Design of a magneto-rheological damper for motorcycle racing

Final thesis report
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**ABSTRACT**

The aim of this project is to show the possibility of magnetorheological controlled damping in motorcycle racing. This study goes with theoretical models, prototypes and tests.

**KEY WORDS**

MRF, damper, motorcycle racing
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Preface and thanks

This report sums up my final thesis on the adaptation of magnetorheological technology to motorcycle racing dampers. It has been four months of work at IFMA, knowledge I learnt in Australia at UOW are also noticable. These last months sums up four years of engineering school and are a condition to my graduation.

This work would not have been possible without the help and advices of many persons. First, I want to thank Régis Dubois, Shaft Racing owner, for being the person to follow me in this adventurous project. I also want to thank Partice Touzet for the help the gave me whenever I needed it and for being comprehensive on the different problems and delays I went through. Finally, thanks to Dr Weihua Li from the University of Wollongong, Australia, since he he introduced me to magnetorheological damping during my research visit. Some of the results used here come directly from his work on magnetorheological fluids. I thank them for giving me a chance to further this very interesting study.

After these three main characters come some professors, industrials, motorcycle riders, mechanics, fans, engineers who gave me advices, help and/or material. They come here in an alphabetical order: Stéphane Ardouin (S’AR Motos), David Baffeleuf (Supermoto rider), Thierry Charbonnier (Dakar rider), Eric Delcamp (Delcamp Racing), Jeremie Garbe (Lord corp.), Christophe Gouinaud (ISIMA), James Gunn (Zeno Damper), Oliver Kennedy (UOW), Stéphane Mézard (AZ Motos), Arthur Mota (Flying Tractors), Claude Rouelle (Optimum K), Pascal Schoofs (WDS air suspension), Jean-Jo Willé (WDS air suspension). I also want to thank all the UOW SAE team who I started this project with back in Australia.

Finally I want to thank all the person whom I shared this project and six last months with. I mostly think of my roommates Mathilde Ortynski, Paul Boué, Guillaume Terrenoir and Gaétan Prod’Homme who provided me a nice work atmoshpere.

THANKS YOU EVERYBODY!
Introduction

While cars manufacturers (Audi, Cadillac, Holden,...) start to use the magnetorheological technology for their prestige models’ dampers, it seemed interesting to introduce it to motorcycle suspensions, and especially in racing. The magnetorheological fluid are still marginal in damping but the improvement of their properties and applications could bring good results, especially in Off-Road racing where the track takes riders on numerous conditions of adherence.

A scientific description of the magnetorheological fluids is given in chapter one. This introduction gives some basis on these fluids and their model.

The chapter two describes the different steps the study went through for the design of the first prototype from calculation to fabrication. It also gives some advices for the next evolution.

The last chapter sums up the different tests done on the prototype. A quick introduction to damper testing is also provided.
Chapter 1

Magneto-rheological fluid theory

This chapter briefly sums up the theory of the magneto-rheological fluids. The resultant equations will be used as a source for the comparison between theoretical and practical results.

Jacob Rabinow’s first demonstrations suspending a 117 pound woman from a swing using MR fluid, the fluid’s yield strength needed to be over 100 kPa to suspend the woman. This experiment proved the strength and capabilities of this fluid.

Sources:

"The shock absorber handbook (second edition)" John C. Dixon

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1.1 Introduction to magnetorheological fluids

A magnetorheological (MR) fluid is a type of intelligent fluid which consists in a suspension of micrometer-sized iron particles in a carrier fluid. MR fluids change by the application of a magnetic field across the fluid. The induced field causes alignment of the iron particles in column structures parallel to the field lines. The structure resists the fluid flow & causing a change in fluid apparent viscosity to the point of becoming a viscoelastic solid.

The viscosity of the carrier fluid does not really change, but the shear strain rate and yield stress of the fluid increase with an increasing field. The ferrous particles that form each of the chains resist movement out of their flux lines. MR fluids develop a yield stress as it thickens, the resistance of the ferrous dipole chains to move result in a restriction of the fluid to flow. This behavior can be compared to a Bingham plastic with variable yield strength.

History of electrorheological and magnetorheological fluids.

Small electrorheological (ER) effects have long been known, but large-scale effects with possible practical applications were first studied by Winslow (1947). The first important work on MR is attributed to Rabinow (1951). For forty years significant efforts were made on ER, largely neglecting MR, but since 1990, when work on MR increased, it has become apparent that MR may be much more practical because of the lower operating voltage, lower power requirement, higher shear yield stresses achievable, broader operating temperature range, and greater tolerance of the liquid to contamination, particularly water. Against this must be weighed much greater expense, and some significant hazards in manufacture. Winslow’s 1947 patent, with the mysterious title ‘Method and Means for Translating Electrical Impulses into Mechanical Force’ is really a patent on various configurations of clutches and relays using ‘electro-fluids’, although the 22nd and last claim is very general:

The method of instantaneously increasing the viscosity of a force-transmitting fluid composed of a dielectric liquid and a finely divided substance suspended therein; which consists in applying an electric field to the fluid.

This would surely have included ‘electrofluid’ motor vehicle dampers if such an application could have been conceived at the time. Figure 1.1, shows Rabinow proposition for an MR controlled clutch. The possibility of a controllable rotary damper is evident to modern eyes.

The specific patent claims included:

5. A device for controlling the transmission of torque from one element to an adjacent relatively movable element comprising members fixed to said elements respectively and having substantial, opposed, closely spaced surface areas separated by a fixed distance, a mass of contiguous relatively movable discrete paramagnetic particles in the space between the said surface areas and means for creating a magnetic field between said opposed members and including said particles, to produce a coupling effect between said spaced members ....

7. The invention according to claim 6 in which said particles are mixed with a non-magnetic fluid. 8. The invention according to claim 7 in which said particles consist of iron powder and said fluid is a light oil.

Here it may be seen that Rabinow emphasises the low operating voltage, large force generation and rapid response. He made public demonstrations with a human supported by an MR link at a stress of 100 kPa, using 90% by mass iron and 10% mass light mineral oil.
1.2 Bingham flow

A Bingham plastic or liquid is a material with a yield stress that subsequent to yield behaves as a liquid with a viscosity. This is of interest to damper analysis because electrorheological and magnetorheological fluids do this. Their yield stress depends on the electrostatic or magnetic field respectively.

Considering the flow of a Bingham material through a circular pipe, for a small pressure there will have no flow. The pressure must be sufficient to cause yielding at some point. Considering a circular rod element along the pipe, the driving force is proportional to the pressure and the circular end area, whereas the shear resistance is proportional to the yield stress and the circumference. Therefore, the yielding will first occur at the largest radius. At higher driving pressures, there will be a unyielded central core flowing as a solid slug, with an annular shearing region around it.

1.3 Liquid-solid suspension

A finely divided solid powder within a carrier fluid is called a suspension. These arise in ER (electrorheological) and MR (magnetorheological) liquids for controllable dampers. Typically they use aluminium silicate and soft iron particles respectively, in a low viscosity oil, the suspension has the general property

$$\rho = \frac{m}{V}$$  \hspace{2cm} (1.1)

Using subscripts for $S$, $L$ and $M$ for solid, liquid and mixture

$$\frac{m_S}{m_L} = \frac{\rho_S V_S}{\rho_L V_L}$$  \hspace{2cm} (1.2)

$$V_M = V_S + V_L$$  \hspace{2cm} (1.3)

$$m_M = m_S + m_L$$  \hspace{2cm} (1.4)
\[ \rho_M = \frac{m_S + m_L}{V_M} = \frac{\rho_S V_S + \rho_L V_L}{V_M} \quad (1.5) \]

As a matter of interest, the volume summation is not always very accurate when two liquids are mixed because of packing of the two shapes of molecules, and the small discrepancy may be of practical significance in some cases, particularly when attempting to deduce the mixture ratio from a posteriori measurement of the mixture density. However, it is sufficiently accurate in the present context. Note that the solid phase to be used in the above equations is the material density, not the bulk powder density which is significantly lower because of the imperfect packing of the individual fragments in bulk.

