Design and Simulation of Magneto-Rheological Dampers for railway applications

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SUMMARY. The properties of magneto-rheological fluids have been investigated since the end of the II World War[1][2]. However, problems related to cost and stability of the fluid properties have stopped the development of industrial products based on this technology until the nineties, when further basic research activities have lead to more durable and reliable fluids[3][4]. After this improvements, applications of magneto-rheological fluids to controllable dampers and clutches have been widely explored especially for automotive[5][6] and military applications[7]. Recent studies have addressed technical problems related to railway applications, in particular for semi-active suspension systems of high speed trains[8][9]. In this study the authors show a method to design magneto-rheological semi active dampers for railways application by FEM models. As a case study, a mono-tube damper for the secondary suspension stage of a high speed train is described.

1 INTRODUCTION

In order to improve train performance, several studies have been carried out over railway mechatronic sub-system. In the latest days, thanks to the improvements in magneto-rheological fluids, several leader companies are investing heavy resources in the development of mechatronic boogies since improvement in the behavior of this key subsystem allows to increase train speed and passenger comfort. In this work authors have investigated the feasibility of a magneto-rheological semi-active damper that can be applied to the EMU V250 developed by AnsaldoBreda for the Dutch railways. So, in order to ensure the interchangeability of the existing passive damper with the designed new semi-active one, specifications concerning forces, encumbrances, dynamical response have been fulfilled. In order to evaluate the cross coupling between mechanical, fluid dynamics and magnetic behavior of the system, a multi physics FEM model has been developed in the Comsol Femlab™ environment. To validate the model, a comparison with experimental data of a real damper available in the literature [12] has been done. After this verification, the authors have chosen a standard magneto-rheological fluid, the MRF-132DG [10], produced by Lord™ corporation. A damper has been then designed by defining diameter, gas reservoir pressure, and windings placed in the body piston in order to achieve the desired response by means of force/damper coefficient.

2 DESIGN AND VALIDATION OF the FEM MODEL

As stated above, a magneto-rheological damper is a component where mechanical, magnetic and fluid dynamics properties interact with each other. In order to evaluate this cross coupling, the multi-physics FEM environment Comsol Femlab™ has been used. In this code it is possible to take into account the interaction of different physics phenomena over a common mesh. In order to correctly model the highly non-Newtonian behavior of the magneto-rheological fluid, the chemical package has been used and, in particular, the Carreau model that is often used for injection molding of plastic materials.

In order to validate this kind of procedure, the authors have compared experimental results available in the literature [12] with data obtained from a simple FEM model. In fact Y.K. Lau e W.H. Liao [12] have realized an hand made magneto-rheological damper and in their article geometrical dimension and the chosen fluid properties are shown in Fig.1 .

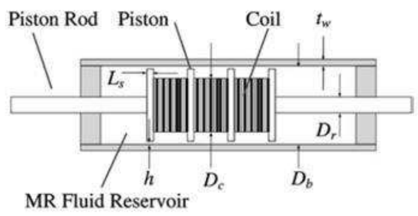

Table 1 Parameters of the MR damper

Body tube inner diameter (D_b)	80 mm
Thickness of the casing wall (t_w)	4 mm
Piston rod diameter (D_r)	25 mm
Piston core diameter (D_c)	40 mm
Total pole length (L)	42 mm
Thickness of gap (h)	0.8 mm
Cylinder length	218 mm
Piston rod length	616 mm
Stroke length	80 mm
Wire gauge	19 AWG

Figure 1: Magneto-rheological damper scheme and dimensions of the Y.K. Lau e W.H. Liao prototype.

As visible in Fig.1, the electrical coil is split into three and wrapped in opposite ways; by doing so, it is possible to distribute the magnetic field along the damper duct in a more uniform way.

The properties of the magneto-rheological fluid chosen by Y.K. Lau e W.H. Liao are similar to those of the MRF-122EG fluid of the Lord$^{\text{TM}}$ corporation. The different behavior between this kind of fluids with respect to Newtonian ones is clearly visible in Fig.2. The behavior of a magneto-rheological fluid is well represented by the Bingham (1) model:

$$\tau = \tau_0(H)\mathrm{sign}(y) + \eta y, \tag{1}$$

where:

- τ = Shear stress $[\mathrm{Pa}]$
- τ_0 = Static shear stress $[\mathrm{Pa}]$ function of the magnetic field value
- H = Magnetic field $\left[\frac{\mathrm{A}}{\mathrm{m}}\right]$ applied to the fluid
- y = shear rate $\left[\frac{1}{s}\right]$
- η = fluid viscosity $[\mathrm{Pa} \times \mathrm{s}]$

