

## DESIGN, ANALYSIS AND PERFORMANCE EVALUATION OF THE LINEAR, MAGNETORHEOLOGICAL DAMPER

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**Abstract:** The paper issues the design and analysis of the linear, magnetorheological damper. Basic information concerning the characteristics of the typical magnetorheological fluid and the damper incorporating it, were presented with the short description of the applied fluid MRF-132 DG. Basing on the computations, the prototype damper T-MR SIMR 132 DG was designed, manufactured, and tested under different operating conditions. Presented calculations were verified with the experimental results and their accuracy was evaluated. The conclusions and observations from the research were compiled in the summary.

**Key words:** Vibration, Damping, Magnetorheological Fluid, MRF

### 1. INTRODUCTION

The recently observed increase of the interest in the field of smart fluids by the industrial centres, academic and research institutes, unfortunately does not translate into their popularisation in the engineering applications.

The number of solutions utilizing smart materials like magnetorheological (MR) fluids still persists in the number of applications supported with the typical, hydraulic and pneumatic systems. What holds back the application engineers from mass deployment of such devices might be the poor range of proposed and evaluated solutions incorporating the MR fluids. Another important element is the constantly high price of the fluids, provided mainly by the LORD Corporation, which dominated the market. The difficult access to the high-quality MR fluids also did not encourage constructors to design new solutions. The evolutionary devices are mainly prototype units, designed at the academic and research centres, and they are still in the experimental phase.

### 2. DESIGN OF THE PROTOTYPE DAMPER

The better market access to other manufacturers' fluids, as well as their improved parameters and quality, stimulates and encourages to creating, and modifying the shock absorbers, clutches, brakes and other devices.

Therefore there is a need to develop, a comprehensive algorithm for designing devices utilizing the unique properties of the magnetorheological fluids. The presented paper is a proposition of such an algorithm, and its experimental verification.

The linear damper prototype is presented as a blueprint in Fig. 1. The photo of the device, and the cross-section view of the annular flow gap, is presented in Fig. 2.

The typical magnetorheological fluid named MRF-132 DG manufactured by Lord Co., was used in the prototype. The fluid is a suspension of a 10 µm diameter sized, magnetically susceptible particles, in a carrier, hydrocarbon fluid. According to the

datasheet, the density of the liquid is around 3 g/cm<sup>3</sup> and viscosity of a 0.09 Pa·s. The maximum yield stress value is 50 kPa and it is achieved with the magnetic induction of 1.5 T.

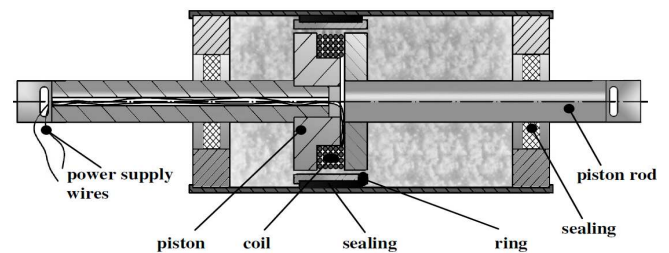


Fig. 1. Linear magnetorheological damper prototype, with its basic assemblies

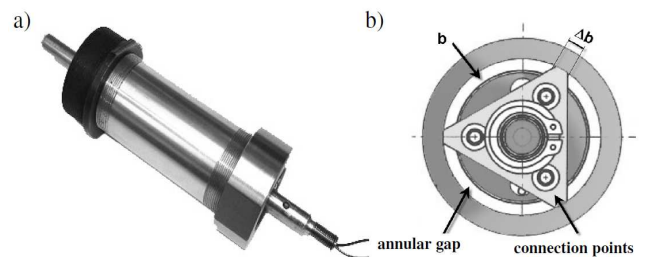


Fig. 2. a) Prototype of the T-MR SIMR 132 DG damper; b) cross-section view of the annular flow gap and the mount

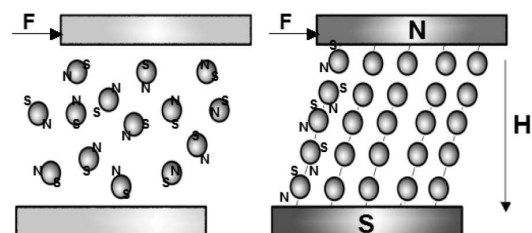


Fig. 3. The magnetorheological particle chains assemblies, in the presence of the magnetic field

When exposed to a magnetic field, the rheology of the fluid reversibly and instantaneously changes from a free-flowing liquid to a semi-solid state, with the controllable yield strength, as a consequence of the sudden change in the particles arrangement. Upon application of the magnetic field, particles align with the direction of the flux lines (Fig. 3), in a chain-like combination, thereby restricting the fluid's movement within the gap, in proportion to the strength of the magnetic field.

