Characterizing a Racing Damper’s Frequency Dependent Behavior with an Emphasis on High Frequency Inputs

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Abstract

As a racecar negotiates a track, it is subjected to many inputs at both high and low frequencies. These inputs come from the track surface, the motion of the body, and from aerodynamic disturbances. The damper’s ability to control these inputs leads to improved grip at the tires, which increases overall handling of the vehicle. Since dampers have always been assumed to be primarily velocity dependent, little work has gone into exploring damper’s frequency dependent nature. Therefore, this study evaluates the effect input frequency has on the damper’s output force.

Utilizing experimental testing, with a state of the art damper dynamometer, and computer simulation with a parametric damper model developed for this study, several inputs and key parameters are tested, and the damper’s frequency dependent nature starts to emerge. Constant peak velocity sinusoidal and sinusoidal sweep inputs are used for the experimental testing. The results show that as the input frequency is increased, the damper’s output force lissajou transitions from the characteristic shape of a damper’s lissajou to a shape characteristic of a spring’s lissajou. This change in the lissajou is linked to hysteretic effects, which includes the gas spring effect. Damper parameters that are suspected to contribute to the hysteretic effects are explored with computer simulation and additional experimental testing. The results from this show that fluid preparation, fluid type, initial gas pressure, and friction have a predictable effect on the damper’s output force.
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# Content

Abstract ........................................................................................................................................ i

Acknowledgments ..................................................................................................................... ii

List of Figures .......................................................................................................................... vi

List of Tables ............................................................................................................................ xii

1 Introduction ......................................................................................................................... 1
  1.1 Motivation ....................................................................................................................... 1
  1.2 Objectives ...................................................................................................................... 2
  1.3 Approach ....................................................................................................................... 2
  1.4 Contributions ................................................................................................................. 3
  1.5 Outline ............................................................................................................................ 4

2 Background ......................................................................................................................... 5
  2.1 Damper’s Role on a Vehicle ......................................................................................... 5
    2.1.1 Historical ............................................................................................................... 5
    2.1.2 Passenger Vehicles .............................................................................................. 6
    2.1.3 Racecars .............................................................................................................. 6
  2.2 Damper Component Outline ......................................................................................... 7
  2.3 Damper Operation Overview ....................................................................................... 11
  2.4 Damper Dynamometer and Test Procedures .............................................................. 13

3 Investigation into Damper Behavior at Low and High Frequencies ................................. 17
3.1 A Fundamental Look at Damper Behavior

3.1.1 Limit Equation Analysis

3.1.2 Linear Response

3.2 Sinusoidal Input with Constant Velocity

3.2.1 Experimental Setup for a Sinusoidal Input with Constant Velocity

3.2.2 Experimental Results

3.3 Sine Sweep Input

3.4 Results and Discussion

4 High Frequency Behavioral Study Utilizing a Parametric Damper Model

4.1 Model Development

4.2 Model Tuning

4.3 Model Study on Parameter Variation and its Effect on Damper Output Force as Frequency Increases

4.3.1 Initial Gas Pressure Variation in the Model

4.3.2 Friction Variation in the Model

4.3.3 Density Variation in the Model

4.4 Results and Discussion

5 Experimental Evaluation of Parameters Hysteretic Effects

5.1 Experimental Evaluation of Parameters Effect on Hysteresis in the Damper’s Output Force

5.1.1 Initial Gas Pressure Variation
List of Figures

Figure 2-1: Cutaway View of a Penske 7300 Race Damper ............................................................. 8
Figure 2-2: Cutaway View of a Damper’s Head Valve ................................................................. 9
Figure 2-3: Cutaway View of a Damper’s Piston/Shaft Assembly ................................................. 9
Figure 2-4: Piston Types and their Force vs. Velocity Plot ........................................................... 10
Figure 2-5: Bleed Flow in the Main Piston During Rebound ....................................................... 11
Figure 2-6: Valve Flow through the Rebound Shim Stack in the Main Piston .............................. 12
Figure 2-7: Bleed Flow and Valve Flow in the Head Valve .......................................................... 13
Figure 2-8: Roehrig Electromagnetic Actuated Damper Dynamometer ...................................... 14
Figure 2-9: Damper Dynamometer Test Configuration ............................................................... 16
Figure 3-1: Spring and Damper in Series .................................................................................... 18
Figure 3-2: Output Force vs. Displacement and Force vs. Velocity for a Linear Damper with a Constant Damping Coefficient of 3 .................................................................................. 21
Figure 3-3: Output Force vs. Displacement and Force vs. Velocity for a Linear spring with a Constant Coefficient Spring of 30 .............................................................................................................. 21
Figure 3-4: Spring and Damper in Series .................................................................................... 22
Figure 3-5: Output Force vs. Displacement and Force vs. Velocity with c = 3 and k = 0.3 ......... 24
Figure 3-6: Output Force vs. Displacement and Force vs. Velocity with c = 3 and k = 30 .............. 25
Figure 3-7: Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 4 Hz ................................................................. 26
Figure 3-8: Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 16 Hz ................................................................. 26
Figure 3-9: Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 32 Hz ................................................................. 27
Figure 3-10: Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 64 Hz ................................................................. 27

Figure 3-11: Shim Stack Letter Designations and Corresponding Output Force ("From: Race Car Vehicle Dynamics, Figure 22.42")...................................................................................................... 30

Figure 3-12: Description of the Parts of a Damper’s Force vs. Velocity Lissajou ................................................................. 32

Figure 3-13: Illustration of Hysteresis Occurring over a Complete Sinusoidal Input Cycle. Note, in this Illustration, Compression is Shown with Negative Velocity and Rebound with Positive Velocity ................................................................................................................................................................. 33

Figure 3-14: Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 1 Hz ................................................................................................................. 34

Figure 3-15: Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 4 Hz ................................................................................................................. 35

Figure 3-16: Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 16 Hz ................................................................................................................. 36

Figure 3-17: Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 32 Hz ................................................................................................................. 36

Figure 3-18: Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 64 Hz ................................................................................................................. 37

Figure 3-19: Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 128 Hz ................................................................................................................. 37

Figure 3-20: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 4 Hz ................................................................................................................. 38

Figure 3-21: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 16 Hz ................................................................................................................. 39

Figure 3-22: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 32 Hz ................................................................................................................. 39

Figure 3-23: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 64 Hz ................................................................................................................. 40
Figure 3-24: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak
Velocity, at Frequencies of 1 Hz and 128 Hz

Figure 3-25: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak
Velocity, at Frequencies of 1 Hz and 32 Hz and 64 Hz

Figure 3-26: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak
Velocity, at Frequencies of 1 Hz and 32 Hz and 128 Hz

Figure 3-27: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and
a Frequency Range from 1 to 10 Hz

Figure 3-28: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and
a Frequency Range from 40 to 50 Hz

Figure 3-29: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and
a Frequency Range from 90 to 100 Hz

Figure 3-30: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and
a Frequency Range from 1 to 10 Hz vs. 20 to 30 Hz

Figure 3-31: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and
a Frequency Range from 1 to 10 Hz vs. 40 to 50 Hz

Figure 3-32: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and
a Frequency Range from 1 to 10 Hz vs. 70 to 80 Hz

Figure 3-33: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and
a Frequency Range from 1 to 10 Hz vs. 90 to 100 Hz

Figure 3-34: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity,
Normalized Displacements, and a Frequency Range from 1 to 10 Hz vs. 90 to 100 Hz

Figure 3-35: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Velocity Peak and
a Frequency Range from 1 to 10 Hz vs. 90 to 100 Hz

Figure 4-1: Free Body Diagram of Piston and Shaft Assembly

Figure 4-2: Free Body Diagram for Gas Piston

Figure 4-3: Free Body Diagram for Shim Stack Stiffness
Figure 4-4: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz................................................................................................................................. 57

