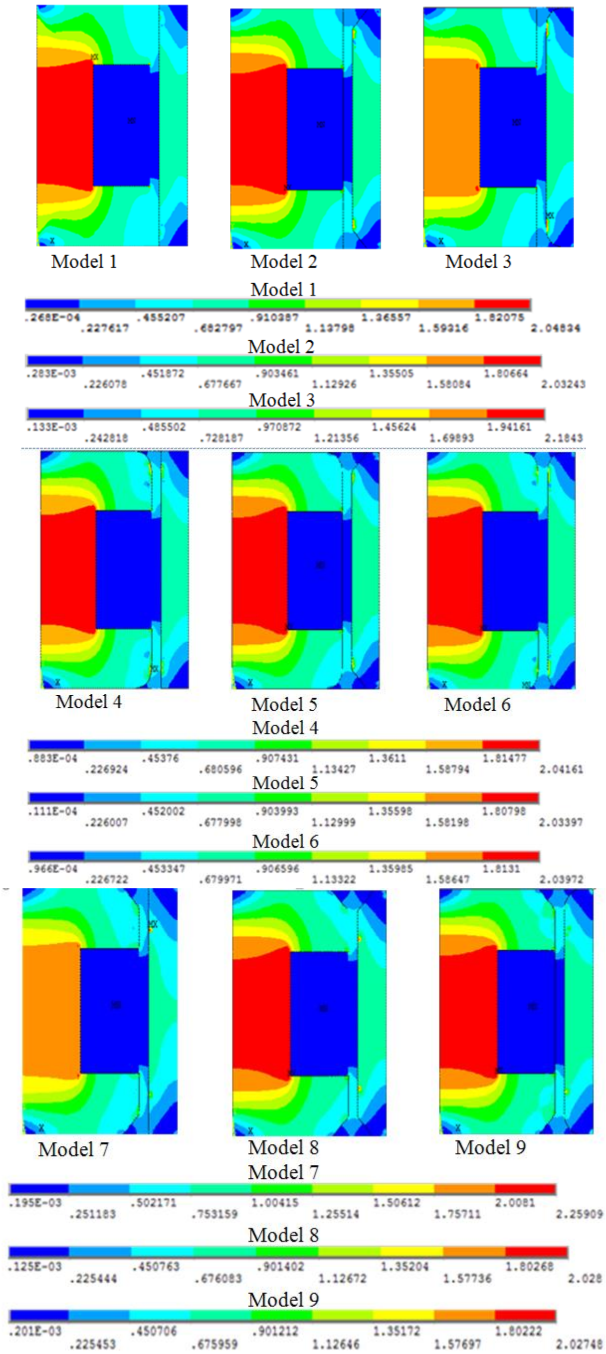

Figure 6. Magnetic flux density (Tesla) of the models at 1 Ampere.

