Chapter 2

Background

The purpose of this chapter is to provide the necessary background for this research. This chapter will first discuss the tradeoffs associated with typical passive single-degree-of-freedom base-excited suspension systems. The conceptual basis for skyhook control will be developed, along with the anticipated improvements in performance. Next, a practical realization of the semiactive skyhook control will be developed such that it can be implemented. The chapter will conclude with a discussion of the practical realization and performance of a semiactive MR damper.

2.1 Skyhook Control of an SDOF System

This section will introduce the typical passive SDOF seat suspensions that are in common use on heavy vehicles. The concept of skyhook control will be introduced, along with the necessities that are required for practical implementation.

2.1.1 Passive Suspension for an SDOF Base-Excited System

The seat suspension system that is typically used on heavy vehicles is a passive single-degree-of-freedom suspension system, which can be modeled as shown in Fig. 2.1. We can derive the transmissibility of the passive seat suspension as

$$\frac{X_1}{X_2} = \frac{1 + j2\zeta_P(\frac{\omega}{\omega_n})}{1 - (\frac{\omega}{\omega_n})^2 + j2\zeta_P(\frac{\omega}{\omega_n})} \tag{2.1}$$

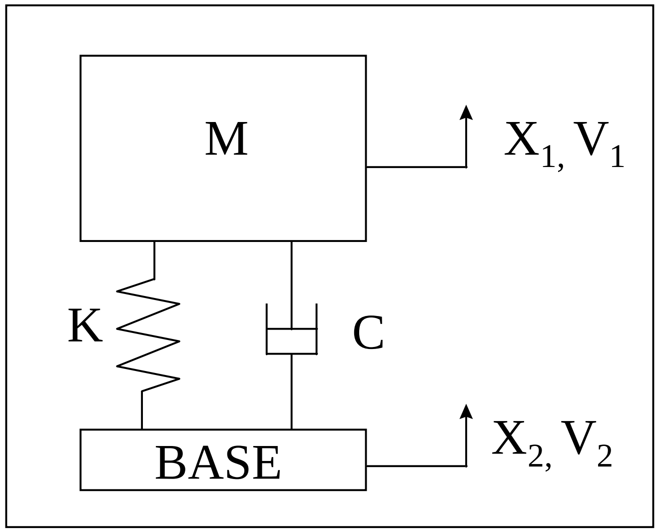


Figure 2.1. SDOF Passive Base-Excited System.

where ζ_P is the passive damping ratio [31]. We can then plot the transmissibility as a function of the quantity $\frac{\omega}{\omega_n}$, resulting in Fig. 2.2 for various damping ratios. Notice that at low passive damping ratios, the resonant transmissibility (around $\omega = \omega_n$) is relatively large, while the transmissibility at frequencies above the resonant peak is quite low. The opposite is true for relatively high damping ratios. Figure 2.2 demonstrates the inherent tradeoff of passive seat suspension systems. If we choose a low damping ratio, we gain superior high frequency isolation but poor resonant frequency control. However, as we increase the damping ratio, we begin to trade off the high frequency isolation for resonance control. Most seat manufacturers tend to favor resonance control over the high frequency isolation, and the resulting ride that the driver experiences is often deemed harsh. Some independent drivers remove the seat damper in favor of a smoother ride while risking the possibility that the seat will eventually hit the physical limits of suspension travel.