Figure 4-5: Experimental Data and Filtered Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz................................................................. 58

Figure 4-6: Experimental Force Data vs. Model Force Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz................................................. 59

Figure 4-7: Experimental Force Data vs. Model Force Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz................................................. 60

Figure 4-8: Warping Filter which Adds Hysteresis to the Model’s Output Force ........................................ 60

Figure 4-9: An Ideal Output Force Backbone $F(v)$ is Shown in Black, and the Warping Function $g(v)$ is Shown in Red ................................................................................................................... ..... 61

Figure 4-10: Warping Filter Parameters Found from Experimental Output Force Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz ................................................................................................................................................. 63

Figure 4-11: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz............. 63

Figure 4-12: Experimental Force Data, Model Force Data, and Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz.............................................................................................................................. 64

Figure 4-13: Experimental Force Data, Model Force Data, and Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz.............................................................................................................................. 64

Figure 4-14: Error vs. Time and Experimental Force Data, and Model Force vs. Time for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz ................................................................................................................................. 64

Figure 4-15: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz............. 66

Figure 4-16: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 40 to 50 Hz .......... 67
Figure 4-17:  Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 40 to 50 Hz .......... 67

Figure 4-18:  Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 90 to 100 Hz .......... 68

Figure 4-19:  Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 90 to 100 Hz .......... 68

Figure 4-20:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Initial Gas Pressure Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 72

Figure 4-21:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Initial Gas Pressure Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 73

Figure 4-22:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Initial Gas Pressure Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 74

Figure 4-23:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Friction Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 75

Figure 4-24:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Friction Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 76

Figure 4-25:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Friction Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 77

Figure 4-26:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Fluid Density Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 78

Figure 4-27:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Fluid Density Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 79

Figure 4-28:  Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Fluid Density Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz ........................................ 80

Figure 5-1:  Sinusoidal Input, Constant Peak Velocity, 1Hz Frequency, Initial Gas Pressure Variation .. 85

Figure 5-2:  Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Initial Gas Pressure Variation . 86

Figure 5-3:  Sinusoidal Input, Constant Peak Velocity, 16 Hz Frequency, Initial Gas Pressure Variation 86
Figure 5-4: Sinusoidal Input, Constant Peak Velocity, 64 Hz Frequency, Initial Gas Pressure Variation 87

Figure 5-5: Cutaway of Main Piston with out any Modification .............................................................. 88

Figure 5-6: Cutaway of Main Piston with Modification ................................................................. 89

Figure 5-7: Sinusoidal Input, Constant Peak Velocity, 1Hz Frequency, Internal Friction Variation....... 90

Figure 5-8: Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Internal Friction Variation....... 90

Figure 5-9: Sinusoidal Input, Constant Peak Velocity, 16 Hz Frequency, Internal Friction Variation..... 91

Figure 5-10: Sinusoidal Input, Constant Peak Velocity, 64 Hz Frequency, Internal Friction Variation .. 91

Figure 5-11: Vacuum Degassing of Damper Fluid ............................................................................. 93

Figure 5-12: Sinusoidal Input, Constant Peak Velocity, 1Hz Frequency, Fluid type Variation .......... 94

Figure 5-13: Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Fluid Type Variation .......... 94

Figure 5-14: Sinusoidal Input, Constant Peak Velocity, 16 Hz Frequency, Fluid Type Variation .......... 95

Figure 5-15: Sinusoidal Input, Constant Peak Velocity, 64 Hz Frequency, Fluid Type Variation .......... 95

Figure 5-16: Sinusoidal Input, Constant Peak Velocity, 1 Hz Frequency, Fluid Preparation Variation... 96

Figure 5-17: Sinusoidal Input, Constant Peak Velocity, 1 Hz Frequency, Fluid Preparation Variation... 97

Figure 5-18: Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Fluid Preparation Variation... 97

Figure 5-19: Sinusoidal Input, Constant Peak Velocity, 16 Hz Frequency, Fluid Preparation Variation. 98

Figure 5-20: Sinusoidal Input, Constant Peak Velocity, 64 Hz Frequency, Fluid Preparation Variation. 98
List of Tables

Table 3-1: Amplitude, Speed and Frequency of the Sinusoidal Inputs ..................................................... 25

Table 3-2: Damper Configuration for Experimental Testing .................................................................... 29

Table 3-3: Amplitude, Speed, and Frequency of the Sinusoidal Inputs .................................................... 31

Table 4-1: Experimental Data vs. Model without Warping Filter ............................................................. 59

Table 4-2: Experimental Data vs. Model Output Force with Warping Filter ............................................ 69

Table 4-3: Tuned Parameters for 1 – 10 Hz, 40 – 50 Hz, and 90 – 100 Hz .............................................. 69

Table 4-4: Difference in Output Force as the Parameter is Varied ........................................................... 74

Table 4-5: Difference in Output Force as the Parameter is Varied ........................................................... 77

Table 4-6: Difference in Output Force as the Parameter is Varied ........................................................... 80

Table 5-1: Measured Gas Force for 50, 100, 150, and 200 psi Initial Gas Pressure ................................. 84

Table 5-2: Density and Specific Gravity for Different Fluids ................................................................. 92
1 INTRODUCTION

This chapter opens this study by first presenting the reader with the motivation for this work. Next the objectives of this study and the approach to complete these objectives are presented. The contributions of this study are discussed and an outline for the rest of this study concludes this chapter.

1.1 Motivation

Vehicle suspensions have evolved over the last one-hundred years from their humble existence on horse drawn carriages, to their complex forms on modern automobiles. Along with this growth in technology, comes an increase in our understanding of the components that make up a modern suspension. One of the components that is yet to be fully understood is the automotive damper or shock absorbers.

The modern damper’s operation may appear to be simple, fluid flowing through combinations of valves and orifices to dissipate energy stored by the springs and the kinetic energy from the body and unsprung mass. However, interactions of the damper’s internal mechanisms make it a very complex device. Because of the complexity of the damper, there have been numerous damper studies conducted, and many damper models developed for passenger vehicles. However, little has been written about the modern racing damper. This lack of literature can, to some extent, be attributed to the secrecy associated with the racing industry, and also is attributed to the lack of understanding of the modern race damper.

This study will give a fundamental perspective of the modern race damper, based primarily on experimental testing. Beyond that, this study will develop and utilize a parametric model of the Penske 7300 race damper to examine the effects of specific parameters on the damper’s output force. From this parametric model, areas of potential gains can be explored and validated with testing of a physical damper on a damper dynamometer.
1.2 Objectives

A modern, highly adjustable race damper is far more complex than a regular automotive damper, due to the freedom a race technician has to change damper settings and components. Because of this complexity, an all inclusive study of a race damper is beyond the scope of this work. This work will, however, lay the foundation for future work, and provide a base of knowledge for those who work with and use race dampers. Therefore, the primary objectives of this study are to:

1. Better understand the fundamental physical characteristics of race dampers in both low and high frequency regimes.

2. Reexamine the commonly held notion that a damper’s force is only a function of velocity.

3. Evaluate how the force characteristics of a damper changes from various combinations of displacements and frequencies with peak velocity held constant.

4. Provide test methodologies which can be used for accurately evaluating damper characteristics at various frequencies.

5. Provide analytical models, supported by test data for performing parametric studies on race dampers.

1.3 Approach

In order to complete the objectives presented above,

- Damper testing will be conducted with a Roehrig shock dynamometer.

- A damper model will be developed to conduct a parametric study.
The damper dynamometer that will be used for testing is state of the art in that it uses electromagnetic actuation, which will allow for precise control and measurement of the damper’s displacement, velocity, and force. This will allow for high frequency, low displacement experiments to be conducted on the damper. The damper model used in this study is based on a previously developed model of a race damper that does not contain all of the features of the dampers available today. This model is updated so that it contains the new features of current race dampers. One of the features added to current dampers is the head valve or base valve.