From a computational point of view implementing the Bingham equation is quite difficult due to the $\mathrm{sign}(\bullet)$ operator discontinuity so, in order to achieve a smoother characteristic the Carreau model has been used (2).

$$\eta = \eta_\infty + (\eta_0 - \eta_\infty)[1 + (\lambda y)^2]^{\frac{n-1}{2}}, \tag{2}$$

where

- η = value of viscosity $[\mathrm{Pa} \times \mathrm{s}]$

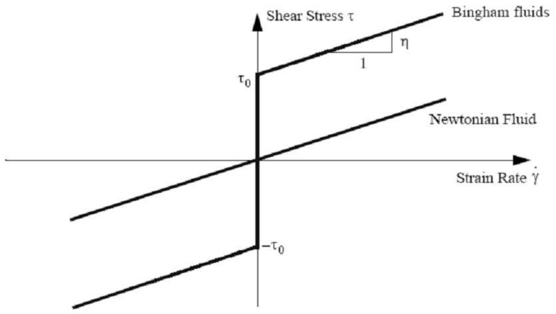

Figure 2: Newtonian and Bingham fluid behavior.

- η_0 = viscosity for shear rate equal to zero $[\mathrm{Pa} \times \mathrm{s}]$
- η_∞ = viscosity for high shear rate $[\mathrm{Pa} \times \mathrm{s}]$
- λ = Carreau constant $[s]$
- n = Carreau constant $(n \leq 1)$
- y = shear rate $\left[\frac{1}{s}\right]$

The influence of η_0 and λ over the behavior of the viscosity/shear stress curve is well explained in Fig.3 where the Carreau behavior is represented.

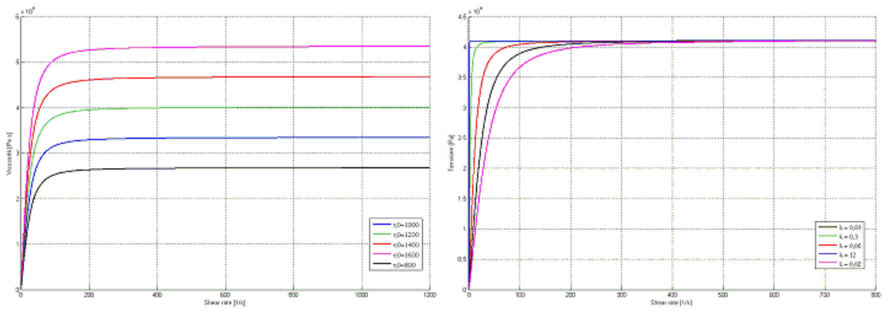

Figure 3: Carreau fluid behavior with different value of η_0 and λ.

By looking at Fig.3 it is easy to understand why this kind of representation is suited to model a magneto-rheological fluid:

- higher values of λ lead to a Bingham behavior of the equation;
- change of η_0 translate, without a change in the slope, the viscosity/shear rate curve as for the magnetic field effect of the Bingham model.

In addition, this kind of representation is not affected by the discontinuity of the $sign$ operator so it is more affordable from a computational point of view. In order to model the MRF-122EG, after some attempts, the authors have chosen the values for the Carreau model shown in Table 1.

η_∞	λ	n
[Pa $\times$ s]	[s]	
0.092	0.03	0

Table 1: Carreau parameters

After the definition of the mathematical model to be used for the magneto-rheological fluid, the authors have realized a Comsol FEM model of the damper described in [12]. In order to reduce the computational time of the analysis, and considering the symmetry of the damper, the authors have used an axially symmetric model as shown Fig.4

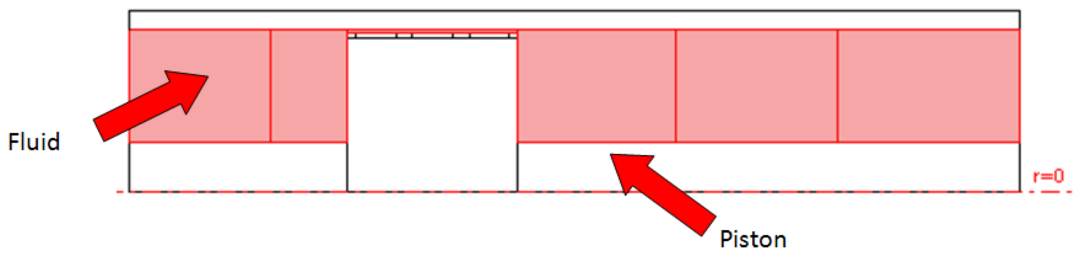

Figure 4: Axially symmetric model of the magneto-rheological damper.