It’s not quite so straightforward to determine the volume or mass ratio of constituents when the volume and density of the mixture are known. Then the two convenient and relevant equations are conservation of volume and mass through the mixing process:

\[ V_M = V_S + V_L \quad (1.6) \]
\[ \rho_M V_M = \rho_S V_S + \rho_L V_L \quad (1.7) \]

These must be solved simultaneously, giving

\[ \frac{V_S}{V_M} = \frac{\rho_M - \rho_L}{\rho_S - \rho_L} \quad (1.8) \]
\[ \frac{V_L}{V_M} = \frac{\rho_S - \rho_M}{\rho_S - \rho_L} \quad (1.9) \]

The calculation is not really well conditioned when the densities are similar. It is useful to premix known ratios of the constituents and to measure and plot the resulting density. For a mean particle volume \( V_P \) and mean spacing particle centres \( X \), the cubical cell volume is \( X^3 \). The mixture density is then

\[ \rho_M = \frac{\rho_L + ((\rho_S - \rho_L)) V_P}{X^3} \quad (1.10) \]

For a spherical particle diameter, or a mean effective diameter, \( D \), then the mixture density is

\[ \rho_M = \rho_L + \frac{\pi}{6} (\rho_S - \rho_L) \left( \frac{D}{X} \right)^3 \quad (1.11) \]

The mean particle spacing \( X \) may therefore be estimated from

\[ \frac{X}{D} = \left( \frac{\pi}{6} (\rho_S - \rho_L) \right)^{1/3} \left( \frac{\rho_M - \rho_L}{\rho_M - \rho_L} \right) \quad (1.12) \]

The mixture compressibility \( \beta_M \) combines the component properties by volume (to a good approximation), so

\[ \beta_M = \frac{(V_S \beta_S + V_L \beta_L)}{V_M} \quad (1.13) \]

The solid compressibilities are insignificant in relation to that of the oil, so in effect

\[ \beta_M = \frac{V_L \beta_L}{V_M} \quad (1.14) \]
Table 1.1: MR liquid density

<table>
<thead>
<tr>
<th></th>
<th>Solid</th>
<th>Oil</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>7874</td>
<td>800</td>
<td>2920</td>
</tr>
<tr>
<td>Volume</td>
<td>40</td>
<td>60</td>
<td>100</td>
</tr>
<tr>
<td>Mass</td>
<td>104</td>
<td>48</td>
<td>152</td>
</tr>
</tbody>
</table>

Table 1.2: MR general material properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Units</th>
<th>Oil</th>
<th>Iron</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density</td>
<td>$\rho$ kg/m$^3$</td>
<td>800</td>
<td>7874</td>
</tr>
<tr>
<td>Compressibility</td>
<td>$\beta$ %/MPa</td>
<td>0.04</td>
<td>–</td>
</tr>
<tr>
<td>Volume expansion</td>
<td>$\alpha$ ppm/K</td>
<td>1000</td>
<td>36</td>
</tr>
<tr>
<td>Specific heat</td>
<td>$c_p$ J/kgK</td>
<td>2500</td>
<td>450</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$k$ W/mK</td>
<td>0.14</td>
<td>81.0</td>
</tr>
<tr>
<td>Viscosity</td>
<td>$\mu$ mPas</td>
<td>40</td>
<td>–</td>
</tr>
</tbody>
</table>

The volumetric thermal expansion properties of the mixture are also proportional to the volumetric constitution, so using the volumetric expansion properties of the solid and liquid:

$$\alpha_M = \frac{(V_S\alpha_S + V_L\alpha_L)}{V_M} \tag{1.15}$$

The specific thermal capacity (specific heat) properties of the mixture are obtained in the mass ratio of the constituents, so

$$c_{PM} = \frac{(m_Sc_pS + m_Lc_pL)}{m_M} \tag{1.16}$$

The thermal conductivity of a suspension is not easy to analyse, and would be obtained experimentally. In practice, a coarse estimate may be made according to the conductivities of the solid and liquid, but the material in non-homogeneous. In the case of ER and MR liquids, with the field on the properties are non-isotropic.

### 1.4 MR fluids

Practical MR liquids have a high proportion of dense solids in small particles (a few micrometres) carried in a lower-density liquid, giving a very high density mixture, as high as 4 g/cm$^3$. Example values are shown in Table 1.4. The component properties are given in Table 1.4.

The solid phase, being denser, is subject to some settlement and separation over time, particularly when there are larger particles with a large density difference. This could be a problem for MR liquids, but with appropriate additives the problem is not severe, and remixing occurs very rapidly.

For practical numbers in an MR liquid, at a solid mass content of 80%, the solid volume content is 40.64%. The mixture density is 2844 kg/m$^3$, so the spacing ratio $X/D$ is 1.22. This illustrates the close spacing of the particles in practical MR liquids.

The compressibility of pure light mineral oil is very small at about 0.04 %/MPa, so pure MR liquids are about 0.02-0.03 %/MPa. This is insignificant unless increased by gas
A light mineral oil has a high volumetric expansion coefficient of 1000 ppm/K (parts per million per kelvin), iron is relatively low at 36 ppm/K (three times the linear value to make it volumetric). The solids therefore tend to reduce the expansion of the mixture. Nevertheless, the expansion may be important. For the pure oil between -40 and 130 °C the volume change is about 17%.

A typical light mineral oil has a specific thermal capacity \( c_p \) of 2500 J/kg K, pure iron is about 450 J/kg K. The solids therefore reduce the thermal capacity significantly. Volumetrically, the oil specific thermal capacity is 2.0 J/cm³ K, iron is 3.51 J/cm³ K, so the mixture thermal capacity variation by volume is relatively small, and increases with solid content.

For a typical mineral oil, the thermal conductivity is 0.14 W/m K, for pure iron it is 81 W/m K. The solids will therefore increase the conductivity, particularly in the case of iron, although the solid content and greater viscosity will reduce thermal currents and consequent free convection where this is significant. With the magnetic field on, the particles are organised into the fibrils along the field, so the material becomes isotropic with, thermal and electrical conductivities being dependent on direction.

The viscosity even at zero field is difficult to analyse from first principles. The solid content will increase the effective viscosity compared with the pure oil, but this is hard to quantify. Obviously the solid material internal properties are not relevant, but the volume present and the particle size distribution are important. The particle geometry, smooth sphere or irregular and jagged, may also be significant.

The performance of MR materials has been characterised by the quality factor

\[
Q = \frac{\mu_0}{\tau^2 y}
\]

which is the zero-field viscosity divided by the square of the field-on yield stress. An MR device can operate with a reduced volume of fluid. For example, a car damper could operate at the required forces with an active liquid volume of 0.3-0.5 cm³, much less than the usual passive-damper 70 cm³ of oil. However, long life of the MR liquid is really the limiting factor (damage associated with total energy dissipated per liquid volume), so ample total liquid is still required, even though the field-controlled volume is small at any one time.