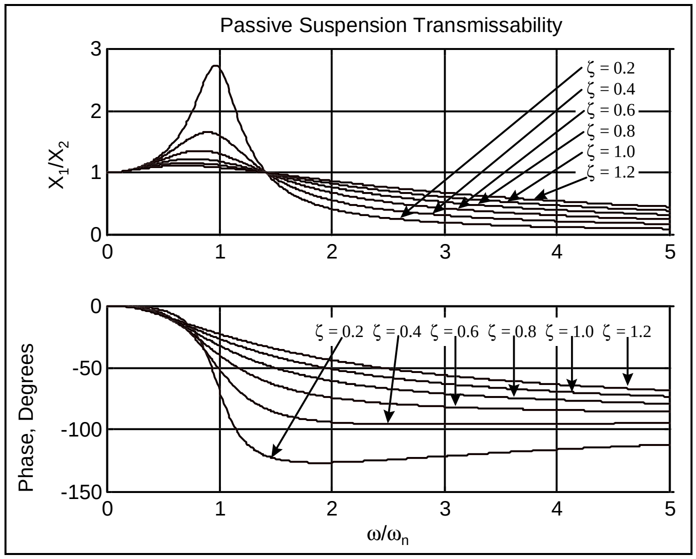


Figure 2.2. Passive Suspension Transmissibility.

2.1.2 Ideal Skyhook Control

One method to eliminate the tradeoff between resonance control and high frequency isolation is to reconsider the configuration of the suspension system. For instance, consider moving the damper from between the suspended mass and the base to the position shown in Fig. 2.3. The damper is now connected to an inertial reference in the sky (i.e., a ceiling that remains vertically fixed relative to a ground reference). Notice that this is a purely fictional configuration, since for this to actually happen, the damper must be attached to a reference in the sky that remains fixed in the vertical direction, but is able to translate in the horizontal direction. Ignoring this problem at the moment, we will focus on the performance of this configuration.

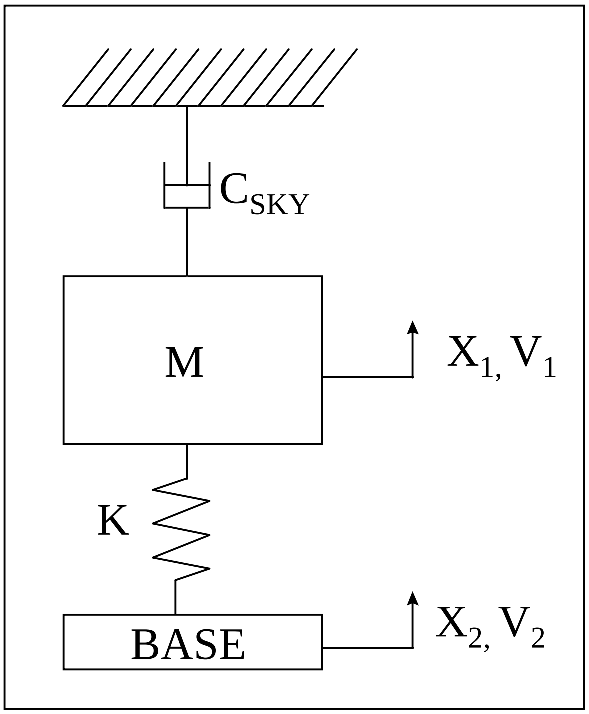


Figure 2.3. Ideal Skyhook Configuration.

The transmissibility of this configuration can be derived to be

$$\frac{X_1}{X_2} = \frac{1}{1-(\frac{\omega}{\omega_n})^2 + j2\zeta_{SKY}(\frac{\omega}{\omega_n})} \tag{2.2}$$

where, in this case, ζ_{SKY} is the ideal skyhook damping ratio. Once again, if we plot the transmissibility for various values of ζ_{SKY}, we find the results shown in Fig. 2.4. As in the passive case, as the skyhook damping ratio increases, the resonant transmissibility decreases. Increasing the skyhook damping ratio, however, does not increase the transmissibility above the resonant frequency. For sufficiently large skyhook damping ratios (i.e., above ζ>0.707), we can isolate even at the resonance frequency. This is encouraging since we have removed the tradeoff associated with passive dampers.

There exist a large number of studies on the effectiveness of the skyhook control policy along with other optimal control techniques. Most of these studies indicate that skyhook control is the optimal control policy in terms of its ability to isolate the suspended mass from the base excitations [26].