1.4 Contributions

The intended contributions of this study are to:

- Give a fundamental understanding of a racing damper’s operation.
- Provide an experimentally validated analytical model for parametric studies.
- Present testing methodologies that allow for damper testing at both low and high frequencies.

Even though this study will be conducted with only one type of race damper, the information it presents will be useful in characterizing other dampers, both for racecars and for passenger vehicles.

This study will also add to the base of knowledge associated with the Virginia Institute for Performance Engineering and Research (VIPER). VIPER is an organization which brings together Virginia Tech, The Institute for Advanced Learning and Research (IALR), and Old Dominion University (ODU) to bridge the gap between academic institutions and industries which focus on advanced research, such as the racing industry. To help bridge this gap, VIPER uses advanced research equipment such as an 8-post test rig and a state of the art damper dynamometer. The results of this study will be beneficial to future research done by VIPER.
1.5 Outline

To provide the reader with the direction of the rest of this study, this section outlines the rest of this document. Besides this first introductory chapter, there are five additional chapters. Chapter two gives the reader an understanding of the damper’s role on the vehicle, the components within the damper, how the damper works, and how the damper is tested. In chapter three, there is an experimental investigation of the damper. Chapter four presents a model of the damper, and shows the effect specific parameters have on damper output force. The results from the parameter study in chapter four are validated in chapter five with more experimental testing. The overall study is brought together in chapter six where conclusions and recommendations for future work are presented.
2 BACKGROUND

This chapter is intended to give the reader, who is not familiar with dampers, or more specifically race dampers, an understanding into their components and operation. Starting with the damper’s historical role on a vehicle, the first section proceeds to reveal the damper’s role on passenger vehicles and racecars. The next section will familiarize the reader with the components of the damper used in this study. A brief discussion of how a damper operates follows. Concluding this chapter is a discussion on damper testing.

2.1 Damper’s Role on a Vehicle

This section will begin by briefly looking back at the damper’s development throughout history. Attention is then turned to the role the damper plays on passenger vehicles and racecars.

2.1.1 Historical

Through an extensive literature study, Dixon [5] has brought together the history of the automotive damper’s development. The following paragraph references his work, with additional discussion of current damper technology.

One of the first modes of transportation that was owned by an individual or family was the horse-drawn wagon. The damping on horse drawn wagons came from friction between the beams of the leaf springs on semi-elliptic and full-elliptic suspensions. Development of the automobile, and the increase in vehicle speed, lead to an increase in damper use. Still, damper acceptance was slow at first. The following gives an outline of the dampers use throughout history:

- Before 1910, dampers had a limited presence on motorized vehicles.
• Between 1910 and 1925, the damping of a vehicle’s suspension came in the form of dry snubbers.

• From 1925 to 1980 hydraulic dampers were dominated first by dampers with simple constant force blow-off characteristics, then by dampers with proportional characteristics, and finally by adjustable dampers.

• In the early 1980’s (1980-85), there was a considerable amount of research conducted with active suspensions. The development of active suspensions has come about slowly, mainly due to high production costs.

• From 1985 on, there has been an increase in research in auto-adjusting dampers, including Magneto-Rheological (MR) dampers. This has lead to the inclusion of MR dampers on certain high end production vehicles including Corvettes and some Cadillacs.

2.1.2 Passenger Vehicles

The damper’s role on a passenger vehicle is based on a compromise between ride comfort and vehicle handling. If the damper is too stiff, high frequency vibrations will be transmitted to the passengers. With too low of damping, low frequency motion will be passed through to the passengers, and motion sickness can occur. The performance of a passenger vehicle damper can be measured by the root mean square (RMS) acceleration at the driver’s seat [18]. Passenger vehicle dampers are designed for long life, and are replaced rather than serviced. With the exception of aftermarket dampers, passenger vehicle dampers cannot be tuned for different driving applications.

2.1.3 Racecars

Unlike a passenger vehicle damper, a racing damper’s primary role is to contribute to the overall handling characteristics of the vehicle. This is accomplished with higher
damping. Low frequency motion of the body is unacceptable on racecars, and along with controlling the motion to the unsprung mass, the higher damping also improves road holding of the racecar [18]. The only drawback to this increased damping is the increase in the harshness of the ride. According to Dixon, “in particular, the dampers (race dampers) affect traction and tire grip, and influence the fore-aft distribution of lateral load transfer during body roll at corner entry and exit, which has a significant effect on transient handling behavior. This is especially important for racing cars with extreme aerodynamics, which are very sensitive to ride height.”[5]

### 2.2 Damper Component Outline

Now that we have looked at the damper’s progression throughout history, and its use on passenger vehicles and racecars, this next section will give a detailed perspective of a race damper. Although there are a wide variety of damper types, this study is limited to one particular damper, the Penske 7300. This damper is a monotube damper for use in NASCAR Nextel Cup racecars. Unlike most automotive dampers, racing dampers are highly adjustable, and comprised of many components. This section highlights and describes the major components of the Penske 7300. The main damper components, described in this section, can be seen in Figure 2-1.

Starting at the top of the damper, the gas chamber housing which is threaded to the damper’s main body has a high angularity spherical joint on top that allows the damper to be connected to the chassis. A Schrader valve is threaded through the gas chamber housing, which allows for the adjustment of initial gas pressure.
Internally, the gas chamber houses the gas piston which separates the inert gas, usually nitrogen, from the damper’s fluid. Below the gas piston is the head valve, which separates the head valve chamber from the compression chamber. The head valve is stationary during operation, and is threaded into the damper body so that a technician can remove it to change its’ shim stacks or bleed jet, as shown in Figure 2-2. The head valve is designed to accommodate many different shim stack and bleed jet configurations. Like the main piston, the thickness and number of shims determine whether the flow through the head valve is bleed flow or orifice flow or a combination of both. For this study, the top set of head valve shims were made up of a traditional triangle stack with five shims of varying diameter, while the bottom shim stack configuration was two shims of the same thickness and diameter. The head valve also has a removable bleed jet that allows the technician to adjust the bleed orifice diameter.
Below the head valve is the compression chamber, which is separated from the rebound chamber by the main piston. The main piston provides the majority of the damper’s dissipative force, and determines how this output force reacts to different inputs. The components that the main piston assembly is comprised of include: piston (type), rebound shim stack, compression shim stack, and bleed jet (needle valve), Figure 2-3. The piston type allows a technician to change the output force characteristics. Some of the common piston types, for this damper, and their force velocity plots are shown in Figure 2-4.
Having different rebound and compression shim stacks allows for asymmetric tuning of the damper. By changing the shim stack stiffness, the technician can change the damper’s output force. The bleed jet is threaded into the top of the damper’s shaft. The different types of bleed jets are open, compression, and rebound. Rebound bleed jets allow for flow only when the damper is in rebound, while compression bleed jets allow for flow when the damper is in compression. Open bleed jets allow for flow in both rebound and compression. The piston is secured to the damper’s shaft. At the end of the shaft there is the other high angularity spherical joint, which connects to the lower control arm of the racecar, and the external bleed adjuster, as shown in Figure 2-1. As the bleed adjuster turns, it moves in one click increments. Starting from zero, the bleed adjuster can move thirty clicks when turned clockwise. Each clockwise click moves an aluminum rod inside the damper’s shaft, up which in turn moves the needle valve. The bleed orifice is fully closed when the adjuster has been turned clockwise thirty clicks. With the knowledge of all of the parts which comprise the Penske 7300, the damper’s operation is presented in the next section.
2.3 Damper Operation Overview

As a racecar negotiates the race track, the track introduces many inputs to the damper. In addition to controlling roll, pitch, and heave motions, the damper must also control road inputs and aerodynamic forces. Not only is the damper tuned to control the inputs from the racecar, and the track, but it must also be tuned according to driver feel.