Costs are always important in manufacturing. MR liquids are expensive. Current price runs at 100 to 200 $/litre for bulk purchases. For a 70 cm³ damper this is 7-14 $/damper, 28-56 $/vehicle material cost. This is a considerable additional cost in relation to the initial cost of a conventional damper.

**Colloidal Ferrofluids.**

MR fluids should be distinguished from so-called ferrofluids, which are colloidal, having very small particles, typically magnetite (Fe₃O₄) or manganese-zinc ferrite. The particle size is 5-10 nm, 1/1000 of the diameter of MR liquid particles, 10-9 times the individual particle mass. On the application of a magnetic field, ferrofluids remain liquid, they do not form particle fibrils, and so do not generate a yield stress. The ferrofluid experiences a body magnetic force which attracts it to regions of high field strength.
Chapter 2

Design process of the prototype.

This chapter describes how the design and the fabrication of the prototype went on.

This figure shows how works a damper (shock absorber) using exclusively the magnetorheological properties of the fluid. On the left side we notice two different states of the fluid (random and columns arrangements) and on the right a design for the damper valve to apply efficiently a magnetic field.

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2. Design process of the prototype.

2.1 Shock absorbers.

2.1.1 Current damping technologies.

Practical dampers are based on dissipating energy primarily by turbulences, usually by allowing the liquid to pass through a small hole, giving a turbulent exit jet which dissipates in a bulk of liquid.

The dynamic loss kind of valve, where the pressure loss for a given volumetric flow is more dependent upon the fluid density than its viscosity, introduces a new problem. The pressure loss is now dependent upon the square of the exit velocity. This means that for a simple orifice or fixed area the pressure loss depends on the square of the volumetric flowrate, which will give a damper force proportional to damper velocity squared. This is completely unacceptable. However, unlike to problem of viscosity variation with temperature, this has a entirely practical solution — the valve area can be made to vary to produce the desider characteristic.

All dampers have this area variation in a passive form, with a larger pressure difference forcing the valve to open to a larger area, giving a moderate fluid exit speed.

Valves can be made to be responsive to many factors. The obvious ones are: (1) position, (2) velocity, (3) acceleration. The damper is essentially a device for dissipating energy and as such the velocity sensitivity is the basic one. All other valve sensitivities are just variations of the basic theme.

Basic mechanical damper valves may be classified conveniently by the design configuration used in variability of area. There are numerous possible forms, but the four basic ones are:

1. disc valve (a)
2. rod valve (b)
3. spool valve (c)
4. shim valve (d)

These are illustrated in figure 2.1.

![Figure 2.1: Basic valve types.](image-url)
2.1.1.1 Disc valves.

The simple disc with coil spring, Figure 2.1 (a), is obvious in operation, being sealed until an opening pressure difference is reached, depending upon the active pressure area and the preload force on the coil spring. The active pressure area, that is the effective area of the disc within the seal, may be much larger than the cross-sectional area of the flow passage. The flow area, is the circumference of the seal times the lift. If the coil spring is of low stiffness (but not necessarily of low preload force) then further opening can happen easily, and large flow areas and flow rates will be possible with little increase of the pressure difference. Alternatively, a stiff spring with low preload will give a more gradual increase of the area. In practice, because the flow area is just a few square millimeters, even a small lift of this kind of valve tends to give a “large” area, so it is difficult to make this type truly progressive. Rather, it acts like a simple blow-off valve with a constant pressure characteristic. Nevertheless, this type may be suitable in some case, especially for a low-preload valve to constrain flow to one direction only with quite low forward pressure drop, as in a foot valve. Unless the disc is guided, it will probably open asymmetrically because of imperfect symmetry of the spring.

2.1.1.2 Rod valves.

To obtain a progressive pressure drop with a disc type of valve, it requires a small circumference to allow a worthwhile lift, leading to a rod valve, in which the fluid is controlled by a flat rod end on a hole, Figure 2.1 (b). The hole may be as small as convenient manufacturing permits. This allows a relatively large progressive valve lift, but a lift of \( h = \frac{\pi R^2}{2\pi R} = \frac{R}{2} \) makes the exit area equal to the hole area, so there is a limit to the actual flow area. Also, the spring has to be designed to give the desired force increase over this distance, so the stiffness will still be high.

To allow a greater lift for a given flow area, a tapered rod can be used. When the taper is long, this is known as a needle valve. These are nowadays mainly used to allow a quick rebound hydraulic setup change. A straight taper is easier to manufacture, but more elaborate tapers are possible.

2.1.1.3 Spool valve.

The spool valve or bobin valve, as shown in Figure 2.1 (c), has automatic radial positioning, usually opening against a coil spring although a leaf spring would also be a possibility. A simple form of spool valve cross-drilled with four round exit holes is a common choice for extension foot valve. The exit hole(s) in the spool may be given a suitable profile to provide a flow area varying with spool position in any required way. The valve spool is particularly adaptable in this aspect. \( Q \) and \( P \) being respectively the flow rate and the pressure, a linear \( Q = f(P) \) characteristic is obtained with a flow slot that reduces as the square root of the opening length. In principle any required characteristic may be designed in, limited by the practical precision of the exit hole profile and the manufacture costs. An accurately shaped hole could be made by broaching. With a single sided exit, the spool will react laterally to the fluid exit momentum, creating friction, a possible hysteresis and inconsistency in the spool position for a given pressure difference. This may be prevented at extra costs, by using of two or more symmetrically positioned exit holes.

A piston spool needs to be compact to fit between the rod and the piston outer diameter. Four may be used, two acting each way.
2.1.1.4 Shim valve.

The fourth main type of valve is the shim with basic principle as shown in Figure 2.1 (d). In practice a pack of shim is used with varying diameters, a system particularly common on racing dampers, partially because the characteristic can be changed easily, Figure 2.2. Shim valves are nowadays used on most motorcycles. On passenger cars the shim valve shows to advantage because it is relatively easy to set up accurately with consistent results. This because the flat shim sit naturally on the piston without problems of of manufacturing dimensional inconsistencies affecting the preload.

Figure 2.2 shows the usual configuration of one pack on each side of the piston, which will typically contain six holes, three for fluid motion in each direction. The valve opening height is only a fraction of a millimeter, so the flow path is roughly two-dimensional.

Figure 2.2: Complete double-acting shim valve as used in practice

2.1.2 Magnetorheological damping technology.

The design of an MR damper can be more conventional, since the apparent viscosity of the fluid can be changed thanks to a magnetic field, although the piston size is increased somewhat from a normal damper.

Figure 2.3: A basic design for a single-tube MR damper.

The magnetic field is generated by the axial coil, for which connecting leads are conveniently brought out through the rod. The product of the current $I$ and the number of turns $N_T$ (all of the same handedness) is $M = N_T I$, the magnetomotive 'force' or magnetic potential. The field strength that this will produce in the liquid depends on the entire magnetic circuit. This should have a low reluctance, so soft iron or steel is used for the piston. For best results a low-carbon steel with high permeability and high flux saturation level is desirable. For an air-gap electromagnet, the gap has high resistance relative to the iron (the material permeabilities being in a ratio...
2. Design process of the prototype. 2.1. Shock absorbers. 