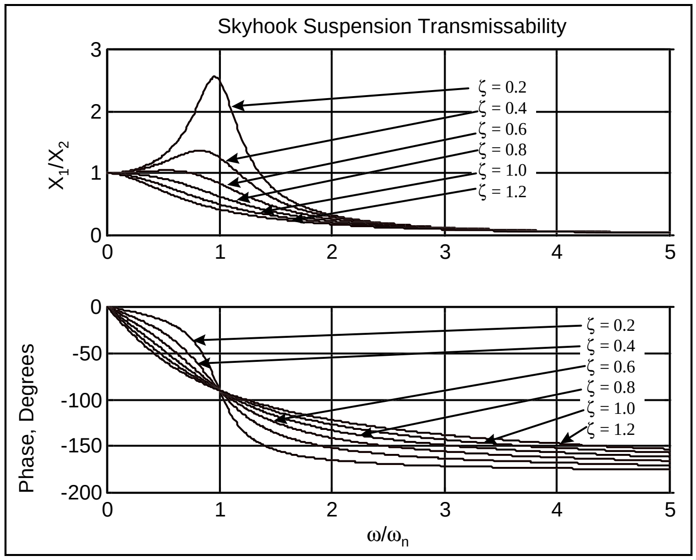


Figure 2.4. Ideal Skyhook Transmissibility.

One method of generating the skyhook damping force is to remove the passive suspension (i.e., both the damper and the spring) and replace it with an active force generator. This can be achieved by using a hydraulic actuator, however, the resulting system is rather complex and requires a significant amount of power [32]. Another approach to achieving skyhook damping is to use semiactive dampers. Semiactive dampers allow for the damping coefficient, and therefore the damping force, to be varied between high and low levels of damping. Early semiactive dampers were mechanically adjustable by opening or closing a bypass valve. The only power required for the damper is the relatively small power to actuate the valve. For this research, we are using a magnetorheological damper which varies the damping by electrically changing the magnetic field applied to the magnetorheological fluid. The SDOF model shown in Fig. 2.1 modifies to Fig. 2.5, where the damping coefficient, $C_{CONTROLLABLE}$, varies over time. This configuration will be referred to as the semiactive suspension.

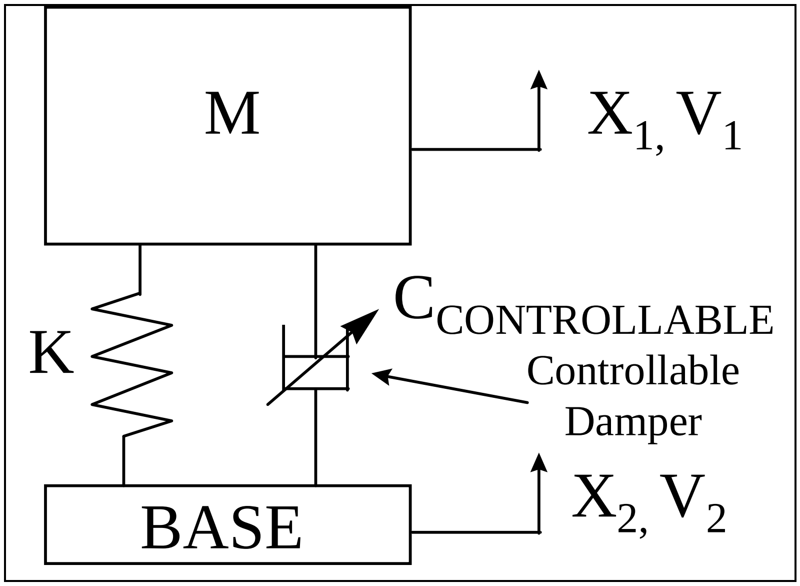


Figure 2.5. Semiactive Suspension.