Once defined the geometry of the model the authors have realized the mesh of the entire system. Of course, due to the electromagnetic analysis of the damper, either the piston or the metallic parts of the piston housing have been meshed Fig.5.

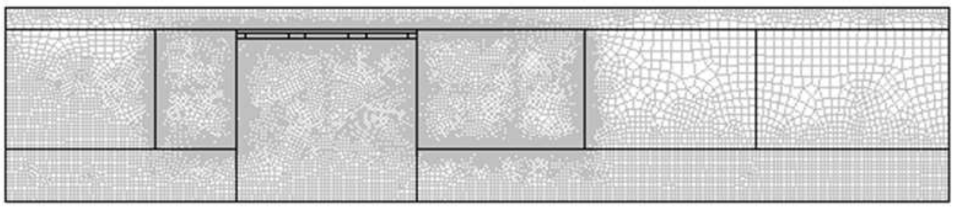

Figure 5: Mesh of the axially symmetric model of the magneto-rheological damper.

2.1 *Electromagnetic analysis*

In order to perform the electromagnetic analysis is necessary to describe two different kind of materials: the iron and the magneto-rheological fluid. For both this kind of material it is possible to get the electromagnetic induction/magnetic field characteristics so it is easy, after the definition of the boundary conditions, to compute the magnetic field distribution along the damper duct Fig.6.

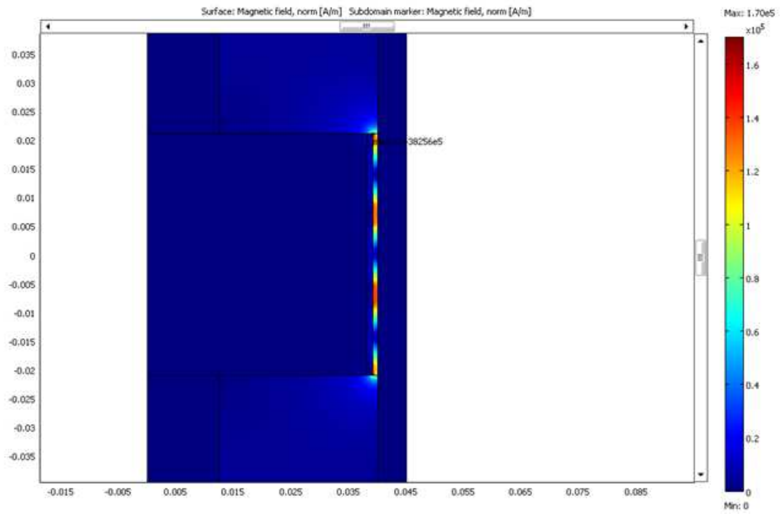

Figure 6: Magnetic field behavior in the dumper duct.

As expected, the magnetic field reaches maximum values in the duct in front of the coil winding so, splitting the coil is an effective strategy in order to get a better distribution of the field on the fluid duct.

2.2 *Fluid dynamic analysis*

After the magnetic analysis the authors have set the FEM model in order to do the fluid dynamic analysis. Of course the analysis has been performed on the mesh where the fluid flows and taking into account the previous calculated values of the magnetic field. To perform a correct analysis the authors have placed the reference system on the damper shank. In this way the metallic part of the damper is fixed in the space and it is possible to analyze the damper behavior only by moving the fluids on the opposite way of the shaft. Boundary conditions are the velocity profile for the inlet port and the pressure for the outlet port. By knowing the pressure drop in the chamber and the piston area of the damper it is possible to compute the force applied to the damper shaft. Of course, in order to solve this kind of problem, it is necessary to describe the viscosity/shear rate and viscosity/magnetic field of the magneto-rheological fluid as the ones published by Lord^{TM}corporation Fig.7.

With the results obtained from this second simulation, which takes into account even the result of the first magnetic simulation, is possible to compare (Tab.2) the force value obtained with COMSOL with the ones obtained by Y. K. Lau e W. H. Liao [12].

As reported in Tab.2 the experimental results [12] are almost the same of the ones evaluated by the FEM model. This result validates this model and this procedure so, in this way, it is possible to evaluate and design even different semi-active magneto-rheological damper.