For an MR liquid, the permeability may be quite high (it may be 80 \% iron), and a careful magnetic circuit design is necessary. At high flux density, the iron may saturate, and be a limiting factor, so the cross-section of the iron must be adequate all around the magnetic circuit. The total flux in the circuit is the same at all sections around the circuit, so the critical point of the iron is the section of least cross-sectional area. The magnetic circuit of the example design is axisymmetric, Figure 2.4. The flux passes through the coil axially, expands radially outward through the disc at one end, through one MR fluid gap, back along the iron sleeve, radially inward through the disc at the other end and back into the core completing the circuit. Design of a magnetic circuit is more complex than for an electric current circuit because of the nonlinear behaviour of the materials, and nowadays is likely to be done using a suitable software package (Ansys for example). However, a linear model analysis, as will be done here, is useful for preliminary design and gives useful understanding and an appreciation of the units and numerical values.

A magnetic circuit is analogous to an electric circuit. In the latter case, an electric potential around the circuit (from, e.g., a battery) measured in volts (V) produces a current in amps (A). According to the local cross-section of the wire, there is a local current density (A/m²). Locally, the current density is related to the electric field strength in V/m (volts per metre) by the electrical conductivity (A/V m). In most circuits the analysis is greatly simplified by using lumped parameters such as a resistor with a specified resistance, voltage and current, rather than dealing with current densities and fields.

The magnetic circuit has a magnetic potential (e.g. from a coil with a current) measured in amp-turns (A). The result is a magnetic flux around the circuit measured in webers (Wb). According to the local cross-sectional area of the circuit there is a local magnetic flux density measured in tesla (T, which is just Wb/m²). The local flux density is related to the local magnetic field strength (A/m) by the permeability (Wb/A m, the same as henry per metre, Hlm, 1 H = 1 Wb/A). The magnetic circuit is most easily understood in terms of its physical parts, each of which has a magnetic reluctance, analogous to electrical resistance. Reluctances add in series and parallel in the same way as do resistances.

The permeability of vacuum is, by definition,

$$\mu_{MO} = 4\pi 10^{-7} H/m$$  \hspace{1cm} (2.1)

For air, the value is effectively the same. Ferromagnetic materials have a high value, possibly several thousand times as large. The ratio of the absolute permeability to the permeability of a vacuum is called the relative permeability. The relative permeability of MR fluids is typically around 5 or 6. The high iron content can be imagined to almost short out the reluctance, which depends mainly on the oil gaps between the iron spheres, about 1/6 of the total distance.
Figure 2.5(a) shows a curve of flux density against magnetic field strength $B(H)$ for a steel such as might be used in an MR damper piston. Here, the considerable nonlinearity is apparent. Also, there is saturation. There is a flux density limit, so a field beyond $2$ kA/m has negligible effect. This fairly realistic curve has been generated by the expression

$$B = B_1 \left(1 - e^{-H/H_1}\right)^p$$  \hspace{1cm} (2.2)

where $B$ is the asymptotic value, $H_1$ scales the magnetic field axis, and the index $p$ shapes the part near to the origin.

Figure 2.5(b) shows a simple linear model, in which the consequent flux is proportional to the field up to a sudden saturation. The MR fluid itself is characterised magnetically by a fluid shear yield stress related to the magnetic field or flux density.

MR fluids are quite similar to electrorheological fluids and ferrofluids because all three fluids are a non-colloidal suspension of polarizable particles. A MR fluid has 20 to 50 times the strength of an ER fluid.

MR fluids can be used in three principal modes of operation:

1. pressure driven-flow (valve) mode
2. direct-shear mode
3. squeeze-film mode

2.1.2.1 Pressure driven valve.

Pressure driven-flow mode or valve mode, has two fixed magnetic pole plates through which pressurized MR fluid flows, Figure 2.6. When applying a magnetic field, the iron particles align parallel to the applied field lines and resist the flow of the pressurized MR fluid. The name ‘valve mode’ is used because this model is similar to the operation of a normal valve. Every time you increase the field intensity, the flow resistance increases. The pressure driven-flow mode is the most common mode used in MR dampers, but can be useful for other applications in which a variable flow resistance is required.
2. Design process of the prototype.

2.1.2.2 Direct shear mode.

Direct-shear mode has the two magnetic pole plates move relative to each other, shearing the fluid between them (Fig. 2.7). An applied magnetic field aligns iron particles perpendicular to the pole plates, while the shearing motion tries to bend the particle iron chains along the flux lines. As the field intensity increases, the MR fluid’s resistance to shearing increases. The direct-shear mode can be used in low force devices, and has also found uses in magnetic brakes and clutches applications, Figure 1.1, page 10.

2.1.2.3 Squeeze film mode.

Squeeze-film mode works by squeezing the two magnetic pole plates together on a thin film of MR fluid, Figure 2.8. The application of force on the plates parallel to the direction of flux lines compresses the chain structures evolutions. The intensity of the induced magnetic field determines the ability of the MR fluid particle columns to resist buckling.

2.1.2.4 Comparison of MR Fluid modes.

2.2 First prototype design and evolutions.

2.2.1 Australian prototype.

2.2.1.1 Base of work.

In order to make a cheap but valuable prototype, We selected an external gas reservoir piggyback that we found in the OEM market. The idea is to adapt and modify it to introduce a MR valve instead of the actual shim valve and connect it to the control system.
2. Design process of the prototype.

2.2. First prototype design and evolutions.

After an extended search of different brands, a shock absorber for pitbikes model MV-DNM-COMP seems to of a good quality to price ratio. Its design fits the needs since only the case of the OEM shock is used. The idea is to disassemble the shock and rebuild it with the valve for the MR fluid. By the way the rebound and compression tuning sets are removed. Since it is the base of work, the first step is to model the parts that will be reused on the prototype with care and precision.

2.2.1.2 Design of the MR valve.

One of the requirements for the damper to be used in racing is to have tuning possibilities such as rebound force and compression force. Actual technolgies allow to do some manual adjustment to the suspension before the race to adapt the suspension to the road conditions, weather, velocities, etc.

The MR valve inside the shock provides automatical adjustment of rebound and compression damping characteristics.

**Operating mode of the valve.**

There are different types of valve configuration for a MR damper but the concentric-cylinder configuration is founded in many MR devices. This kind of valve consists in a core, a flux return and an annulus through which MR fluid flows, Figure 2.9. This configuration gets rid of all moving parts like the ones used in classical damping, reducing friction and also hysteresis.

The bobbin shaft is wound with insulated wire. A current is applied through the wire coil around the bobbin and creates a magnetic field in the gap between the core and the flux.

---

**Table 2.1: Comparison of MR fluid valve modes**

<table>
<thead>
<tr>
<th>Property</th>
<th>Force</th>
<th>Best relative movement</th>
<th>Best application</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure driven flow</td>
<td>Medium - High</td>
<td>Large and linear motion</td>
<td>Dampers, servovalves</td>
</tr>
<tr>
<td>Direct shear mode</td>
<td>Low</td>
<td>Rotational and linear movement</td>
<td>Dampers, brakes, clutches</td>
</tr>
<tr>
<td>Squeeze film mode</td>
<td>High</td>
<td>Low motion</td>
<td>Dampers, vibration control</td>
</tr>
</tbody>
</table>

---

![Figure 2.8: Squeeze-film mode.](image)
return. This magnetic field increases the yield stress of the MR fluid and reduces the velocity of it by blocking the fluid, reducing the volume flow rate, and increasing the pressure difference, $\Delta P$.

![MR valve scheme](image)

Figure 2.9: MR valve scheme.