2.1.3 Semiactive Realization of Skyhook Control

Once we have decided that we will use a semiactive damper, we must determine how to modulate the damper such that it emulates a skyhook damper. We first define the velocity of the suspended mass relative to the base, V_{12}, to be positive when the base and mass are separating (i.e., when V_1 is greater than V_2) for both systems. Now assume that for both systems, the suspended mass is moving upwards with a positive velocity V_1. If we consider the force that is applied by the skyhook damper to the suspended mass, we notice that it is in the negative X_1 direction, or

$$F_{SKY} = -C_{SKY} V_1 \tag{2.3}$$

where F_{SKY} is the skyhook force. Next, we need to determine if the semiactive damper is able to provide the same force. If the base and suspended mass in Fig. 2.5 are separating, then the semiactive damper is in tension. Thus, the force applied to the suspended mass is in the negative X_1 direction, or

$$F_{CONTROLLABLE} = -C_{CONTROLLABLE} V_{12} \tag{2.4}$$

where $F_{CONTROLLABLE}$ is the force applied to the suspended mass. Since we are able to generate a force in the proper direction, the only requirement to match the skyhook suspension is

$$C_{CONTROLLABLE} = C_{SKY}\frac{V_1}{V_{12}} \tag{2.5}$$

To summarize, if V_1 and V_{12} are positive, $C_{CONTROLLABLE}$ should be defined as in Eq. (2.5).

Now consider the case in which the base and suspended mass are still separating, but the suspended mass is moving downwards with a negative velocity V_1. In the skyhook configuration, the damping force will now be applied in the upwards, or positive, X_1 direction. In the semiactive configuration, however, the semiactive damper is still in tension, and the damping force will still be applied in the downwards, or negative, direction. Since the semiactive damping force cannot possibly be applied in the same direction as the skyhook damping force, the best that can be achieved is to minimize the damping force. Ideally, the semiactive damper is desired to be set so that there is no damping force, but in reality there is some small damping force present and it is not in the same direction as the skyhook damping force. Thus, if V_{12} is positive and V_1 is negative, we need to minimize the semiactive damping force.

We can apply the same simple analysis to the other two combinations of V_1 and V_{12}, resulting in the well-known semiactive skyhook control policy [33]:

$$\begin{cases} V_1V_{12} > 0 & \mathrm{F_{SA}} = C_{SKY}V_1 \\ V_1V_{12} < 0 & \mathrm{F_{SA}} = 0 \end{cases} \tag{2.6}$$

where F_{SA} is the semiactive skyhook damper force. Eq. (2.6) implies that when the relative velocity across the suspension (V_{12}) and sprung mass (V_1) have the same sign, a damping force proportional to V_1 is desired. Otherwise, the minimal amount of damping is desired. Further, Eq. (2.6) provides a very simple method to emulate the ideal skyhook suspension system using only a semiactive damper.

2.2 Magnetorheological Dampers

The purpose of this section is to introduce the theoretical and practical applications of a magnetorheological (MR) fluid for a controllable MR damper. First, the concept of the MR fluid will be introduced. Next, the practical realization of an MR damper will be discussed. Finally, the performance of the MR damper used for this research will be investigated.

2.2.1 Magnetorheological Fluids

Magnetorheological fluids are materials that exhibit a change in rheological properties (elasticity, plasticity, or viscosity) with the application of a magnetic field. The MR effects are often greatest when the applied magnetic field is normal to the flow of the MR fluid. Another class of fluids that exhibit a rheological change is electrorheological (ER) fluids. As the name suggests, ER fluids exhibit rheological changes when an electric field is applied to the fluid. There are, however, many drawbacks to ER fluids, including relatively small rheological changes and extreme property changes with temperature. Although power requirements are approximately the same [18], MR fluids only require small voltages and currents, while ER fluids require very large voltages and very small currents. For these reasons, MR fluids have recently become a widely studied 'smart' fluid.

Besides the rheological changes that MR fluids experience while under the influence of a magnetic field, there are often other effects such as thermal, electrical, and acoustic property changes. However, in the area of vibration control, the MR effect is most interesting since it is possible to apply the effect to a hydraulic damper. The MR fluid essentially allows one to control the damping force of the damper by replacing mechanical valves commonly used in adjustable dampers. This offers the potential for a superior damper with little concern about reliability, since if the MR damper ceases to be controllable, it simply reverts to a passive damper.