## 2.1. Major Simplifying Assumptions Concerning the Magnetorheological Fluid

Simplifications concerning the magnetorheological fluid behaviour were assumed prior to the experiments. The fluid features include the fast response time, and high yield strength in the presence of the magnetic flux, and very low yield strength in the absence of it, which allows for a wide range of controllability. The fluid also provides hard settling resistance, and it does not abrade the devices.

In further analysis it was assumed that the liquid properties would be described by the phenomenological, viscoplastic Bingham model. The properties of this model are illustrated in Fig. 4.

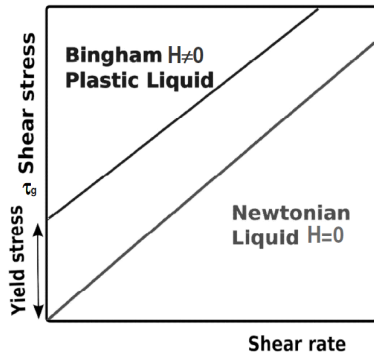


Fig. 4. Shear stress vs. shear rate in Bingham model for different liquid states, activated by the magnetic field

## 2.2. Simplifications concerning damper performance

The work regime of the MR fluid inside the damper is called the valve mode. In the valve mode, the fluid flows through an orifice, as presented in Fig. 5. In this model the annular flow is treated as a flow between two, parallel, still plates. The principle of operation resembles throttling (Kim et al, 2001). The resistance to flow of a liquid through the narrow gap is controlled by the changes in the magnetic field  $H$ , which vector is normal to the direction of the flow. The adjustment of the magnetic field is performed by the change of the current in the coil winding, mounted on the piston.

Further simplifications which were crucial for the computation of the damper's performance were:

- the damping force acts linear;
- the flow gap is formed by the stationary walls;
- the height of the gap is much smaller than its length and the width, therefore the flow is considered as a flow between parallel plates, thus the valve mode simplification is reasonable;
- stress value is constant along the gap, and it depends only

on the value of the magnetic flux in the gap (Mukhlis et al, 2006).

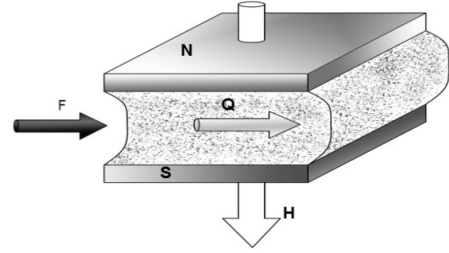


Fig. 5. The valve mode of the flow - the throttled flow through the gap

## 2.3. The Device Parameters Calculation

The prototype basic geometrical dimensions and the desired maximum damping force value were imposed. The gap view, with the basic parameters denotation is presented in Fig. 6.

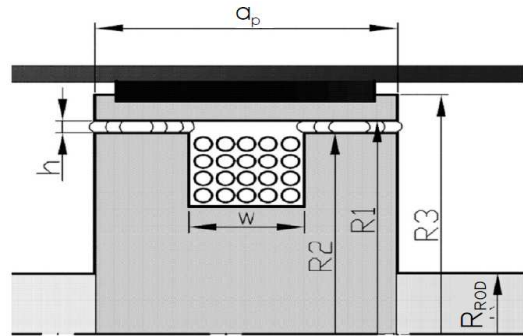


Fig. 6. View of the flow gap and the basic parameters denotation:  $R_1, R_2$  – outer and inner gap radius;  $R_3$  – piston radius;  $R_{rod}$  – piston rod radius;  $h$  – gap height;  $w$  – coil width;  $a_p$  – piston length

Proposing the approximate diameters of the piston and the rod, the effective circumference of the flow gap, decreased by the length of the connection points  $\Delta b$ , which are covering the gap (Fig. 2 b), may be expressed as:

$$b = \pi(R_1 + R_2) - 3 \cdot \Delta b \quad (1)$$

The effective area of the flow gap is equal to:

$$A_{gap} = b \cdot h \quad (2)$$

where:  $h = R_1 - R_2$  is the gap height.