The valve performance analysis can be approximated with the analysis of the pressure-driven mode valve containing MR fluid, which follows the Bingham-plastic flow model.

The design and manufacturing of MR valve based on trial-and-error method is very time consuming and not an accurate method but allow to have a prototype in less time to prove the capacities of such an application. Although, the design of this first valve follows the requirements of Dr Li’s article on the subject. This approach considers the coupling between magnetic field formation mechanism and MR effect formation mechanism.

**Magnetic Circuit of the MR Valve.**

A magnetic flux is necessary to induce changes in the viscosity of the MR fluid. The magnetic field applied to the MR fluid must be predicted by analyzing the magnetic circuit. The MR valve magnetic circuit is similar to the C-shaped electromagnet that is composed of two parts: the steel path with length $L$ and the fluid gap $g$, Figure 2.10.

In the steady-state case, Ampere’s law is given by:

$$\int H dl = NI$$  \hspace{1cm} (2.3)

Where $N$ is the number of turns on the winding, $I$ is the coil current, $H$ is magnetic field, and $l$ is length of magnetic path. Ampere’s law states that the line integral of the field $H$ around a closed path is equal to the number of ampere-turns.

$$NI = H_L L + H_g g$$  \hspace{1cm} (2.4)

Where $H_L$ and $H_g$ are magnetic fields at the steel path and the fluid gap, respectively. The magnetic flux density $B$ is directly dependent on the magnetic field strength $H$.

$$B = \mu_0 \mu_r H$$  \hspace{1cm} (2.5)

Where the constant $\mu$ the vacuum permeability, and $\mu_r$ is the relative permeability of flux path materials that is the steel path.

---

1. Finite Element Analysis and Simulation Evaluation of a Magnetorheological Valve
Assuming no magnetic flux leakage, the magnetic flux density in the steel path, $B_L$, and that in the fluid gap, $B_g$, has the relationship:

$$\Theta = B_L A_S = B_g A_f$$

(2.6)

Where $\Theta$ is the magnetic flux, $A_S$ and $A_f$ are the cross-sectional areas of the steel path and fluid gap, respectively. If $A_S = A_f$ then $B_L = B_g$ and:

$$B = \frac{\mu_0 NI}{(\frac{1}{\mu_l} + \frac{2}{\mu_g})}$$

(2.7)

if $\mu_L$ is really big compared to $\mu_g$ then, the magnetic flux density in the fluid gap can be approximated as:

$$B_g \approx \frac{\mu_0 \mu_L NI}{g}$$

(2.8)

Finally, the magnetic field in the fluid gap is given by:

$$H_g = \frac{NI}{g}$$

(2.9)

The flux density cannot increase infinitely when the applied magnetic field is above the operating point or the saturation point. Magnetic saturation of steel and the MR fluid used should be considered.

The type of MR Fluid used in the shock absorber is the MRF-132DG from LORD Corporation®. The typical properties of the fluid are given in Table 2.2.1.2.

We also know the curve of the yield stress vs. the magnetic field.

**Valve Characteristics.**

The MR valve can be analyzed as a 2-D axisymmetric model due to structural symmetry, Figure 2.4. The dimensions of the valve are:

- $D_{core}$ = diameter of bobbin shaft
- $L_{core}$ = length of bobbin shaft
- $D_{in}$ = inner diameter of the valve, $2R_1$
2. Design process of the prototype. 2.2. First prototype design and evolutions.

Table 2.2: MR fluid properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Appearance</td>
<td>Dark Gray Liquid</td>
</tr>
<tr>
<td>Viscosity @ 40°C</td>
<td>0.092 ± 0.015 Pa</td>
</tr>
<tr>
<td>Density</td>
<td>2.98 - 3.18 g/cm³</td>
</tr>
<tr>
<td>Solids content by weight</td>
<td>80.98%</td>
</tr>
<tr>
<td>Flash pointS</td>
<td>&gt; 150°C</td>
</tr>
<tr>
<td>Operating temperature</td>
<td>- 40 to + 130°C</td>
</tr>
</tbody>
</table>

\( D_{out} \) = outer diameter of the valve, \( 2R_2 \)

\( L_{active} \) = active core length

\( g \) = fluid gap

\( L_{return} \) = thickness of flux return = \( [(D_{out} - D_{in})/2] - g \)

The proposed dimensions of the valve, considering the different aspects like saturation are given by Dr Li’s article.

2.2.1.3 Integration to the existing case.

Now that all the basic features of the valve and command system are theoretically improved, it needs to fit inside the case, to be easily machinable and efficient. Even if the aim was not...
to get a full working prototype, it still needs to be as good as possible it gives good results and proves technology. Being the first prototype, it also has to be reliable to perform as many tests as possible on it to get better improvement with the next step. The first thing to deal with were to adapt the internal and external diameter of the valve to the existing case and rod. Luckily, the measures were close enough not to have to perform another ANSYS analyse.

Referring to our sketch, Figure 2.11 , we need an inner piston where are mounted the coil and a sleeve to protect it from fluid aggressions. We also need an outer piston doing a return path for the magnetic field and seal mounted to prevent from leakages. The main thing to care about is to get a proper magnetic field in the gap where the fluid has to change its properties. That means getting a link between the internal piston and the return path that is strong enough to support damping efforts but not creating a path for the magnetic field to go by.

The first thing is to find the materials to use for these features. As it is said in subsection 2.2.1, we need a low reluctance for the core and the return path, so steel is used. On the other hand, for the link, we use aluminium and nylon for their high reluctance. These materials are common, quite cheap and easy to machine. Let’s detail the design of the principle parts:

**INNER PISTON.**

The inner piston has many features. It links the valve to the rod, holds the cupper bobin and creates a path for the magnetic field. It is made out of steel in two parts. The first one holds the bobin and the second is the top cover. The valve is positioned on the rod axially with the mount and bolt from the aftermarket shock absorber. The radial position is not important in this application.

**OUTTER PISTON.**

The outer piston’s main features are to guide the rod and to seal the two chambers preventing MR fluids to go around and hurt the surface. The cross section has to be homogeneous not to suffer from a magnetic flux saturation that would also decrease the valve capacities.
For the same reason, it’s built out of steel too.

**Linkage.**
In order to be efficient, the linkage has to locate and fix the inner and outer pistons. The easiest way to do it was to put some aluminium pins in the same axis as the rod and to hold them still with nylon screws at each ends. For this size of piston (36mm) and the low forces generated, five of them are strong enough. This materials being non magnetic, the whole field will go by the gap from steel to steel. This should provide an even distribution of the magnetic field in the valve, improving the results theoretically.

**Electromagnetic Bobin.**
The bobin is wrapped on a silicon sealant to prevent shorcut from the wire to the inner piston and to prevent the fluid from sticking on the bobin. To improve the cupper mass in the bobin, 0.4mm cupper wire is used, maximazing the number of turns $N_T$ and the strengh of the magnetic field.

2.2.2 Evolutions
Back in France, a new evolution occured on the first prototype after a failure. The five nylon screws snapped because of traction and fluid adression. There are brass screws instead on the new evolution. The main problem of the first prototype was its steel case. It creates a second return path for the magnetic flux and tends to stick iron particles from the fluid between the outer piston seal and the case. This obviously creates huge friction and hystereris. The new case is machined from plain aluminium, reducing the weight but principally preventing the iron particles to grip the who system.