2.2.2 Construction of an MR Damper

Magnetorheological (MR) fluids are manufactured by suspending ferromagnetic particles in a carrier fluid. The ferromagnetic particles are often carbonyl particles, since they are relatively inexpensive. Other particles, such as iron-cobalt or iron-nickel alloys, have been used to achieve higher yield stresses from the fluid [17]. Fluids containing these alloys are impractical for most applications due to the high cost of the cobalt or nickel alloys.

A wide range of carrier fluids such as silicone oil, kerosene, and synthetic oil can be used for MR fluids. The carrier fluid must be chosen carefully to accommodate the high temperatures to which the fluid can be subjected. The carrier fluid must be compatible with the specific application without suffering irreversible and unwanted property changes. The MR fluid must also contain additives to prevent the sedimentation of and promote the dispersion of the ferromagnetic particles.

A top-level functional representation of the MR damper is shown in Fig. 2.6. The fluid that is transferred from above the piston to below (and vice-versa) must pass through the MR valve. The MR valve is a fixed-size orifice with the ability to apply a magnetic field, using an electromagnet, to the orifice volume. This results in an apparent change in viscosity of the MR fluid, causing a pressure differential for the flow of fluid which is directly proportional to the force required to move the damper rod.

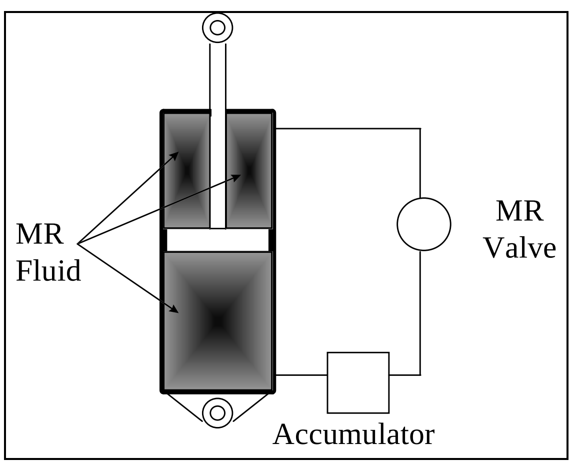
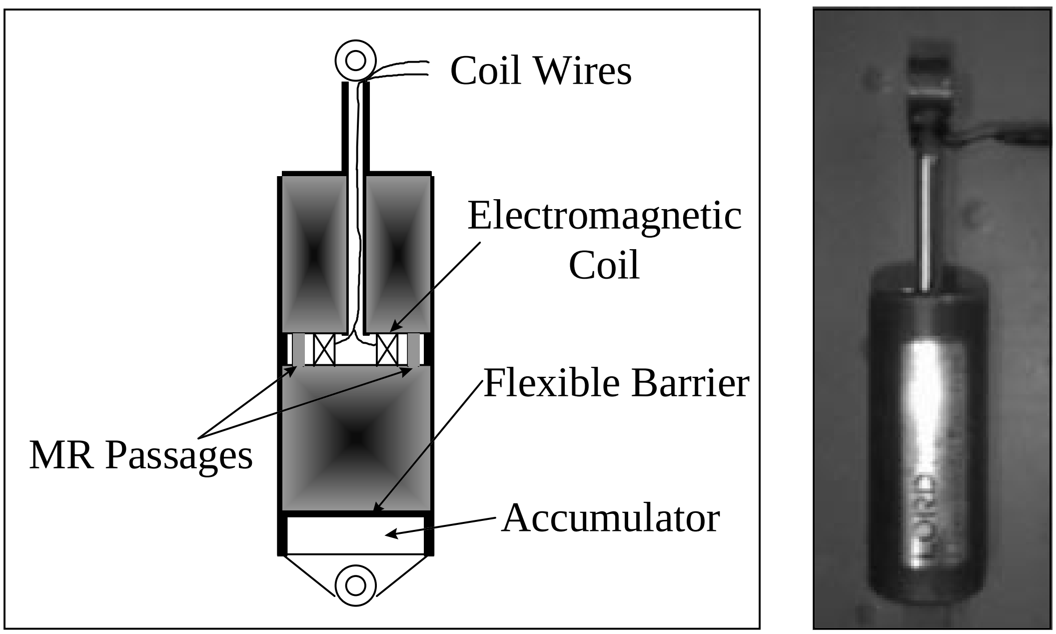


Figure 2.6. Functional Representation of an MR Damper.