When the damper is compressed from an upward movement of the lower control arm, the shaft/piston assembly moves up into the shock body. This is considered the compression stroke. The opposite of this is the downward movement of the lower control arm that causes the piston/shaft assembly to extend out, which is the rebound stroke of the damper. The operation of the damper can be broken into its’ different flows. The different flows within the damper are described next.

Bleed orifice flow within the piston/shaft assembly can come from bleed holes in the piston, the adjustable bleed orifice in the shaft, or a combination of both. This is the primary flow for low speed inputs into the damper. Figure 2-5 shows an example of bleed orifice flow in the shaft during a rebound (or extension) stroke.

![Figure 2-5: Bleed Flow in the Main Piston During Rebound](image)
When the pressure difference between the compression chamber and the rebound chamber is great enough, the shim stack begins to flex open, and valve flow begins. Valve flow primarily occurs at higher input speeds, but if the bleed orifice area is small enough, or completely closed off, valve flow can also occur at a lower speed. Valve flow through the rebound shim stacks is shown in Figure 2-6 for a damper in rebound.

At the top of the damper body is the head valve. The head valve contains a removable bleed jet, which adjusts the bleed orifice area, and shim stacks on each side, that allow for asymmetric tuning of compression and rebound flows. Flow through the head valve during the damper’s compression and rebound strokes are shown in Figure 2-7. The black arrows show bleed flow, the red arrow shows compression shim stack flow, and the blue arrow shows rebound shim stack flow.

Figure 2-6: Valve Flow through the Rebound Shim Stack in the Main Piston
2.4 Damper Dynamometer and Test Procedures

An ideal characterization of an automotive damper would measure damper force and inputs while the damper is on the vehicle. Here, one could see the exact inputs from the tire and the damper’s reaction force to these inputs. Because this is not the most repeatable way to characterize a damper, a compromise is made, and that compromise comes in the form of a damper dynamometer.

In the laboratory setting, a damper dynamometer has been traditionally used to characterize dampers. The most common types of dynamometers used today are hydraulic or crank style dynamometers. These dynamometers are limited in their capabilities, hence a new style of dynamometer will be used for this study. This dynamometer uses Electro-Magnetic Actuation, and has the capability to play back race track data, as well as give precise control of user defined inputs.

All of the testing done on dampers for this study was completed on a Roehrig Electromagnetic Actuated (EMA) dynamometer (Figure 2-8). The EMA is capable of a 0.01 to 7 inch stroke, velocities over 120 inches per second, and frequencies up to 80 Hz (This is the manufacturer’s advertised limit. Its actual frequency range extends well above this). The EMA has a load cell, made by Interface, which is capable of 2000 lbs of...
force, Figure 2-9. Its’ linear position sensor is from Novotechnik, and linear velocity transducer is made by Trans Tek. The damper’s temperature is recorded with a Raytek infrared thermocouple.

Roehrig’s Shock 6.0 software is used to record testing data, and to export the data for use in Matlab. The EMA is capable of recording position, velocity, force, and temperature. In addition to standard tests such as sine wave and triangle waves, the Shock 6.0 can reproduce almost any input. This can be created in the software, or imported from a data file.

Figure 2-8: Roehrig Electromagnetic Actuated Damper Dynamometer
After one of the Penske 7300 dampers is built according to the rebuilding procedures in Appendix A, the damper is tested on the damper dynamometer. An outline of the testing procedure is as follows:

- The dynamometer and the computer are turned on and Shock 6.0 is started.
- The damper is hung from the top clevis and the load cell is zeroed.
- The damper is secured with the top and bottom clevises.
- Tests are constructed with prerecorded data or developed with the Shock 6.0 software.
- The test is run and the results are shown.
- The results are exported as a .csv file to be further analyzed in Matlab.
For the complete testing procedure, see Appendix B. With an understanding of the components of a damper, how it operates, and how it is tested, our focus shifts to experimental testing.
3 INVESTIGATION INTO DAMPER BEHAVIOR AT LOW AND HIGH FREQUENCIES

Now that the race damper’s components, operation, and testing methods have been shown, a fundamental look at damper behavior will be presented. After this fundamental look at damper behavior, we begin looking at results from experimental testing. This testing will show the influence input frequency has on the damper’s output force.

Modern automotive dampers are designed to provide maximum passenger comfort and acceptable handling qualities. The damper’s frequency dependence becomes important when considering noise, vibration, and harshness (NVH). Race dampers on the other hand are designed to achieve maximum handling capabilities, with a secondary design goal of driver comfort. In order to achieve maximum grip, these dampers must control the vibrations from the tire/wheel’s input, control weight transfer, and control aerodynamic force disturbances. The damper’s dependence on frequency plays an important role in how well the damper controls vibrations from the tire/wheel input, and ultimately, the maximum forces which the tire is able to achieve.

3.1 A Fundamental Look at Damper Behavior

In order to gain a better understanding of the fundamental operation of the race damper presented in this study, a fundamental look at theoretical damper operation is presented. This will allow us to better evaluate the experimental data presented later on in this chapter. A limit equation analysis will be presented, followed by a look at the response of a simple linear damper model.

3.1.1 Limit Equation Analysis

To begin our fundamental look at a damper, we will represent the monotube race damper used for this study as a spring and damper in series. The spring element will represent
the gas spring effect, due to the compressed nitrogen in the gas chamber. The damper element represents the damping forces generated by the main piston.

![Diagram of spring and damper in series](image)

**Figure 3-1:** Spring and Damper in Series

From Figure 3-1, the system’s transfer function is derived (the complete transfer function derivation is shown in Section 3.1.2). The magnitude of this transfer function in frequency domain is

\[
\left| \frac{F(j\omega)}{v} \right| = \frac{|ck|}{\sqrt{(c\omega)^2 + k^2}} \quad (3.1)
\]

With this representation of the system, we can look at three different cases. The first case is when one of the elements goes to zero, the second is when one of the elements goes to infinity, the third case is when one of the elements is much larger the other. Equations (3.2) and (3.3) represent the first and simplest case, one of the elements going to zero. If this occurred, in both cases, the output force would be zero. For the spring element to go to zero, the motion of the gas piston would have to be unrestricted. This situation would occur if the gas chamber was open to the atmosphere. For the damping element to go to zero, the main piston would have to be removed, so that there were no flow restrictions.
$k \to 0 \quad \left| \frac{F}{v}(j\omega) \right| = \lim_{k \to 0} \frac{|ck|}{\sqrt{(c\omega)^2 + k^2}} = 0 \quad (3.2)$

$c \to 0 \quad \left| \frac{F}{v}(j\omega) \right| = \lim_{c \to 0} \frac{|ck|}{\sqrt{(c\omega)^2 + k^2}} = 0 \quad (3.3)$

Limit equations (3.4) and (3.5) show the second case. As the spring coefficient, $k$, becomes infinitely large, the output force behaves like a damper alone. The opposite is true as the damping coefficient, $c$, goes to infinity. With infinite damping force, the system is without damping, and behaves like a spring. The input to the system’s transfer function is velocity and the output is force. The magnitude of a velocity input in frequency domain is equal to the magnitude of the displacement input times input frequency. Therefore, the omega will cancel out in equation (3.5) and the system’s output force will be a function of the spring coefficient and the displacement input.

For the spring element to become infinitely large, the initial gas pressure would have to be large enough to completely restrain the motion of the gas piston. If all of the flow restrictions at the piston were blocked, the damping element would become infinitely large, and the only force output would be from the gas spring (assuming the hydraulic fluid is incompressible).