2.3 Design of the new prototype
In order to get further into testing and improving the technology, a new prototype is allready drafted. This one also uses a classical shock absorber from the motorcycle market. It is a WP pds rear suspension set usually mounted on KTM off-road bikes. This suspension set works without linkage and has two pistons, one for classical damper velocity sensitivity and another one based on position sensitivity to prevent from topping out the damper.

After a quick look at motorcycle and car racing regulation, it becomes obvious that a damper using only the properties of the fluid for damping is unacceptable. Hence, electronic loop systems having a value watched and compared to a command is forbidden. It suppresses the possibility to use the first valve design since its control would be done thanks to a linear potentiometre and a damping mapping function of the rod relative velocity to the case. Having a mechanical free adaptative damping also means that in case of an electronic failure, the damping on the vehicle goes straight to unacceptable and unsafe.

2.3.1 Valve
A BRAND NEW DESIGN.
In order to keep MR fluids properties for slight adaptation of the vehicle handling in specific conditions, the new valve will use both mechanical and magnetorheological features. The subsection 2.1.1 gives a good understanding of how to generate an evolutive mechanical damping. The choice would normally go straight to shim kind of valve since they are easy to set up again and again which make them really interesting for testing and forward thinking, racing. The matter is that being so thin (0.05-0.2mm thick), a small magnetic
field would saturate the shims and the efficiency of the magnetorheological feature would be that reduced. Rods valves and disc valves being hard to use for a main hydraulic damping, the new valve is designed with a spool configuration that should let more mechanical and magnetorheological coherence. Also, after a quick research, nothing like a combination of spool valve and MR valve seems to be patented yet, this open the door for further studies.

### Spool Valve Calculation

A spool valve is a basic damping system but it’s possible to adjust it in a good measure thanks to complex exit profiles in combination with the good spring. This valve prototype will be constituted by two sets of spool valve on for each motion way, that also allows to keep some tunability of the rebound and compression damping curves by using different spring rates. Obviously both spring sets and exit profiles have to be designed at the same time since the displacement of the spool limited by the spring force will open or close the exit orifice. As seen in subsection 2.1.1.3:

\[
Q = f(P)
\]

\(Q\) and \(P\) being respectively the flow rate and the pressure, a linear \(Q = f(P)\) characteristic is obtained with a flow slot that reduces as the square root of the opening length

#### 2.3.2 Control

The command part of the project is slightly reduced now since regulations do not allow electronic loop systems. Nevertheless, the bi-spool valve configuration allows us to have two different bobbins, one for the compression flow and the other for the rebound flow. That means that thanks to the magnetorheological feature of the valve, it will be possible to adjust the vehicle behaviour by reducing or rising the damping coefficients on the two directions independently. This opens tons of possibilities for these suspensions in mechanical races, especially Off-Roads.

#### Supermoto.

Supermoto being the motorcycle race that ask for adaptation and efficiency for the rider as for the bike, having a possibility to change the damping characteristic from the asphalt to the dirt could be a great advantage. Indeed, the tracks are made of asphalit on 60% and the other 40% tends to be a reduced supercross track, that means that mechanics adapt the suspensions to work well in both parts. Having a good set for the dirt and another good one for the asphalt could be the fastest way to give magnetorheological dampers their first race results.

#### Motorcycle Rally Raid and Enduro.

Motorcycle rally raid and enduro riders also need to adapt theirselves to anykind of riding properties from flat salt lakes to tricky stony tracks. They also need different settings for their suspensions in order to be as efficient as possible everywhere without getting off the bike every now and then.

#### Car Rally Raid possibilities.

It is obvious that this advantage would also work on rally and rally raid cars since they are often racing with motorcycles and have the same adherence variability on the track.
2.4 Conclusion

This part of designing prototypes and building prototypes is really interesting but can’t be credible without the first tests. This is what the following chapter aims at.
Chapter 3
Tests and results

This chapter shows how I went through the test part of the project and gives results and comparison with theoretical calculation.

This figure shows how the curves theoretically change associating MR technologies to shim valves.

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3.1 Typical damping tests

Rig testing of complete dampers or their separate parts may be placed under three different headings:

1. to measure performance
2. to check durability
3. to test theoretical models

Testing of theory, especially in our application, is required to validate methods of analysis and to give confidence in theory for design work. This is likely to involve testing of separate parts, of complete valve in steady-flow circuit, or testing of complete dampers to relate damper characteristics to valve characteristic, to investigate piston or rod friction effect, etc. Performance testing is required in competition to check that the given valve set-up gives the expected behaviour and that dampers are consistent and in match pairs. Consistency and matching tests.

**SHOCK ABSORBER DYNAMOMETER.**

In order to save time, rigs got designed to test accurately dampers characteristics. Early cyclic tests were achieved by reciprocating the damper in a roughly sinusoidal manner, with a slider crank mechanism using a connecting rod. The force was measured by elastic deflection of a transverse beam holding the other end of the damper. The inclination of the connecting rod introduces a substantial harmonic into the damper motion, which is therefore quite significantly nonsinusoidal for practical connecting rod length. This can be eliminated by using a Scotch Yoke mechanism which gives a true sinusoid. In cyclic test, the frequency or amplitude is varied to give a desired peak velocity, using:

\[ X = X_0 \sin(\omega t) \]  \hspace{1cm} (3.1)

\[ V = \omega X_0 \cos(\omega t) \]  \hspace{1cm} (3.2)

\[ A = -\omega^2 X_0 \sin(\omega t) \]  \hspace{1cm} (3.3)

with the velocity amplitude (i.e. amplitude of the sinusoidal velocity graph)

\[ V_0 = \omega X_0 = 2\pi f X_0 \]  \hspace{1cm} (3.4)

With the electromechanical drives described it is usual to adjust the frequency either by use of a variable speed DC motor or by a variable gearbox ratio. Variation of stroke may be possible by disassembly of the apparatus, so the stroke is set to give the desired maximum speed, within the limits of the damper and test apparatus. For a linear damping coefficient \( C_D \), at a stroke \( S \) and amplitude \( X_0 = S/2 \), and frequency \( f \), the peak sinusoidal speed is the velocity amplitude

\[ V_0 = \omega X_0 = 2\pi f X_0 \]  \hspace{1cm} (3.5)

The peak power of dissipation is \( C_D V_0^2 \). For sinusoidal motion, the mean power dissipation in one-half of that. The energy dissipation \( E_C \) in one cycle

\[ E_C = \frac{C_D V_0^2}{2f} = 2\pi^2 C_D f X_0^2 \]  \hspace{1cm} (3.6)

Because they are limited in power and control, electromechanical testers are usually limited to small low-powered units. These are suitable for limited testing and low speed comparative
work, including matching at low speeds.

For larger tests it is usually preferred to use hydraulic drive. The hydraulic ram is double acting, typically operating at a pressure around 1 MPa with a force capability of 10 kN. These also allow, thanks to their voltage input, to follow complex profiles:

1. sinusoidal wave
2. triangular wave
3. square wave
4. random motion
5. external input

Such a system can be used for testing dampers or for testing the vehicle complete suspension. In practice the sinusoidal activation is the one mainly used for damper testing. However it mustn’t be forgotten that a damper reaction depends on its past motions and condition.

The basic parameters to be measured may include instantaneous values of:

1. position $X$
2. velocity $V$
3. acceleration $A$
4. force $F$
5. pressure $P$
6. temperature $T$
These need sensors, plus data processing and suitable display. In addition the data stream will be processed to give items such as cyclic extreme position and force.