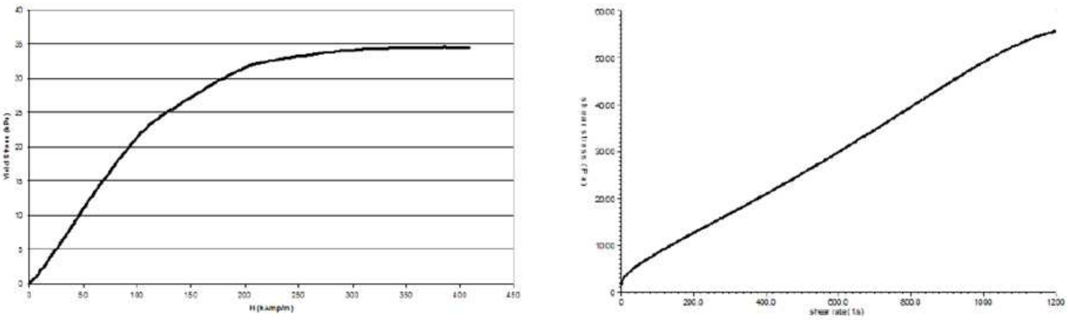

Figure 7: Technical properties of the MRF-122EG magneto-rheological fluid.

Velocity $[\frac{\mathrm{m}}{\mathrm{s}}]$	Coil Current [A]	Experimental force [kN]	FEM model force [kN]
0.06	0	1.5	1.4
0.06	0.5	4.4	4.5
0.06	1	7.1	7.4
0.06	1.5	9.6	9.9
0.06	2	11.5	11.8

Table 2: Comparison of the FEM model and experimental result

3 DESIGN OF SEMI-ACTIVE DAMPER FOR EMU V250

After the validation of the FEM model methodology the authors have designed a new magneto rheological damper thinking to an existing high speed train, the EMU V250, developed by Ansaldo-Breda for Dutch railways. In order to design a new magneto-rheological damper it is necessary to make some assumption firstly the authors have decided to use a standard mono-tube one, with gas reservoir in order to assure fluid pre-load and three electrical coil with metal windings, placed in the body of the pistons, and wrapped in opposite ways in order to assure an adjustable homogeneous magnetic field over the magneto-rheological fluid in the damper duct. By using Eq. (3) it is possible to obtain a first roughly dimension of the piston metal core and so the maximum radial dimension of the electrical coil.

$$\phi_{\mathrm{fl}} = \phi_{\mathrm{st}} \Rightarrow B_{\mathrm{fl}} A_{\mathrm{fl}} = B_{\mathrm{st}} A_{\mathrm{st}} \quad , \tag{3}$$

where:

- ϕ_{fl} = magnetic flux in the fluid
- ϕ_{st} = magnetic flux in the steel core of the piston
- B_{fl} = magnetic flux density in the fluid
- A_{fl} = area of the fluid duct
- B_{st} = magnetic flux density in the steel core of the piston

- A_{st} = area of the steel core of the piston

By using the two equations supplied by Lord™ corporation to evaluate the magnetic (4) and viscosity (5) forces, it is possible to compute the total force (6) of the damper as a function of the magnetic field and flow rate of the fluid.

$$F_\tau = \frac{c\tau_0 L_{\mathrm{pc}} A_{\mathrm{p}}}{g} \tag{4}$$

$$F_\eta = \frac{12\eta Q L_{\mathrm{p}} A_{\mathrm{p}}}{\pi g D_{\mathrm{p}}} \tag{5}$$

$$F = F_\tau + F_\eta = \frac{c\tau_0 L_{\mathrm{pc}} A_{\mathrm{p}}}{g} + \frac{12\eta Q L_{\mathrm{p}} A_{\mathrm{p}}}{\pi g D_{\mathrm{p}}} \quad , \tag{6}$$

where:

- F_τ = force in the fluid due to the magnetic field
- c = parameter function of $\frac{F_\tau}{F_\eta}$
- F_η = force in the fluid due to the fluid viscosity
- τ_0 = yield stress due to the applied magnetic field
- L_{pc} = total length of the polar expansion
- A_{p} = piston area
- g = height of the fluid duct
- η = kinematic viscosity
- Q = volumetric flow rate
- L_{p} = piston length
- D_{p} = piston diameter
- F = total force of the damper

From (4)(5)(6) it is possible to achieve a first-attempt design of the geometrical features of the desired damper. From this result a rough design of the FEM model (Fig.8) of the semi-active damper has been designed. From this first model the authors have performed several magnetic and fluid-dynamics analysis in order to iteratively adjust the dimensions of the damper to avoid saturation of the magnetic circuit and to obtain the desired maximum force. At the end of this optimization procedure they got the results described in the following.