The accumulator is a pressurized volume of gas that is physically separated from the MR fluid by a floating piston or bladder. The accumulator serves two purposes. The first is to provide a volume for the MR fluid to occupy when the shaft is inserted into the damper cylinder. The second is to provide a pressure offset so that the low pressure side of the MR valve is not reduced enough to cause cavitation of the MR fluid.

An elegant and compact design of the MR damper developed by Lord Corporation and used for this research is shown in Fig. 2.7. All of the external components have been incorporated internally. This provides a compact design that is very similar in size and shape to existing passive dampers. The only external parts are the two electrical leads for the electromagnet, which are connected to the controller.



(a) (b)

Figure 2.7. Lord MR Seat Damper, (a) Schematic Representation, (b) Actual Hardware.

2.2.3 Performance of the MR Damper

For typical passive dampers, the damper performance is often evaluated based on the force vs. velocity characteristics. For an ideal viscous damper, the force vs. velocity performance is shown in Fig. 2.8. The slope of the force vs. velocity line is known as the damper coefficient, C. Frequently, the force vs. velocity line is bilinear and asymmetric, with a different value of C for jounce (compression) and rebound (extension), as shown in Fig. 2.9. In the case of a vehicle suspension, the damping curve is shaped (or tuned) by a ride engineer for each particular application. Therefore, the operational envelope of a passive damper is confined to a pre-designed force-velocity characteristic.

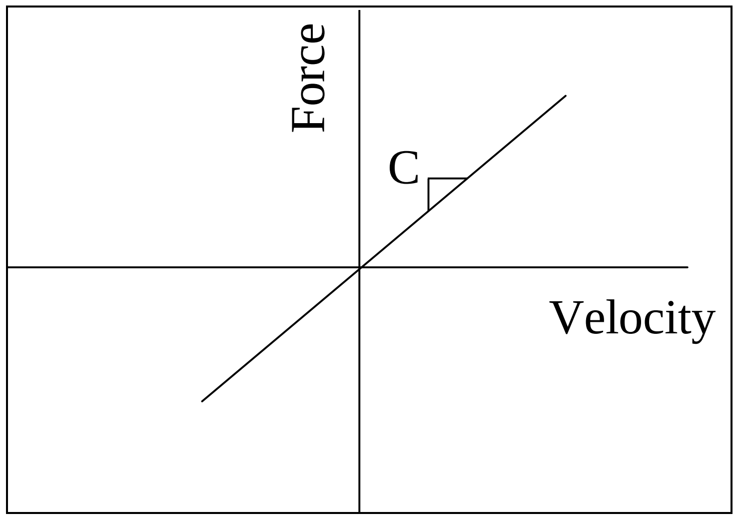


Figure 2.8. Linear Damper Characteristics.

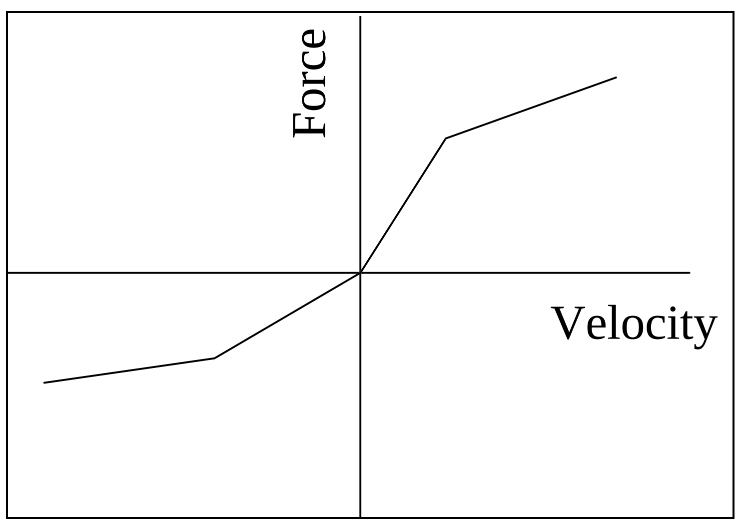


Figure 2.9. Bilinear, Asymmetric Damping Characteristics.

