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# Multiphysics behavior of a magneto-rheological damper and experimental validation

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ABSTRACT

This investigation deals with the design, manufacturing, and testing of a large-capacity MR damper prototype. The MR damper uses external coils that magnetize the MR-fluid as it moves out of the main cylinder through an external cylindrical gap. In its design, multi-physics numerical simulations are used to better understand its force–velocity constitutive behavior, and its eventual use in conjunction with tuned mass dampers for vibration reduction of high-rise buildings. Multi-physics finite element models are used to investigate the coupled magnetic and fluid-dynamic behavior of these dampers and thus facilitate the proof-of-concept testing of several new designs. In these models, the magnetic field and the dynamic behavior of the fluid are represented through the well-known Maxwell and Navier–Stokes equations. Both fields are coupled through the viscosity of the magneto-rheological fluid used, which in turn depends on the magnetic field strength. Some parameters of the numerical model are adjusted using cyclic and hybrid testing results on a 15 ton MR damper with internal coils. Numerical and experimental results for the 15 ton MR damper showed very good agreement, which supports the use of the proposed cascade magnetic-fluid model. The construction of the 97 ton MR damper involved several technical challenges, such as the use of a bimetallic cylinder for the external coils to confine the magnetic field within a predefined magnetic circuit. As it should be expected, test results of the manufactured MR damper show that the damping force increases with the applied current intensity. However, a larger discrepancy between the predicted and measured force in the large damper is observed, which is studied and discussed further herein.



## 1. Introduction

In recent years large scale magneto-rheological dampers have reached more attention in the area of structural response control [18]. An example of such an MR damper is a large scale bypass type MR damper with 400 kN force capacity and ±47.5 cm stroke [19], or a large scale annular orifice type MR damper with a force capacity of 300 kN and a stroke of ±28 cm [20]. MR dampers presented in the literature have been designed using relatively simple analytical expressions [1,16,17]. Detailed design and testing aspects have also been thoroughly described in [1], as well as a semi-active seismic clipped optimal control algorithm based on a linear square regulator [15]. Different phenomenological models can be found in the literature, e.g. [5], where a modified Bouc–Wen model has been successfully proposed, and other non-parametric and neural network models have also been studied [6]. A good example of the latter are the ANFIS models used to represent the force–velocity and displacement device behavior of MR dampers [13]. However, all these are black-box models which present parameters that need to be estimated experimentally. Therefore, these models are not capable of capturing relevant design and practical topics such as the confinement of the magnetic flux within the damper coil, among other things.

A mathematical model was presented elsewhere [1] to represent the device force–displacement and force–velocity behavior of an MR damper that includes variability on the damper geometry, viscosity of the fluid without magnetization, and level of shear stress of the magnetized MR fluid. In Ref. [8] mathematical expressions are provided, which consider the friction force effects between the MR fluid and the walls of the device, which also modify the behavior of the damper. Another important reference to this work is a mathematical model of a double-tube MR damper using the well-known annular flow solution and fluid compressibility [9].

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Although all this work has been particularly useful in designing MR dampers, their application to understand certain aspects of their behavior is not straight forward. Examples relevant to our work such as the rise in temperature due to energy dissipation and the consequent rise in internal pressure, or the magnetic flux within the damper, cannot be analyzed by phenomenological or ANFIS models. However, they need to be carefully considered during the design phase of an optimal damper.

Consequently, multi-physical modeling of interacting fields has become relevant in designing MR dampers. With this motivation in mind, this paper deals with the design, manufacturing and testing of a large-capacity MR damper using a FE numerical model describing the magnetic behavior and dynamics of the magneto-rheological (MR) fluid to optimize the design of the damper. Ref. [7] presents a complete system of constitutive equations for an isotropic MR fluid within the framework of the electro-dynamical and thermo-mechanical theories for non-Newtonian incompressible MR fluids. Results of the numerically solved equations are not given in this reference. However, Refs. [4,14] analyze the magnetic circuit of an MR damper for a 2D-axisymmetric model using Ansys, and use these results, together with analytical results of a flow of an MR fluid through a parallel duct, to estimate the force of the damper.

Multi-physics finite-element numerical models (MPFEM) are also becoming increasingly sophisticated in their capacities to model complex interacting fields such as magnetism, fluid dynamics, stress–strain, and heat transfer. The state-of-the-art in FEM modeling enables us to study the cyclic behavior of an MR damper under constant current and sinusoidal excitations, thus helping in understanding the general trends of design of an optimal MR damper prior to considerable iterative experimental work.