$k \to \infty \quad \left| \frac{F}{v}(j\omega) \right| = \lim_{k \to \infty} \frac{|ck|}{\sqrt{(c\omega)^2 + k^2}} = |c| \quad (3.4)$

$c \to \infty \quad \left| \frac{F}{v}(j\omega) \right| = \lim_{c \to \infty} \frac{|ck|}{\sqrt{(c\omega)^2 + k^2}} = \left| \frac{k}{\omega} \right| \quad (3.5)$

Similar to equations (3.4) and (3.5), equations (3.7) and (3.9) show what happens if either the spring or damping element is much smaller than the other. If the spring element is much smaller than the damping element, then the system tends to behave like a spring. This is shown in equation (3.7). When the damping element is much smaller than the
spring element, the system can be represented by equation (3.9). For small input frequencies, the omega term can be eliminated and the spring coefficients will cancel out. In this case, the system will behave like a damper. Since one element is represented as much less than the other, these equations are a more realistic representation, of what could happen in the system, rather than one of the element becoming infinitely large or equal to zero.

\[
k << c \quad \left| \frac{F}{v} (j \omega) \right| \approx \frac{c^2 k^2}{c^2 \omega^2 + k^2} \approx \frac{k^2}{\omega^2} \quad (3.6)
\]

\[
\left| \frac{F}{v} (j \omega) \right| \approx \frac{k}{\omega} \quad (3.7)
\]

\[
c << k \quad \left| \frac{F}{v} (j \omega) \right| \approx \frac{c^2 k^2}{c^2 \omega^2 + k^2} \approx \frac{c^2 k^2}{\omega^2 + k^2} \quad (3.8)
\]

\[
\left| \frac{F}{v} (j \omega) \right| \approx \frac{ck}{\sqrt{\omega^2 + k^2}} \quad (3.9)
\]

### 3.1.2 Linear Response

The limit equation analysis shows the effect on the system if the damper or spring element is at an extreme. Our analysis continues, starting with the most basic representation of a damper, given in equation (3.10), and shown in Figure 3-2 where the damping coefficient, c, was chosen arbitrarily.

\[
F = cv \quad (3.10)
\]
Similarly for a spring, equation (3.11) represents the simplest model of a spring, and the force vs. displacement and force vs. velocity lissajous are shown in Figure 3-3. The spring constant, $k$, was chosen arbitrarily.

$$F = kx$$  \hspace{1cm} (3.11)
It is important to note that the lissajous in the force vs. velocity plots and force vs. displacement plots for the spring and damper are opposite of each other. When the damper’s lissajou is a straight line, the spring’s lissajou is more of an oval or circular shape. Knowing what the lissajous for the individual elements looks like will be helpful in observing the damper’s behavior transform from a damper to that of a spring as frequency increases.

Looking at a more realistic case than equation (3.10) or equation (3.11), development of a simple model of a race damper begins by representing it as a spring element and a damping element in series.

![Figure 3-4: Spring and Damper in Series](image)

Starting from first principles, the system’s forces are summed with

$$\sum F = ma \quad (3.12)$$

In this model, the system is assumed to be mass-less so this eliminates the inertia term

$$m\ddot{a} = 0 = -ky - c(\dot{y} - \dot{x}) \quad (3.13)$$
Combining and a little rearrangement gives

\[ c\ddot{y} + ky = c\ddot{x} \quad (3.14) \]

Now we begin arranging the equations in terms of the output force

\[ F = ky \quad \dot{F} = k\dot{y} \quad (3.15) \]

Multiplying equation (3.14) by \(k\) gives

\[ c(k\ddot{y}) + k(k\dot{y}) = ck\ddot{x} \quad (3.16) \]

Substituting in for \(F\) and dividing by \(k\) gives

\[ \frac{c}{k}\dot{F} + F = c\ddot{x} \quad (3.17) \]

Equation (3.17) in the Laplace domain is

\[ \frac{c}{k}SF + F = csx \quad (3.18) \]

Substituting in \(v\) for \(sx\) gives

\[ \frac{c}{k}SF + F = cv \quad (3.19) \]

Rearranging equation (3.19) gives a transfer function that relates the output force to the velocity input.

\[ \frac{F}{v} = \frac{ck}{cs + k} \quad (3.20) \]
Using equation (3.20), the linear system is analyzed in Matlab with the `lsim` command. The values for $c$ and $k$ are chosen so that the output force in this linear analysis comes close to the output force from the experimental testing. To begin this linear analysis, the observations from the limit equation are tested. Two cases are presented; the first case in Figure 3-5 shows the behavior of the damper spring system with a spring constant that is less than the damping coefficient. This configuration is comparable to limit equation (3.6). In the second case, the damping coefficient is less than the spring constant, which is similar to limit equation (3.8).

Figure 3-5 and Figure 3-6 show, much like the limit equations, the effect increasing the spring stiffness has on the damper’s output. In Figure 3-5, the spring constant is a tenth of the damping constant, and the damper and spring system behaves like a spring. The spring constant is now ten times that of the damping constant, in Figure 3-6, and the damper and spring system shows more of the characteristics of a damper than of a spring. These results are plotted with individual spring and damper elements in order to show each of the element’s behavior more clearly.

**Figure 3-5:** Output Force vs. Displacement and Force vs. Velocity with $c = 3$ and $k = 0.3$
In order compare the linear analysis to the forthcoming experimental results, the linear representation of the system is subjected to the same inputs as the damper in Section 3.2, Table 3-1. Values of the linear damping and spring coefficient were chosen to make the force vs. velocity and force vs. displacement plots have close to the same magnitude as the experimental results. This resulted in the damping coefficient, $c$, being set to 30 (force/(length/time)) and the spring constant, $k$, being set to 30,000 (force/length).

Table 3-1: Amplitude, Speed and Frequency of the Sinusoidal Inputs

<table>
<thead>
<tr>
<th>Amplitude</th>
<th>Speed</th>
<th>Frequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5 inches</td>
<td>9.425 inches/s</td>
<td>1 Hz</td>
</tr>
<tr>
<td>0.375 inches</td>
<td>9.425 inches/s</td>
<td>4 Hz</td>
</tr>
<tr>
<td>0.09375 inches</td>
<td>9.425 inches/s</td>
<td>16 Hz</td>
</tr>
<tr>
<td>0.046875 inches</td>
<td>9.425 inches/s</td>
<td>32 Hz</td>
</tr>
<tr>
<td>0.023475 inches</td>
<td>9.425 inches/s</td>
<td>64 Hz</td>
</tr>
</tbody>
</table>

To see the system’s output force transition more clearly, the 1 Hz case will be compared with all other frequencies that the system was subjected to. The first two figures, Figure 3-7 and Figure 3-8, show the system behaving like a damper, as expected. The force vs. velocity plot for the 1 Hz case and the 4 Hz case are almost identical, and the output force the lissajou reaches is the same although the displacements are different, Figure 3-7.
the 1 Hz verses 16 Hz case, Figure 3-8, the system’s lissajou for the 16 Hz input begins to show some discrepancy from the 1 Hz input. The force vs. displacement plot shows the same maximum output force with the different displacements.

**Figure 3-7:** Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 4 Hz

**Figure 3-8:** Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 16 Hz
The next two figures compare the 1 Hz input with a 32 Hz input and a 64 Hz input. The force vs. velocity plot in Figure 3-9 shows continued deviation from the 1 Hz case as frequency is increased. In the final comparisons, shown in Figure 3-10, the 1 Hz input is compared to the 64 Hz input. Here, the force vs. velocity plot shows the lissajou not only widening, but also shifting in a clockwise manor about the origin. The force vs. displacement plot shows that the output force for the 64 Hz input is not as great as for the 1 Hz input.

**Figure 3-9:** Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 32 Hz

**Figure 3-10:** Comparison of Linear Simulation Data of a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 64 Hz
Figure 3-7 through Figure 3-10 show a simple linear representation of the monotube race damper. Looking at these figures, it is apparent that as frequency increases, the damper’s output force behavior tends toward that of a spring.