**Curve Analysis.**

The raw data, except for temperature, change through the cycle, and therefore cannot be displayed effectively by analogue or digital meters. Therefore this form of data are usually presented on an oscilloscope, showing variation with time, or, more usuefully, as a loop such as force against position, $F = f(X)$. A storage type of oscilloscope is much superior in showing the cycle shape. With digital data acquisition, it can of course be displayed on computer. The typical plots of interest are:

1. position vs time
2. velocity vs time
3. acceleration vs time
4. force vs time
5. force vs position
6. force vs velocity

The kinematic parameters $X(t)$, $V(t)$ and $A(t)$ are mainly of use as a visual confirmation that the cycle shape is as desired.

This part of the report illustrates different valving combinations in the form of graphs. The graph shown is force vs. displacement graph. The force vs. displacement graph is a very accurate and simple way to assess valving characteristics. If you are not familiar with this type of graph, it is explained on the following page along with the graph above, showing the four different quadrants.

![Figure 3.3: Oval graphic (Force/Displacement).](image)

**Quadrant #1**

This is the beginning of the compression stroke. Where the graph crosses the zero line (pounds) in quadrant #1 begins the compression stroke. Approximately the first 1/2" of displacement is formed with relation to the low speed bleed bypass. When the shaft reaches
3. Tests and results

3.2 Testing the MR valve at IFMA

3.2.1 Shock absorber dynamometer

Fatigue Machine.

A shock absorber dynamometer being really expensive and hard to pay off with studies, they are not that common. Hopefully, the IFMA owns a fatigue machine for elastomers which works just like an hydraulic tester. Although it limits a bit the possibilities in terms of speed and amplitude combined, it is a strong base to get results on our first prototype. The only need was to design and machine two parts that fits in the holding features in order to mount the prototype.

Software and Postprocessing.

Since its first job is to measure deformations and forces in plastic and in fatigue, the software linked to the fatigue machine is designed to provide time, position and forces values. These are precisely the ones needed to run good and usefull tests on a damper. From that and because of the cycle character of the studies, we still need some postprocessing application to display the results in quadrants. We therefore created a spreadsheet with Excell and Visual Basic to display to usefull quadrants from columns of results. It spots the beginning of the steady sinusoidal mode and takes ten cles after that. It orders and spots the quadrants and save a pdf file of the results. -

3.2.2 Theoretical results

The Principle of Continuity is a statement of the conservation of the mass for a fluid. For a control volume, in steady state, the mass flow rate in the control volume equals the mass flow rate out. In unsteady state, the rate of increase of mass inside the control volume equals the mass inflow rate minus the mass outflow rate. From a practical point of view, this is the basis of many assumption in fluid flow analysis, including the idea that, for example, the mass flow rate along the pipe is the same at all transversal section of the pipe.
Bernouilli’s equations may be considered to be an expression of the conservation of energy for a flowing fluid at constant density. However, it is usually more convenient to think of it as an equation about pressure. In a steady state condition it may be applied to a series of point along a streamline or a series of sections of a streamtube, provided that the losses are negligible. Then, the streamline or streamtube, and the relevant two points or sections must be specified. For uniformity of static pressure over a flow cross-section, the streamlines passing through the section should be straight and parallel. Bernouilli’s equation may then be expressed as

\[ P_1 + \frac{1}{2} \rho u_1^2 + \rho gh_1 = P_2 + \frac{1}{2} \rho u_2^2 + \rho gh_2 \]  

(3.7)

For damper analysis, the potential pressure term \( \rho gh \) does not change significantly, so this term may be omitted. Hence Bernouilli’s equation simplifies to

\[ P_1 + \frac{1}{2} \rho u_1^2 = P_2 + \frac{1}{2} \rho u_2^2 \]  

(3.8)

The pressure terminology is

1. \( P \): static pressure
2. \( q \): dynamic pressure \( q = \frac{1}{2} \rho u^2 \) at fluid speed \( u \)
3. \( P_{St} \): stagnation pressure \( P_{St} = P + q = P + \frac{1}{2} \rho u^2 \)

The static pressure is the pressure actually experienced by the liquid, including, for example, a small particle being swept along with the liquid. Bernouilli’s equation therefore gives a relationship between the static pressure and the flow velocity. Considering flow from a reservoir (i.e. region of zero or negligible speed) to a point where there is a velocity \( u_2 \), then neglecting losses, Bernouilli’s equation gives

\[ P_1 = P_2 + \frac{1}{2} \rho u_2^2 \]  

(3.9)

\[ u_2 = \sqrt{\frac{2(P_1 - P_2)}{\rho}} \]  

(3.10)
We also know that $u_2$ equals to the volumetric flow rate $Q$ divided by the annular orifice area $A_a$, $Q$ is equals to the damper speed $V$ by the piston area $A_P$. Also $P_1$ equals to the viscosity driving force $F_{vis}$ divided by the piston area minus the annular orifice area $A_F$. This is for the case to the compression motion, for rebound, the rod section needs to be subtracted to $A_P$ and $A_F$.

$$A_a = \pi((R_1 + g)^2 - R_1^2) - 5gD_{pin} \tag{3.11}$$

Where $D_{pin}$ is the diameter of the aluminium inserts. So the result is

$$A_a = 31.63 \text{mm}^2 \tag{3.12}$$

We can know get $A_P$ easily

$$A_F = \pi R_2^2 - A_a \tag{3.13}$$

$$A_F = 986.23 \text{mm}^2 \tag{3.14}$$

Bernoulli’s equation gives

$$\frac{QA_F}{A_a} = \sqrt{\frac{2(F_{vis}/A_F - P_2)}{\rho}} \tag{3.15}$$

We can know get the force of reaction of the damper in function of the damper speed.

$$F_{vis} = A_F \left[ \frac{\rho}{2} \left( \frac{V A_F}{A_a} \right)^2 + P_2 \right] \tag{3.16}$$

$P_2$ is the static pressure in the second reservoir, it equals the nitrogen pressure in the remote reservoir, hence $p_2 = 10\text{bar} = 10^6\text{Pa}$. From that, we get the $F_{vis} = f(V)$ relationship for compression motion

$$F_{vis} = 1532.70V^2 + 986.23 \times 10^{-5} \tag{3.17}$$

Now that we have the reaction characteristic of the valve in terms of pure viscosity, we can work on the magnetorheological effect considering the reaction of a Bingham fluid between two plates. Let’s consider consider the gap as two parallel plates of length $L = 2L_{active}$, width $w = \pi(2R_1 + g) - 5D_{pin}$ and spacing (i.e. gap) $g$. The Bingham liquid has a shear yield stress $\tau_y$ and a subsequent marginal viscosity $\mu$. The driving pressure is $P$ giving flowrate $Q$.

The driving force is $F = Pwg$. The resistive shear force without flow may be as large as $F = 2Lg\tau_y$. The resisting force to the flow given by the magnetorheological effect is therefore

$$F_{MR} = 2Lw\tau_y \tag{3.18}$$

We also know that $\tau_y$ is dependent on the magnetic field strength $H$ and obviously to the magnetic flux density in the gap $B_{gap}$.

For the proposed MR valve, the magnetic flux density in the fluid exhibits a linear relationship with the applied coil current

$$B_{gap} = 0.5I (I < 1.6A) \tag{3.19}$$

where the unit of $B_{gap}$ is the Tesla (T).