The total linear damping force may be expressed as a sum of the particular forces (Gavin et al., 2001):

$$F_{damp} = F_\tau + F_{ns} = F_\tau + F_\mu + F_f \quad (3)$$

where:  $F_\tau$  – controlled force, related to the actual yield stress of the MR fluid,  $F_{ns}$  – non controlled damping force, related to the viscosity and friction,  $F_\mu$  – viscotic resistance force,  $F_f$  – friction force between piston and the cylinder.

For the axisymmetric flow, the equations describing the pressure gradient in the flow gap, simplified by the parallel plates model, for the Bingham characteristics, reduce to the 5th degree

equation (Phillips, 1969). This equation can be used for all type of the viscotic dampers with the parallel plates type of a gap:

$$3(\chi - 2\sigma)^2[\chi^3 - (1 + 3\sigma - \nu)\chi^2 + 4\sigma^3] + \sigma\nu^2\chi^2 = 0 \quad (4)$$

where:  $\chi$  – dimensionless pressure gradient,  $\sigma$  – dimensionless stress gradient.

The value of the dimensionless velocity is expressed as:

$$\nu = \frac{bh\nu}{2Q} \quad (5)$$

where  $\nu$  – velocity of the piston.

The volumetric flow rate of the MR fluid through the gap can be calculated as:

$$Q = \nu \cdot A_p \quad (6)$$

where in the formula above, the surface area of the piston, denoted as  $A_p$  is calculated as

$$A_p = \pi R_3^2 - A_{gap} - \pi R_{rod}^2 \quad (7)$$

The pressure gradient  $\chi$  and the stress gradient  $\sigma$  are described by the formula (Milecki, 2010):

$$\chi = \frac{bh^3 \Delta p}{12 \cdot Q \mu a_p} \quad (8)$$

$$\sigma = \frac{bh^2 \tau_0}{12 Q \mu} \quad (9)$$

Parameter denoted as  $a$  is the length of the area, in which the magnetic flux influences the MR fluid:

$$a = a_p - w \quad (10)$$

In the absence of the magnetic field the dimensionless stress  $\sigma = 0$ , therefore the equation (4) simplifies to a form:

$$\chi = 1 - \nu = 1 - \frac{bh\nu}{2Q} \quad (11)$$

From the equation (8) we can state, that the pressure drop caused by the Newtonian fluid viscotic flow, can be expressed as:

$$\Delta p_\mu = \frac{12 A_p^2 a_p \nu \mu \chi}{bh^3} \quad (12)$$

The damping force related only to the viscosity of the fluid is equal to:

$$F_\mu = \Delta p_\mu A_p \quad (13)$$

Substituting the relation (12) to (13):

$$F_\mu = \frac{12 A_p^2 a_p \nu \mu \chi}{bh^3} \quad (14)$$

Combining the (11) and (14) the equation, the total viscotic force value is obtained:

$$F_\mu = \left(1 - \frac{bh\nu}{2Q}\right) \frac{12 A_p^2 a_p \nu \mu \chi}{bh^3} \quad (15)$$

Including the previous equations, and utilizing the simplifying formula ( $A_p \gg b$ ), the equation above can be transformed into:

$$F_\mu = \frac{12 A_p^2 a_p \nu \mu}{\pi(R_1 + R_2)h^3} \quad (16)$$

Using the approximate solution of (4) we obtain:

$$\chi(\sigma, \nu) = 1 + 2,07\sigma - \nu + \frac{\sigma}{1 + 0,4\sigma} \quad (17)$$

The decrease of the pressure along the gap, caused by the non-zero yield stress value is given by the equation (Kęsy, 2008):

$$\Delta p_\tau = c \frac{\tau_0(B)a}{h} \quad (18)$$

The constant value  $c$ , which is related to the geometrical dimensions of the damper and the damping force, can be approximated with the accuracy of 3% by the relation below (Poynor, 2010):

$$c = 2,07 + \frac{1}{1 + 0,4\sigma} \quad (19)$$

Taking into consideration the simplifying assumptions, we can determine the value of the damping force, controlled with the magnetic field:

$$F_\tau = k_p \Delta p_\tau A_p \quad (20)$$

where:  $k_p$  – constant, correction factor, related to the roundness of the magnetic field lines that are increasing the magnetic field area.