### 3.2 Sinusoidal Input with Constant Velocity

After looking at the damper and spring element system, with a simple linear model, our focus shifts to experimental testing. There are a variety of ways and reasons to test dampers. For Dixon, dampers are tested to: “measure performance, check durability and to test technical analysis.” [5] One of the first individuals to perform damper testing was Weaver. He tested dampers with a free vibration input from a quarter car rig in 1929 [5].

Some of the more common types of damper testing include: Sinusoidal, Sine Sweep (Chirp), Sine on Sine (Double Sinusoid), Quasi – Static, Random, and Road Data. The type of input one chooses to test a damper with depends on what information about the damper you are trying to extract. The intent of this study is to investigate the how the damper’s behavior changes with frequency. In order to isolate the damper’s frequency dependency, all of the inputs for the experimental evaluation will have a constant peak velocity. Some of the benefits of constant peak velocity testing, according to Basso, is that the “energy distributed during motion may be quantified with greater precision and shock models may be validated with more accuracy.” [1]

Unlike modeling, which doesn’t always capture all of the dynamics of a system, controlled experimental testing allows the engineer to see first hand the system’s output for a given input. This section will first describe how the race damper was set-up for its’ experimental tests, and why these settings where chosen. After the damper’s setup is shown, the results of the experimental testing will be presented, and the damper’s dependence on input frequency will start to emerge.
3.2.1 Experimental Setup for a Sinusoidal Input with Constant Velocity

With the damper setup on the dynamometer as described in section 2.4, and Appendix B, the damper was tested, and the results obtained from these tests are presented. For all of the results in section 3.2 and 3.3, the damper was setup according to the configuration presented in Table 3-2. This damper setup is in the middle of the damper’s range of settings. A middle range damper setup with a linear piston was chosen because it gave a better representation of the effect the input to the damper has on the damper’s output force, and it can be accurately represented by a model. Figure 3-11 shows what the approximate damper output force will be for different shim stack letter designations.

<table>
<thead>
<tr>
<th>Thickness</th>
<th>Units</th>
<th>Constants</th>
<th>Letter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Head Valve Top Shims</td>
<td>0.008</td>
<td>inches</td>
<td>3</td>
</tr>
<tr>
<td>Head Valve Bottom Shims</td>
<td>0.004 x 2</td>
<td>inches</td>
<td>1</td>
</tr>
<tr>
<td>Rebound Shims</td>
<td>0.008</td>
<td>inches</td>
<td>1</td>
</tr>
<tr>
<td>Compression Shims</td>
<td>0.008</td>
<td>inches</td>
<td>3</td>
</tr>
</tbody>
</table>

**Table 3-2:** Damper Configuration for Experimental Testing

<table>
<thead>
<tr>
<th>Type</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Piston</td>
<td>Linear 1 deg./1 deg.</td>
</tr>
<tr>
<td>Bleed Jet</td>
<td>Rebound</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Diameter</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Head Valve Bleed Jet</td>
<td>0.015</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Pressure</td>
</tr>
<tr>
<td>Number of Clicks From Zero</td>
</tr>
</tbody>
</table>
3.2.2 Experimental Results

After the damper was setup, as described in section 3.2.1, the damper was tested on the Roehrig dynamometer. Table 3-3 gives the speeds, displacements, and frequencies at which the damper was tested. This section evaluates the dampers response at 1, 4, 16, 32, 64, and 128 Hz frequencies. The frequencies between these six did not show enough of a variation, so they are left out of this evaluation, but they do appear in Appendix C. For each of the given amplitude/frequency combinations, the damper was run on the dynamometer ten cycles. During these ten cycles, the damper’s temperature stayed between 107 and 110 degrees Fahrenheit. Test data was exported from Shock 6.0 and read into Matlab; the steady-state part of the data was found, by looking at plots of the data, and one of the steady state cycles was chosen for evaluation. Before looking at the data, an overview of the jargon used to evaluate force vs. velocity plots is presented. After that, force vs. displacement and the force vs. velocity plots for each input is evaluated. The final part of this section compares the 1 Hz input to the rest of the inputs, which will show how the damper’s output is affected as frequency is increased.
Table 3-3: Amplitude, Speed, and Frequency of the Sinusoidal Inputs

<table>
<thead>
<tr>
<th>Amplitude</th>
<th>Speed</th>
<th>Frequency</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5 inches</td>
<td>9.425 inches/s</td>
<td>1 Hz</td>
</tr>
<tr>
<td>0.75 inches</td>
<td>9.425 inches/s</td>
<td>2 Hz</td>
</tr>
<tr>
<td>0.375 inches</td>
<td>9.425 inches/s</td>
<td>4 Hz</td>
</tr>
<tr>
<td>0.1875 inches</td>
<td>9.425 inches/s</td>
<td>8 Hz</td>
</tr>
<tr>
<td>0.09375 inches</td>
<td>9.425 inches/s</td>
<td>16 Hz</td>
</tr>
<tr>
<td>0.046875 inches</td>
<td>9.425 inches/s</td>
<td>32 Hz</td>
</tr>
<tr>
<td>0.023475 inches</td>
<td>9.425 inches/s</td>
<td>64 Hz</td>
</tr>
<tr>
<td>0.0117375 inches</td>
<td>9.425 inches/s</td>
<td>128 Hz</td>
</tr>
</tbody>
</table>

When looking at a damper’s force vs. velocity plot, there is certain jargon used within the damper industry, specifically the race damper industry, to describe different parts of the lissajou. Figure 3-12 gives a graphical interpretation of this jargon. The nose area is defined as the low speed linear region in Figure 3-12, which includes velocities from 0 in/s to 0.5 in/s in this figure. The nose area corresponds to the bleed flow in the damper. When the damper lissajou starts to curve slightly, the valves are beginning to open; this area is referred to as the knee of the lissajou and is in the 0.5 in/s through 1.2 in/s velocity range. When the shims are open, the lissajou becomes linear again. Velocities greater than 1.2 inch/s is designated as the slope area of the lissajou. The slope angle of this lissajou is the stiffness of the shim stack in lbs/ (inch/s). This region is also referred to as the linear region. The final restriction to damper flow is the orifice diameter in the piston. When this becomes the limiting factor of damper flow, the slope region will begin to curve up; this is not shown in Figure 3-12.
Another term that is frequently used when describing a damper’s force vs. velocity lissajou is hysteresis. This term will be used extensively throughout this study, so a thorough description is necessary before it is used to describe test data. According to Burness and Nygren from Ohlins, hysteresis is the difference between the accelerating and decelerating parts of the lissajou [20], Figure 3-13. Hysteresis is normally related to the loss of energy in a system, but in a damper, hysteresis relates to energy that is being stored. This stored energy can be attributed to the compressibility within the damper. “The compressibility can be classified into three different groups: elasticity of the damper parts, compressibility of the oil itself and compressibility of the gas in the oil. Also the gas volume causes a hysteresis effect…” [20] According to [2,3,4,16,27], compressibility of the fluid is a major cause of hysteresis. Other studies attributed hysteresis to valve effects (valve flow and opening/closing) and friction [34,35,36]. Hysteresis is not desired in racing dampers (or any dampers), but is rather a problem because it delays the buildup of the output force.

Figure 3-12: Description of the Parts of a Damper’s Force vs. Velocity Lissajou
Now that we are familiar with the jargon used to describe the lissajous in the force vs. velocity plots and force vs. displacement plots, we can begin looking at the experimental data.