The dynamic yield stress, $\tau_y$, for the MRF-132LD can be approximately by a cubic function of the magnetic flux density

$$\tau_y = a_3B_{gap}^3 + a_2B_{gap}^2 + a_1B_{gap} + a_0 \tag{3.20}$$

Here polynomial coefficients were determined by a least-squares fit of the dynamic yield stress data as a function of magnetic flux density from the Lord Co., and are:
1. $a_0 = 0.877 \text{kPa}$
2. $a_1 = 17.42 \text{kPa/T}$
3. $a_2 = 122.56 \text{kPa/T^2}$
4. $a_3 = 86.51 \text{kPa/T^3}$

Therefore we have the following equation for the magnetorheological reaction force

$$F_{MR} = 2Lw(a_3(0.5I)^3 + a_2(0.5I)^2 + a_1(0.5I) + a_0) \quad (3.21)$$

Hence the total compression force reaction of the valve is

$$F = F_{vis} + F_{MR} = A_F \left[ \frac{\rho}{2} \left( \frac{V A_P}{A_a} \right)^2 + P_2 \right] + 2Lw(a_3(0.5I)^3 + a_2(0.5I)^2 + a_1 B(0.5I) + a_0) \quad (3.22)$$

For the rebound motion, it is

$$F = F_{vis} + F_{MR} = (A_F - A_R) \left[ \frac{\rho}{2} \left( \frac{V (A_P - A_R)}{A_a} \right)^2 + P_2 \right] + 2Lw(a_3(0.5I)^3 + a_2(0.5I)^2 + a_1 B(0.5I) + a_0) \quad (3.23)$$

where $A_P$ is the area of the rod, here $A_P = 615.75 \text{mm^2}$.

If these equations give the reaction of a Bingham fluid for a steady-state, we didn’t find anything about its reaction with un unsteady flow-rate. Applying fluid mechanic equations such as Bernouilli’s gives different models for the steady-state which correspond in the current range we use, but then no comparison can be done between the physical and theoretical models. Hence, the valve has a really hard shape to work with for fluid mechanics theories because of its section changes, chamfers and so on...

We are still looking for a way to predicate the reaction of the fluid under magnetic field and unsteady-state, but no relevant models seem to exist at the moment.

### 3.2.3 Practical tests and results

**Test 1:** quick evaluation of the force variation depending on the current.

This very first test on the prototype after its break-up is done in order to check the reaction of the different components together, it is also usefull to see the global reaction and to plot a first tendency of the $F = f(V)$ characteristic depending on the current intensity. It shows that one of the plots (0.5 A) is out of the tendency curve, Figure 3.5. This gives a good reason for the second test.

All the results here are given for a rod speed of 50mm.s$^{-1}$ which is quite limited but the machine can’t go faster.Beside that we can notice that for a given velocity and on our intensity range, the force is linear to this intensity. This is a good thing for further command, it won’t have to integrate complicated calculations on this topic.

**Test 2:** random test to check repetability.

Thanks to a virtual eleven faces random dice (where $X$ is the number of the face, the current intensity for the test is $I = X.100mA$), we get forty different tests to check if the valve reaction is even for a same current intensity and not depending on the previous values. These tests are ran in a steady mode, this means that the actual measures are taken from a moment when the magnetic field and obviously the fluid are stabilized.
3. Tests and results

3.2. Testing the MR valve at IFMA

Figure 3.5: Quick $F = f(I)$ analysis.

It shows that for a given velocity with a 5% tolerance due to the measure timing, the only relative difference over 10% is due to one test at 0.7A. Beside that, the spectrums are quite tidy. This also shows that the prototype is a strong basis to further the tests since it is consistent.

Figure 3.6: Random $F = f(I)$ analysis.

**Test 3: Precise $F = f(I)$ Test.**

Now that we know that the prototype has equal reaction for a given intensity current, the aim of this test is to plot two types of curve. The first one giving the reactive force for a given velocity versus the current intensity is really usefull for further management since it gives to intensity to provide to the bobin to get the desired yield stress (*i.e.* the damping coefficient). We get four curves out of this part of the test, two for rebound and two for compression, one of each for positive absolute acceleration and others for negative.
Hence, we wanted to split the four different aspects, Figure 3.7, the reaction of such an hydraulic device depends on its former actions and its actual condition. Therefore it was safer to have four different curves. In the end it shows that the difference between acceleration and deceleration in rebound or compression motion are very similar one to another. Obviously there is a difference between compression and rebound, it is due to the rod section and the pressurized nitrogen.

These first curves all show linearity of the force \( F \) to the intensity of the current \( I \).

The second curve shows in a good way the damper reaction, we notice easily that for a given speed, the force is enhanced by the intensity in a quite linear way. These \( F = f(V) \) curves also show that used in such a way the valve is totally unacceptable, since the damper force is proportional to damper velocity squared. This was not the aim of this valve anyway.

**Test 4: Test to check that the rod position doesn’t affect results.**

This simple test is ran on different positions on the rod. This shows that the damper is not position sensitive which is a good thing since velocity is the only parameter it was suppose to vary along.

**Test 5: Speed increase.**

This simple test is ran with different damper maximum speed. It shows that even with
greater speeds, the $f = f(V)$ keeps the same tendency.

**Test 6: Amplitude Increase.**
This test’s aim and results are the same as above but are run with greater amplitude.

**Test 7: Both.**
This test’s aim and results are the same as above but are run with greater speed and amplitude. This one doesn’t actually give real improvement in the results since the fatigue machine is not able to reach high amplitude and speeds at the same time being limited by its power.

These four quick tests gave good results since they prove that the prototype is not position or frequency sensitive.

### 3.3 Conclusion

This first prototype which aim is to prove that a magnetorheological damper is possible for use on motorcycle (so with all the weight and dimension limitations). With the results of the test above we can say that its function is totally fulfilled. There is still a blank left because of the lack of theory on such a valve in non-steady state, this will need to be fulfilled by tests in order to learn as much as possible about the reactions.

The tests were also limited by the power of the shock absorber dynamometer, further studies will soon require a strong and powerful one.

The next one will have to go further into development, testings and hopefully track testing.
Conclusion

This report shows the possibilities of the magnetorheological fluids used in motorcycle damping.

The analysis of such a valve requires a lot of knowledge and the theoretical parts found in the report has a very limited value. Hence, it doesn’t take account of lots of fluid mechanics subtilities, this theory actually needs a lot of work to be barely understood. The Finite Element Analysis and optimization of the theoretical valve model are also a whole project, they are the courtesy of Dr Weihua Li from the University of Wollongong.

The valve studied here is obviously not perfect and would need work to get improved to be truly useful, the second chapter actually gives some useful advices for a next prototype that should soon appear. It is to keep in mind that this very first evolution is designed from a modified OEM shock absorber of low quality and is therefore just a pre-work to check the quality and veracity of theoretical models and real applications.

If the lack of comparison between theoretical and physical models appears like a hole in this study, it is to note that there is no perfect damping curve as such. Every single rider has different feelings and therefore requires his own settings. It would have been of great interest though to predicate them in theory to save time on tracks.

The next main step this study will need, beside a new prototype, is to get some track testing with experienced riders as soon as an evolution allows it in terms of quality and safety. This would give us much more understanding of what a rider want and how to reach the closest point to be more efficient than other technologies. Hence, the aim of racing is to win!

This technology, if we take the time and money to work on it, could show great improvement in motorcycle racing by giving the vehicle the adherence it needs in any kind of tracks and conditions. I truly hope to be part of this improvement and get this technology to its best.