The final form of the sum of the forces is then as follows:

$$F_{damp} = k_p c \frac{\tau_0(B)a A_p}{h} + \frac{12 A_p^2 a_p \nu \mu}{\pi(R_1 + R_2)h^3} + F_f \quad (21)$$

The value of the friction force  $F_f$  depends on the type of the used sealing, construction materials, type of their machining, piston velocity, duration of the piston rest time and other.

In the discussed example of the calculations for the prototype device, the friction force value was preliminarily estimated on the basis of the experimental data. From the equation (16) and (20), it can be concluded, that by reducing the height of the flow gap, the maximum damping force of the device can be increased. The increase of the non-controlled visco-elastic force is in proportion to the third power of the gap height. The value of the controlled force increases slower than the value of the uncontrolled force. It is caused by the fact that the height of the gap in equation (20) is in the first power.

The presented order of calculation allows determining most of the parameters which are crucial for the proper design of the linear damper with MR fluid. Additionally, it is necessary to determine the maximum value of the energy dissipated by the damper. The value of the energy corresponds to the value of the work, which is done by the non-Newtonian fluid during its movement through the flow gap, and it can be calculated as:

$$W = Q(\Delta p_\tau + \Delta p_\mu) \quad (22)$$

Another important parameter is the controlled force range, which is the ratio of the controllable force to the non controllable one:

$$D = \frac{F_\tau}{F_\mu + F_f} = \frac{c \tau_0(B) b h^2 \Delta p}{12 W \mu + F_f} \quad (23)$$

To achieve the widest possible range of the controlled damping force, the value of this parameter should be maximized.

### 3. REASERCH OF THE T-MR SIMR 132 DG DAMPER

Three different gap heights were considered in the experiments:  $h = 5 \cdot 10^{-4}$  m,  $7 \cdot 10^{-4}$  m and  $10^{-3}$  m. Utilizing the equations presented in Chapter 2, the damper research data was expected to match the parameters summarized in Tab. 1. Also, the comparison between experimental results and computational ones was attached.

**Tab. 1.** Comparison of the maximum damping force, obtained computationally and experimentally for different operating conditions

DESIGN PARAMETERS										
Dynamic viscosity of the fluid $\mu = 0.092$ Pa · s										
Maximum Yield stress $\tau_0 = 48 \cdot 10^3$ Pa										
Flow gap height $h$ [m]	$5 \cdot 10^{-4}$			$7 \cdot 10^{-4}$			$10^{-3}$			
CALCULATED DAMPING FORCE $F_{damp}$ [N]										
$I$ [A]	0.1	0.2	0.4	0.5	1.0	2.0	0.5	1.0	2.0	
Velocity $v$ [ $10^{-3}$ m/s]	17	755	1110	1395	820	900	1095	400	465	500
	33	790	1145	1430	830	915	1105	405	470	505
	50	825	1180	1465	842	925	1075	410	475	510
EXPERIMENTAL DAMPING FORCE $F_{damp}$ [N]										
Velocity $v$ [ $10^{-3}$ m/s]	17	730	980	1165	830	930	1100	500	565	630
	33	800	1130	1370	930	1065	1230	530	620	700
	50	870	1200	1420	1050	1150	1290	560	675	740