As visible in Figg. 9 and 10, there is a huge difference of behavior when the magneto-rheological fluid is inside a magnetic field. In order to evaluate the goodness and the controllable range of the damper a performance index of D called dynamic range is usually evaluated:

$$D \stackrel{\mathrm{def}}{=} \frac{F_\tau + F_\eta + F_{\mathrm{f}}}{F_\eta + F_{\mathrm{f}}} \tag{7}$$

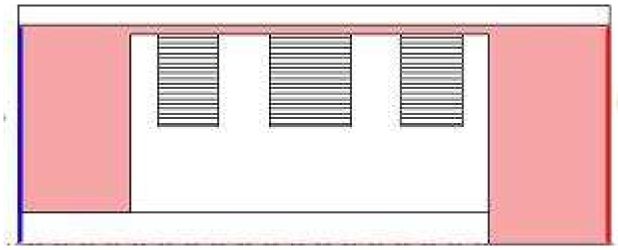

Figure 8: FEM model of the new damper.

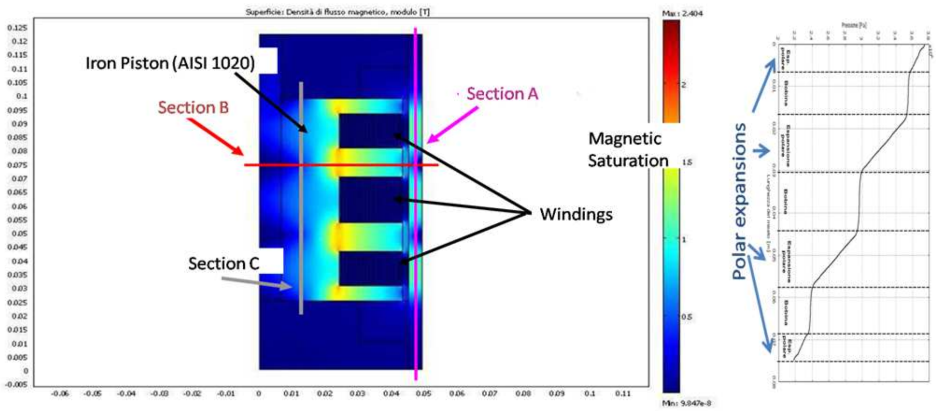

Figure 9: Comsol FEM Model of the damper, magnetic flux distribution, and corresponding pressure drop in the fluid flow on section A.

where F_{f} is the friction force. Friction force F_{f} is difficult to evaluate and it's usually consider equal to F_η in addition $F_\tau + F_\eta \gg F_{\mathrm{f}}$ so it is possible to assume that the final equation is:

$$D = \frac{F_\tau + F_\eta}{2F_\eta} \quad . \tag{8}$$

Where the maximum force achievable from the magneto-rheological damper (maximum current) is compared with the minimum one (no current on the electrical coil). From the simulations performed D results 12.1 which is in line with the magneto-rheological damper state of the art $(10 - 20)$.

4 CONCLUSIONS

In figure 1 a picture of the model is shown, also some results concerning magnetic flux density and corresponding pressure drop on fluid low are visible. Both Steady state and transient simulations have been performed. FEM model validation have been managed comparing results of the simulation

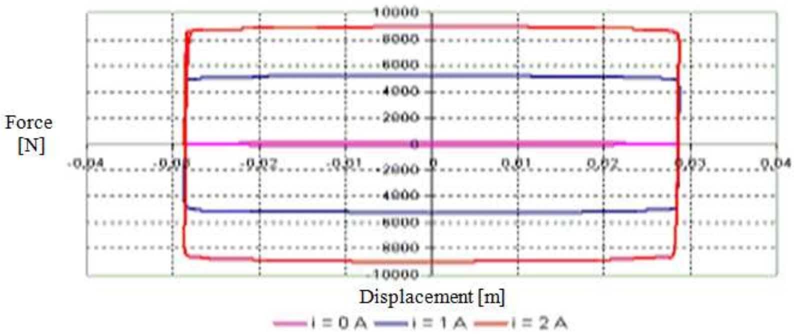

Figure 10: Force exerted by the actuator with sinusoidal imposed motion(max speed about 0.1m/s) with different current applied.

with experimental data of a real damper available in bibliography [12]. In this way it has been possible to indirectly verify that the followed approach was leading to quite realistic results. Once calibration/validation phase has been concluded, geometric shape and windings and the design of the magnetic circuit have been iteratively modified in order to fulfill the imposed specifications. In this way authors have been able to design a semi-active damper with the some encumbrances of the corresponding passive one but able to continuously regulate the exerted force from near to zero to above 8$[\mathrm{kN}]$ has shown in Fig. 10 where is reproduced the simulated response of the damper with an imposed sinusoidal motion. In the picture is clearly recognizable that actuator force is quite independent from the running speed of the actuator. As a consequence this behavior is quite suitable for control related applications

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