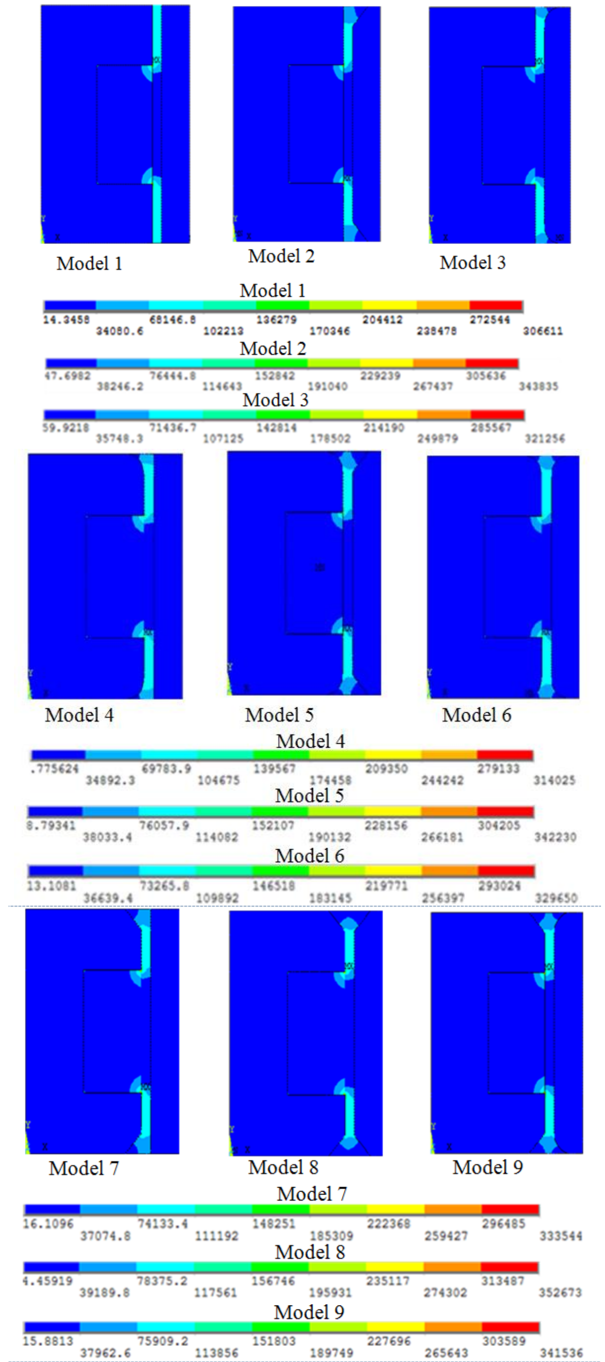

Figure 7. Magnetic field intensity (A/m) of the models at 1 Ampere

Models 5 and 6 also have a relatively low valve ratio. Therefore, when inductive time constant is highly desired, Model 8 performs better at a cost of losing out on the other PI. Where light weight is highly desirable, Model 7 is a suitable configuration for the MR damper valve.

For applications where low valve ratio is desirable, Model 6 is a suitable valve configuration. For applications where the remaining performance indices are desirable, Model 1 is shown to be the better MR damper valve configuration.

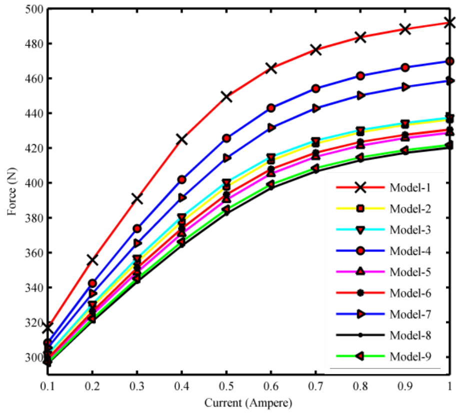

Figure 8. Damping force achieved by the models at different values of applied current.

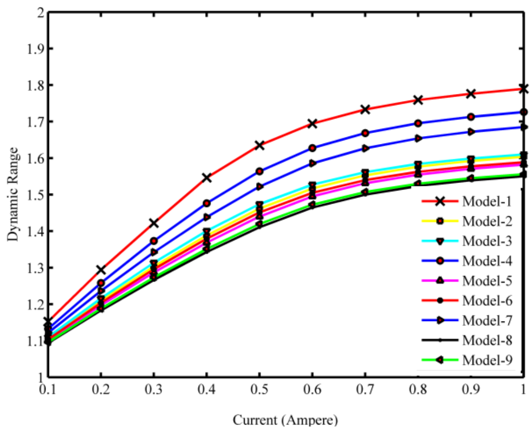

Figure 9. Dynamic range of the models at different values of applied current

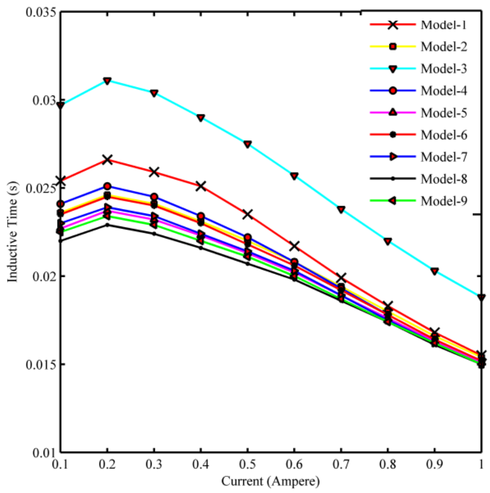

Figure 10. Inductive time constant for the models at different values of applied current.