**Tab. 2.** Comparison of the obtained relative errors of the damping force for different operating conditions

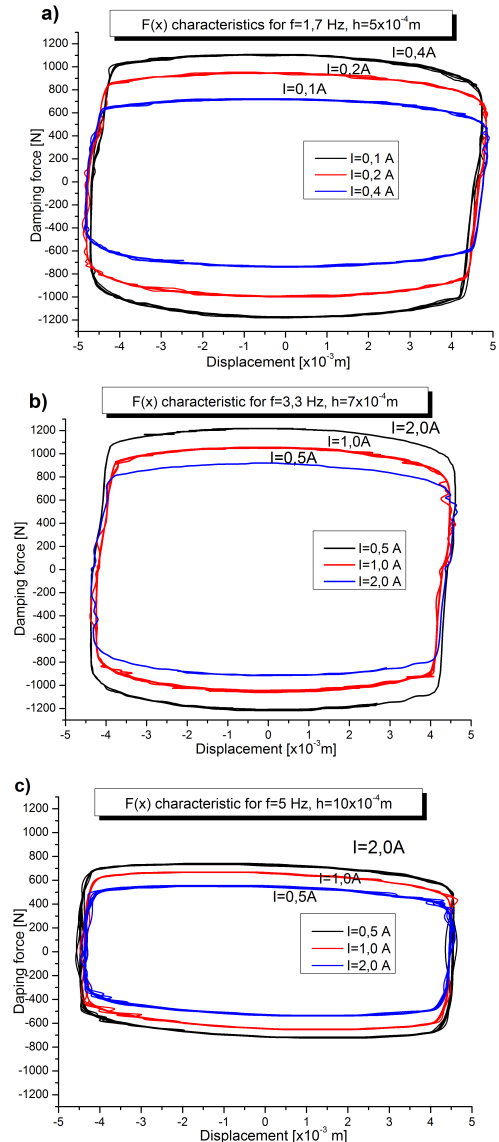
COMPUTATIONAL RELATIVE ERROR $F_{damp}$ [%]										
Flow gap height $h$ [m]	$5 \cdot 10^{-4}$			$7 \cdot 10^{-4}$			$10^{-3}$			
$I$ [A]	0.1	0.2	0.4	0.5	1.0	2.0	0.5	1.0	2.0	
Velocity [ $10^{-3}$ m/s]	17	3	13	19	1	3	1	20	18	20
	33	2	2	4	11	14	10	24	24	28
	50	5	2	3	19	19	16	27	29	31
AVERAGE ERROR OF $F_{damp}$ [%] over a piston velocity										
$\delta_{avg}(v)$	3	6	9	10	12	9	24	24	26	
AVERAGE ERROR OF $F_{damp}$ [%] over a coil current										
$\delta_{avg}(I)$	17	11		2			19			
	33	3		12			25			
	50	3		18			29			
TOTAL AVERAGE ERROR OF $F_{damp}$ [%]										
$\delta_{total}(v, I)$	6			11			25			

The percentage error of the value of the damping force for different operating conditions is presented in Tab. 2. Sample results of the  $F(x)$  relation, obtained for different piston velocities and several coil current values, are presented in Fig. 7.

The results show a relevance of the applied equations for de-

termining the parameters of the designed damper with MR fluid. The smallest error values relating to the damping force are obtained for the narrowest gap. Depending on the coil current, the average error over piston velocity ranged from 3% to 6%. The value of the error over coil current ranged from 3% to 11%.

The increase of the gap height enhanced the disparity between the computational and the experimental values. By analysing the average error over the piston velocity, it can be stated, that the increase of the piston velocity amplifies an error of the damping force estimation. This error is greater, as the height of the flow gap increases. This effect may be related to the simplifications in calculation of the pressure drop.



**Fig. 7.** Experimental characteristics of the  $F(x)$  relation for: a) gap height  $5 \cdot 10^{-4}$  m and frequency 1,7 Hz; b) gap  $7 \cdot 10^{-4}$  and frequency 3,3 Hz; c) gap  $10 \cdot 10^{-4}$  and frequency 5 Hz

Increasing the current in the device's coil enlarges the magnetic field flux and thus, increases the yield stress denoted as  $\tau_0(B)$ . Precise computations of the magnetic induction are complicated, while the accurate experimental research is time and effort consuming. The  $\tau_0(B)$  function provided by the manufacturer of the fluid, is usually only approximate and imprecise,

which negatively influences the accuracy of the computations. It is caused by the fact that the maximum damping force linearly depends on the value of the yield stress. Difficulties in determining the magnetic induction value explain the high value of the  $\delta_{avg}(I)$  error.

The continuous line in Fig. 8 shows the experimental data for the maximum force  $F_{damp}$  for the highest coil current. It was compared with the computational results marked as the dotted line. For the gap height of  $5 \cdot 10^{-4}$ , the calculated values tend to the experimental ones, as the velocity of the piston  $v$  increases. For the other gap heights, the results diverge.

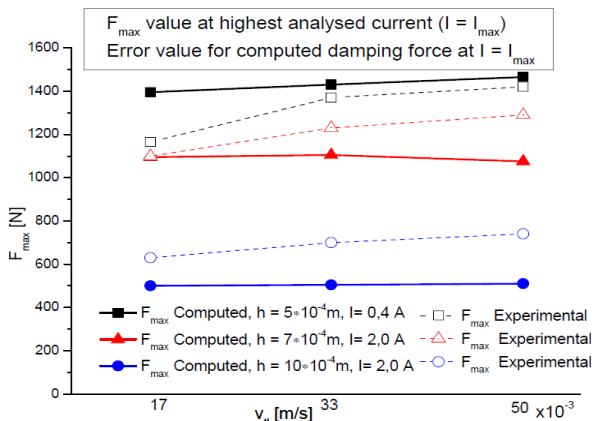


Fig. 8. Comparison of the theoretical and experimental data

Fig. 9 presents the dependence of the relative error of the maximum damping force as a function of the piston velocity, for the highest current values studied.