One standard frequency which dampers are tested at is 1 Hz. The displacement associated with this 1 Hz input is usually between 1 and 2 inches, for this study 1.5 inches was chosen. An input with a 1.5 inch displacement and 1 Hz frequency gave a velocity of 9.425 in/s. This velocity was held constant for the rest of this study. Figure 3-14 gives the baseline for which other single sinusoid plots will be compared to. The input was sinusoidal, with an amplitude of 1.5 inches, a velocity of 9.425 in/s, and a frequency of 1 Hz. Looking at Figure 3-14, the force vs. displacement plot shows the standard oval shaped lissajou. The negative output force/velocity region of the force vs. displacement plots and force vs. velocity plots is the damper’s rebound phase. Its compression phase is in the positive output force/velocity region. The peak force in Figure 3-14 is greater in rebound than compression. This is due to the fact that the bleed

![Figure 3-13: Illustration of Hysteresis Occurring over a Complete Sinusoidal Input Cycle. Note, in this Illustration, Compression is Shown with Negative Velocity and Rebound with Positive Velocity](image)
adjuster at the piston only allows flow in the rebound direction. Looking at the force vs. velocity plot, we can see the lissajou has a very pronounced nose in the first quadrant. This shows that the damper setup for this study did not have any bleed flow when the race damper was in compression. Once there was enough pressure to overcome the preload on the shims, the shim stack gradually opened and valve flow began. The slope region in this plot extends from 0 in/s to 9.425 in/s. In the third quadrant, the force vs. velocity curve shows a nose transitioning into a knee, and at roughly -2 in/s the knee region ends and the slope region begins. The slope region in rebound extends to the damper’s maximum velocity of -9.425 in/s. Notice that there is relatively little hysteresis in this plot. As the experimental testing continued, the amplitude was decreased, and the frequency was increased, while a constant peak velocity was maintained.

Figure 3-14: Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 1 Hz

Figure 3-15 is similar to Figure 3-14 with a slight increase in the amount of hysteresis in the nose/knee area in the first and third quadrants of the force vs. velocity plot.
As the amplitude was decreased to 0.09375 inches and the frequency was increased to 16 Hz, the force vs. velocity plot shows hysteresis over the lissajou’s entire range, Figure 3-16. Hysteresis increases in the force vs. velocity plot over the entire range of the lissajou, tested at 32 Hz, in Figure 3-17. In addition to this, the force vs. displacement plot shows the lissajou starting to shift up in the 2nd and 3rd quadrants and shift down in the 1st and 4th quadrants. Knowing that the lissajou in a force vs. velocity plot for a spring is oval in shape, and a lissajou in a force vs. displacement plot is a straight line, begs the question, if this trend continues, will there be a frequency at which the damper behaves more like a spring than a damper?
Figure 3-16:  Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 16 Hz

Figure 3-17:  Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 32 Hz

The next figure has an input with a 0.023475 inch amplitude and a 64 Hz frequency, and like all of the other inputs, this input also has a constant peak velocity of 9.425 inches/s. The lissajou in the force vs. displacement plot in Figure 3-18 shows significant distortion as the whole oval has been twisted 45 degrees clockwise. Hysteresis has caused the lissajou in the force vs. velocity plot to open up even more than in previous plots.
Figure 3-18:  Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 64 Hz

Figure 3-19 shows the race damper’s behavior at 128 Hz. The lissajou in the force vs. displacement plot is becoming very close to the linear relationship of a spring as in Figure 3-3. This trend is also evident in the force vs. velocity plot as the lissajou becomes more oval shaped, like that of a spring’s force vs. velocity plot.

Figure 3-19:  Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at a Frequency of 128 Hz
To better visualize how frequency affects the damper’s output, the 1 Hz input will be compared to inputs with 4 Hz, 16 Hz, 32 Hz, 64 Hz and 128 Hz frequencies. We begin these comparisons in Figure 3-20, which compares a 1 Hz input with a 4 Hz input. For these inputs, there is little difference in the damper’s output force for velocities greater in magnitude than 2 in/s. Between -2 in/s and 2 in/s there is a noticeable difference in the amount of hysteresis in the damper’s output force. There is an increase in the amount of hysteresis shown in Figure 3-21 which now shows hysteresis over the whole velocity range.

**Figure 3-20:** Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 4 Hz
The amount of hysteresis in the damper’s output force increases in the 1 Hz vs. 32 Hz case, shown in Figure 3-22, and the 1 Hz vs. 64 Hz case, Figure 3-23.

Figure 3-21: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 16 Hz

Figure 3-22: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 32 Hz
Figure 3-23: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 64 Hz

Figure 3-24 shows the output force for a 1 Hz input and a 128 Hz input. Here in the force vs. velocity and force vs. displacement plots, the lissajou for the 128 Hz input has lost the characteristic shape of a damper’s lissajou.

Figure 3-24: Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 128 Hz

40
To make a final case on the effect increasing input frequency has on a race damper’s output force, Figure 3-25 and Figure 3-26 compare 1 Hz and 32 Hz inputs with first a 64 Hz input and then a 128 Hz input.

**Figure 3-25:** Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 32 Hz and 64 Hz

**Figure 3-26:** Comparison of Experimental Data for a Sinusoidal Input with Constant 9.425 inches/s Peak Velocity, at Frequencies of 1 Hz and 32 Hz and 128 Hz
Figure 3-20 through Figure 3-26 have shown the damper’s output force transition from the characteristic shape one would expect of a damper, to a shape which resembles the output force of a spring. Now that we have seen the effect a constant peak velocity sinusoidal input has on the damper’s output force as the frequency is increased, we will shift our focus to another type of input, a sinusoidal sweep input.

### 3.3 Sine Sweep Input

The input described in section 3.2 was a constant peak velocity sinusoidal input at six discrete frequencies. The experimental results showed how the damper behaves as these discrete input frequencies were increased. To further investigate the damper’s dependence on frequency over a broader range of frequencies, a sinusoidal sweep input with a constant peak velocity was used. This input excites the damper over an entire range of frequencies rather than just discrete points, and exciting the damper at a constant peak velocity allows us to look at just the effects of frequency.

For this sinusoidal sweep input, the damper was setup as described in Section 3.2.1, and was tested on the Roehrig dynamometer. The frequency range tested was from 1 to 100 Hz. This frequency range was broken into 10 Hz segments for better overall resolution and to control the temperature of the dampers. Each of these segment’s test lasted thirty seconds, during which the damper’s temperature stayed between 107 and 110 degrees Fahrenheit. Test data was then exported from Shock 6.0 and read into Matlab. Plots generated from these tests are shown in Figure 3-27 through Figure 3-35. The results shown are only for some of the frequency segments, all of the results are shown in Appendix C.

It is important to keep in mind the following when looking at the force vs. velocity plots and force vs. displacement plots that follow. As frequency increases, in order to maintain a constant velocity, displacement decreases, so in the force vs. displacement plots the outer most part of the lissajou corresponds to the lower frequency part of the sweep, whereas the inner part of the lissajou corresponds to the high frequency part. The reverse
of this is true for the lissajou in the force vs. velocity plots. Here the inner portion is from the low frequency part and the outer is from the high frequency part. This is shown in Figure 3-27.

In all of the following plots, the lissajous have some thickness associated with them. This is due to the changing frequency. Looking at Figure 3-27, the lissajou in the force vs. velocity plot shows behavior that would be expected of a linear damper.

![Figure 3-27: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz](image)

Figure 3-27 through Figure 3-29 show the damper’s behavior transitioning away from that of a damper. The lissajou in the force vs. displacement plots starts to rotate counterclockwise and once it has reached a certain point, starts to compress inward. The lissajou in the force vs. velocity plots transitions away from its characteristic shape, and starts to smooth out and become more oval like.
Figure 3-28: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 40 to 50 Hz

Figure 3-29: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 90 to 100 Hz

Although Figure 3-27 through Figure 3-29 do a good job of showing the damper’s transition as frequency is increased, the point is made even clearer in Figure 3-30 through Figure 3-33. Here the 1 – 10 Hz set of data is compared to the 20 – 30, 40 – 50, 70 – 80, and 90 – 100 Hz data sets.
Figure 3-30: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz vs. 20 to 30 Hz

Figure 3-31: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz vs. 40 to 50 Hz
Because the displacement data is an order of magnitude greater for the 1 – 10 Hz case than the 90 – 100 Hz case, it is difficult to see the transition of the lissajou in the force vs. displacement plot as shown in Figure 3-33. To make that transition clearer, Figure 3-34 shows the same force vs. displacement plot as in Figure 3-33, but the displacement is normalized. This is done by dividing the two displacement data segments by the maximum displacement of their respective segment.
Figure 3-34 shows the lissajou rotating clockwise (red dashed arrows) and compressing in on itself (red arrows) as the frequency is increased. As frequency is increased, the lissajou compresses together and takes on more of the characteristics of the linear relationship of a spring.