Consequently, the specific goals of this research are to design, manufacture and test an optimized large-capacity MR damper with external coils. The steps to achieve these goals are: (i) to numerically predict the device force–displacement and force–velocity constitutive behavior of the device using state-of-the art MPFEM software; (ii) to perform a parametric study of the damper design parameters with different geometries and boundary conditions; (iii) to use these MPFEM tools to manufacture an optimized large-scale MR damper; and (iv) to validate and calibrate the damper design procedure with experimental results on the real scale damper. First, the governing mathematical formulation is presented. Two models were considered in cascade, a magnetic model, responsible for computing the magnetic field produced in the damper due to the current running through the coils, and a fluid-dynamics model, responsible of accounting for the fluid yield stress caused by the magnetic field computed from the MPFEM magnetic model. The latter leads to the estimation needed for the damper force. Second, a model validation is performed using experimental results of a 15 [ton] MR damper presented elsewhere [2,3]. Third, a new real-scale MR damper with external magnetic coils and a nominal capacity of 97 [ton] was designed and optimized by a parametric study using the MPFEM results. Finally, a brief summary of experimental results and model calibration is presented.

## 2. Problem formulation

In this research an MR damper with a cylindrical external orifice and external coils is designed, manufactured, and tested. The design of an MR damper is an iterative process and different topics such as magnetic flux confinement, dynamic range optimization for high capacity, and damper force rise time reduction have to be considered to achieve an optimized design. MPFEM are used in this research as an alternative to simplified design models in order to optimize the design of the damper, thus saving costs and experimental time. Most of the material presented in this section is well known and may be skipped by the informed reader. It is included with the single purpose of making the article self-contained and exposing the physical and mathematical models that will be used herein.

For high shear rates, an MR fluid can be well represented by the Bingham shear stress-shear strain rate constitutive model [6]; models for lower shear rates usually depend on parameters that need to be calibrated with experimental results. The Bingham mathematical representation can be written in terms of the dynamic viscosity as:

$$\eta(\dot{\gamma}, \mathbf{H}) = \frac{\tau_o(\mathbf{H})}{\dot{\gamma}} + \eta_\infty, \quad \tau > \tau_o(\mathbf{H}) \tag{1}$$

where $\mathbf{H}$ corresponds to the magnetic field strength (or strength) in the fluid domain; $\eta(\dot{gamma}, \mathbf{H})$ is the dynamic viscosity of the fluid, $\tau_o(\mathbf{H})$ represents the fluid yield stress dependent on the magnetic field strength; $\dot{gamma}$ is the shear strain rate of the fluid; $\eta_\infty$ corresponds to the inherent dynamic viscosity of the fluid in absence of a magnetic field; and $\tau$ represents the shear stress of the fluid. Please note that for the steel and MR fluid, there is a non-linear functional dependence between $\tau$, the magnetic flux density $\mathbf{B}$ and $\mathbf{H}$ as it will be shown later. A computational adaption of this model is required to implement it in CFD software [e.g., Ansys CFX]. According to the model proposed by Beverly-Tanner (B&T) [10], and taking Eq. (1) as a basis, the fluid may be modeled as an equivalent bilinear Newtonian fluid with a field-dependent dynamic viscosity, i.e.

$$\eta = \begin{cases} \frac{\tau_o(\mathbf{H})}{\dot{\gamma}} + \eta_\infty, & \dot{\gamma} \geqslant \dot{\gamma}_o \\ \alpha\eta_\infty, & \dot{\gamma} < \dot{\gamma}_o \end{cases} \tag{2}$$

where $\alpha$ is a non-dimensional impedance parameter in the pre-yield region of the fluid, and $\dot{\gamma}_o$ corresponds to the critical shear strain rate given by $\dot{\gamma}_o = \tau_o(H)/(\alpha - 1)\eta_\infty$. This alternative is compared schematically with the Bingham model in Fig. 1. The higher the value of $\alpha$, the better the consistency between the two models.

The MR fluid used in this research is the Lord MRF-132 DG and has the nominal properties shown in Table 1. The magnetic properties of the fluid are given by the manufacturer and they are characterized by a magnetization ($B$–$H$) curve, and a yield stress–magnetic field strength relationship [11]. A comparison between these non-linear $B$–$H$ curves for the MR fluid and steel SAE 1045 [12] used to manufacture the damper is plotted in a logarithmic axis in Fig. 2(a). It is apparent that magnetization of steel is larger, faster, and occurs for smaller values of $\mathbf{H}$, reaching a saturated state for lower $\mathbf{H}$ than the fluid. Furthermore, the relationship between the yield stress in the fluid and $\mathbf{H}$ is shown in Fig. 2(b), where the yield stress gets essentially saturated at a value of

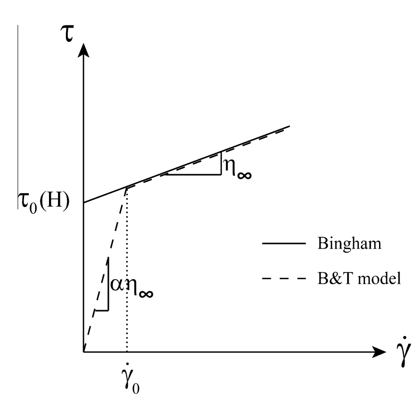


**Fig. 1.** Comparison between the Bingham and Beverly–Tanner model.