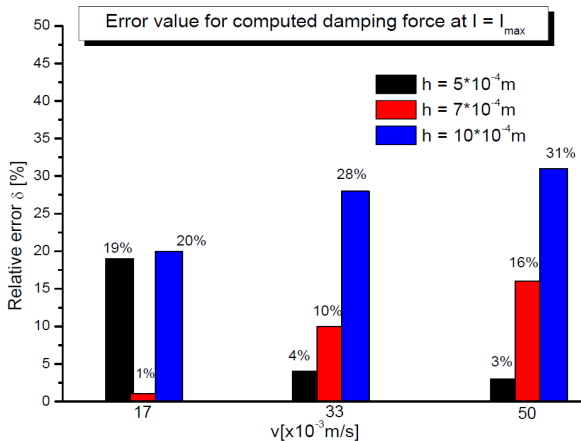


Fig. 9. Error of the damping force value calculation over velocity of the piston, for the highest coil current values

For the gap values  $7 \cdot 10^{-4}$  m and  $10^{-3}$  m, with the increase of the velocity, the total error of force estimation also increases. The greater the annular gap is the higher the inaccuracy caused by the simplification of the parallel plate flow model gets.

For these values of the gap height, the value of the error is associated with imprecise determination of the magnetic induction in the gap. With the increase of the gap height, the magnetic flux lines distorts, which leads to extending of the zone in which the fluid is in the active state. This may explain the fact of the

undervaluation of the computed damping force for the gap of height  $7 \cdot 10^{-4}$  m, and  $10^{-3}$  m compared to the experimental results.

#### 4. FINAL CONCLUSIONS

The ramification of this paper is a numerical tool that allows to initially calculate the damping force value of the linear, magneto-rheological damper. The manufactured prototype allowed verifying the theoretical equations with the experimental results. The assumed simplifications of the phenomena connected with the operation of the device, revealed a major influence on the computational inaccuracies.

It can be concluded that the most accurate calculations can be obtained for the smallest gap height, due to the possibility of the precise determination of the magnetic field, and the small error of the simplified model of the flow between parallel plates.

The analysis suggests the need to develop more precise tools supporting the design process of the devices with MR fluids. It seems reasonable to create a reverse algorithm that will allow estimating the geometry of the device, basing on the desired value of the dissipated energy. In addition, it is necessary to determine more accurately the value of the magnetic induction in the flow gap of the MR device. It would be also interesting to define the influence of the temperature on the viscosity and the yield stress, as well as to take this influence into account for the theoretical calculations.

#### REFERENCES

1. Gavin H., Hoagg J., Dobossy M. (2001), *Optimal Design of MR Device, Proc. U.S.-Japan Workshop on Smart Structures for Improved Seismic Performance in Urban Regions*, Seattle, 225-236.
2. Kęsy Z. (2008), *Sprzęgła z cieczami elektro- i magneto-reologicznymi*, wyd. Politechnika Radomska, Radom
3. Kim J.-H., Lee Ch.-W., Jung B.-B., Park Y. (2001), Design of Magneto-Rheological Fluid Based Device, *KSME International Journal*, Vol. 15 No. 11, pp. 1517-1523.
4. Milecki A. (2010), *Electro- and magneto-rheological fluids and their application in technique (in Polish)*, wyd. II, wyd. Politechniki Poznańskiej, Poznań, 42-60.
5. Mukhlis R. Z., Mohamed N., Mohd Nor M. J. (2006), The design and simulation of flow mode electrorheological damper, *Jurnal Teknologi*, University of Technology Malaysia, 44(A) Jun 2006, 115-125.
6. Poynor J. C. (2001), *Innovative Designs for Magneto-Rheological Dampers*, Master of Science in Mechanical Engineering degree thesis, Mehdi A., Virginia Polytechnic.
7. Phillips R. (1969), *Engineering applications of fluids with a variable yield stress*, Ph.D. Thesis, University of California, Berkeley.