In the case of MR dampers, the ideal force vs. velocity characteristics are as shown in Fig. 2.10. This results in a force vs. velocity envelope that can be described as an area rather than a line in the force-velocity plane. Effectively, the controller can be programmed to emulate any damper force-velocity characteristic or control policy within the envelope.

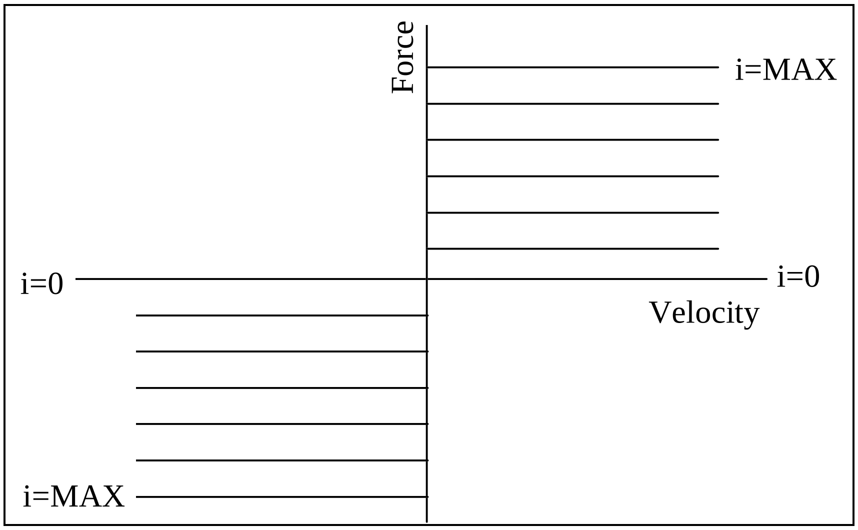


Figure 2.10. Ideal MR Damper Performance.

We can model the ideal MR damper according to

$$F_{MRDAMPER} = \alpha i \tag{2.7}$$

where α is a constant and i is the damper current.

Figure 2.11 shows the nonlinear force-velocity characteristics for the MR damper used for this research. The model in Eq. (2.7) and Fig. 2.10 does not capture the fine details of the actual MR damper, but it captures the gross behavior of the MR damper. Some of the effects missing from the model include the magnetic field saturation, hysteresis, and the force due to the pressurized accumulator. As will be shown in later chapters, this approximation is sufficient for designing MR dampers for most applications, including the seat suspensions.

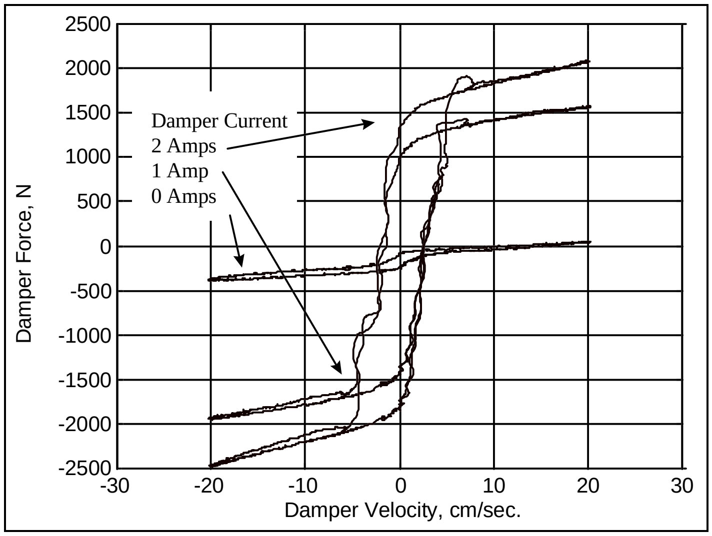


Figure 2.11. MR Damper Performance Envelope.