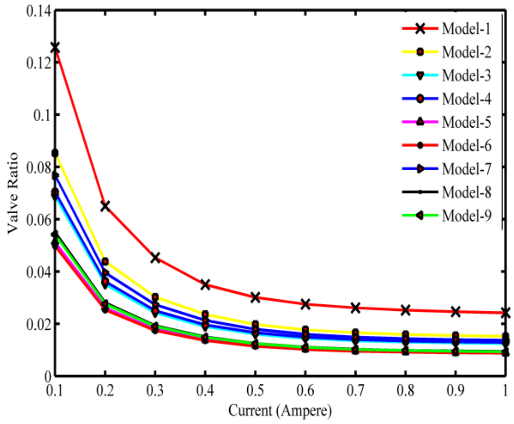

Figure 11. Valve ratio of models versus applied current.

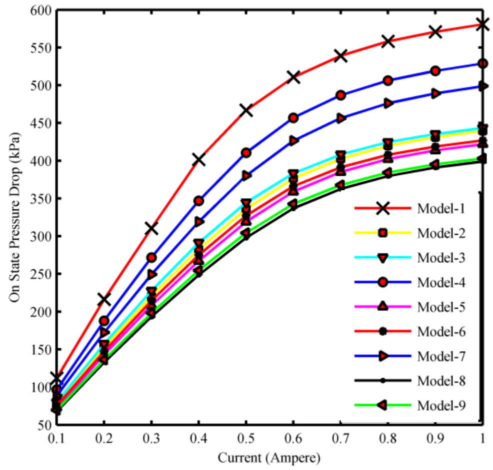

Figure 12. On state pressure drop achieved by the models at different values of applied current.

5. CONCLUSION

In this work, nine commonly used MR damper valve configurations have been investigated through finite element analysis for their damping performance indices. From simulation results, it can be concluded that when the designer is not very particular about inductive time constant, weight and valve ratio, the valve without modified ends of the piston performs better. When inductive time constant alone is more desirable than the rest of the PI, then the valve with chamfered ends of inner and outer pistons is an appropriate choice. For applications where valve ratio is more desirable than the other PI, the valve with fillet ends of inner and outer piston is suitable. For light weight, the valve with chamfered ends of the inner piston is suitable. Further experimental analysis can be carried out to verify the performance indices of the considered damper valves.

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Received 16 November 2016
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چکیده

هدف اصلی این مقاله بررسی پیکربندی‌های مختلف یک دریچه دمپر مگنتورئولوجیکال (MR) و ارزیابی شاخص‌های عملکرد آن از جمله محدوده دینامیکی، نسبت سوپاپ، ثابت زمان القائی و افت فشار است. مشخص شده است که این شاخص عملکرد (PI) دمپر به طراحی مدار مغناطیسی دریچه بستگی دارد. از این رو، نه تنظیمات دریچه‌ای درنظر گرفته شد و مدلهای ریاضی برای آنها توسعه داده شد. یک مدل المان محدود برای تجزیه و تحلیل و بررسی PI یک دریچه دمپر دوبعدی متقارن ساخته شد. تمام تنظیمات دریچه دمپر در داخل یک محدوده داده شده از جریان ورودی و تعداد سیم پیچ، شبیه‌سازی شده و در این محدوده، نیروی میرائی، محدوده دینامیکی، نسبت سوپاپ، ثابت زمان القائی و افت فشار ارزیابی گردید. نتایج شبیه‌سازی نشان می‌دهد که PI دمپر MR بسیار وابسته به شکل دریچه بوده و از این رو شکل دریچه باید بر اساس کاربرد مورد نظر انتخاب شود. نتایج به دست آمده در این کار به طراحان بینشی برای ایجاد دمپر-MR برای کاربردهای خاص می‌دهد.

***doi**: 10.5829/idosi.ije.2017.30.02b.18*