![Sinusoidal Sweep, Constant Velocity, Normalized Displacement, Frequency Ranges: 1-10 Hz vs 90-100 Hz](image)

**Figure 3-34:** Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, Normalized Displacements, and a Frequency Range from 1 to 10 Hz vs. 90 to 100 Hz

The transition of the lissajou in the force vs. velocity plot in Figure 3-33 is apparent without normalizing the input velocity, but a more visual view of this transition is given in Figure 3-35. Like the lissajou in the force vs. displacement plot the lissajou in the force vs. velocity plot also rotates clockwise as frequency is increased (red dashed arrows). This lissajou opens up, and takes on more of an oval shape as frequency is increased (red arrows). Observing these changes in the lissajou show that as the input frequency is increased the damper takes on characteristics of a spring, which has an oval shaped lissajou in a force vs. velocity plot.
3.4 Results and Discussion

Starting with a simple linear model, and augmenting it with experimental data, this chapter has shown how the damper’s output force is affected by frequency. The experimental data was collected using a race damper with settings that are in the middle of the damper’s range of settings. Experimental results show that as frequency is increased for an input with a constant peak velocity, the damper’s output force takes on the characteristics of a spring. This is best observed in the force vs. velocity plot’s lissajou which starts to become more oval shaped as frequency is increased. This oval shaped lissajou in a force vs. velocity plot is characteristic of a spring. The conclusions that were made from the experimental testing were confirmed by simple linear analysis which showed the same results in the output force as frequency increased.

The results from this chapter agree with previous studies in that frequency has an effect on a damper’s output force. One of the first studies done on how frequency affects a damper’s output force was done by Lang in his dissertation. In his study, he developed a
parametric model, validated it over a 1-10 Hz range, and used it to look at a damper’s dependence on frequency. Lang kept his maximum velocity to 30 in/s, and limited his study to 1 – 50 Hz [16,27]. Some of his main findings were:

- “That it is not the generation of vapor (at the vapor pressure of the fluid) that accounts for the lower pressure limit in the chambers, but rather the expansion of gas entrapped in the liquid as the pressure approaches zero absolute” [16,27].

- Compressibility of the fluid is a major cause of hysteresis [16,27].

Lang’s paper, which he coauthored with his advisor, Segal, draws the same conclusions [27]. Cafferty also looked at potential causes of frequency dependence and concluded:

- Rubber bushes only introduce displacement dependence to the system, not frequency dependence [2,3,4].

- “Cavitation has little contribution to the output force effects” [2,3,4].

- “Compressibility of the damper oil is the major cause of the shock absorbers frequency dependence” [2,3,4].

Recent studies were done by Yung in 2003 which attributed hysteresis to valve effects (valve flow and opening/closing) and friction [34,35,36]. With the first part of our experimental testing complete, a parametric model was developed that allows the user to see how damper parameters affect the amount of hysteresis in the damper’s output force.
4 HIGH FREQUENCY BEHAVIORAL STUDY UTILIZING A PARAMETRIC DAMPER MODEL

4.1 Model Development

To gain a better understanding of a damper’s performance characteristics, a technician must test a damper on a dynamometer to characterize the output force for a given input and damper setting. To reduce the time it takes to physically test a damper, a parametric damper model is developed to run virtual tests. Here, any input and setting can be changed in a matter of seconds, and the damper’s output force can easily be characterized. This helps define what parameters cause the most significant change, and give the technician a better starting point selecting settings; which can be studied further on the dynamometer. With this in mind, development of a model of the Penske 7300 begins.

By far, the most cited study on damper modeling was conducted by Lang in 1977. He developed a model of a twin tube damper that consisted of 82 parameters, and was validated for a 1 – 10 Hz frequency range [16,27]. The model was programmed into an analog computer, and parameters such as “dynamic discharge coefficient and valve operating force were determined experimentally” [16,27]. With a validated model, Lang conducted a parameter study to see how the damper parameters affected its performance. One of the downfalls of Lang’s model is its complexity. To address this issue, Reybrouck made a simplified model in 1994 which was valid from 0.5 to 30 Hz [23]. Warner developed a damper model, and quarter car model in 1996. Together, he used the damper and quarter car models to study how variations in damper parameters affect suspension performance [32]. In 2000 Duym developed a model for the BMW series 7 damper. The interesting aspect of this model is that it was developed so that all of the model’s coefficients could be measured on a dynamometer with two tests; a quasi-static compression test, and a dynamometer run [7]. Besides parametric models, there are a
number of studies which look at nonparametric modeling of dampers. Major studies include:

- Cafferty, who used higher order frequency response functions [2,3,4].
- Duym, who looked at damper modeling with a force state map [6].
- Rao, who looked at different testing methods to get empirical coefficients [15,24,25].

One damper model that closely relates to the model developed for this study is the one which Talbott developed to characterize an Öhlins race damper in 2002 [29,30]. This model also uses an Öhlins race damper as its platform, but this model is not used for anything other than characterizing the damper. An added feature of the model developed for this study which was not in Talbott’s model, is the addition of a head valve. Head valves were not a component in the damper Talbott modeled, but they are part of the Penske 7300 used in this study. The model in this study also uses results from previous studies to simplify the final model.

To characterize the output force of a damper, the collection of forces which affect the output force must be established. Figure 4-1 shows a free body diagram of the piston/shaft assembly. The damper is subjected to a known input through the shaft, and the reaction force is measured. In the physical world, a load cell measures the dampers output force to a given input from a dynamometer. In the virtual world, we subject the model to a known displacement, velocity, and acceleration, and calculate the output force.

![Figure 4-1: Free Body Diagram of Piston and Shaft Assembly](image-url)
From the free body diagram, the equations of motion are

\[ m_p \ddot{x} = F + p_r A_r - p_c A_c - F_f \]  

(3.21)

The inputs to the damper model are known \((x, \dot{x}, x)\), and the friction force \((F_f)\) can be measured on the dynamometer. To solve for the output force \((F)\), the pressures in the different chambers must be determined. The pressures in the head valve chamber and the gas chamber can be calculated directly from the input, while the pressures in the compression \((p_c)\) and rebound chambers \((p_r)\) are determined from the flows between the chambers. Looking at the gas chamber and the head valve chamber first, the derivation will start by relating the initial gas chamber pressure \((p_{gi})\) and the current gas chamber pressure \((p_g)\) to pressure in the head valve chamber \((p_{hv})\).

Starting with ideal gas law, and assuming an isothermal process, equation (3.22) can be derived by relating the initial volume \((A_{gp}L_g)\) in the gas chamber to the change in gas chamber volume as the shaft moves within the damper body.

\[ p_g = p_{gi} \frac{A_{gp} L_g}{A_{gp} L_g - A_{rod} x} \]  

(3.22)

To capture the motion of the piston in the gas chamber, we start with a free body diagram. The damper fluid is assumed incompressible, and the friction forces at the gas piston and the seal on the shaft are described by the overall friction at the piston \((F_f)\).

Figure 4-2: Free Body Diagram for Gas Piston
The equation of motion from the free body diagram in Figure 4-2 is given by

\[ m_{gp} \ddot{z} = (p_{hv} - p_g) A_{gp} \]  

(3.23)

The gas piston must move in relation to the volume of fluid the shaft displaces \( (A_{rod} \dot{x}) \), since the damper fluid is assumed incompressible. From this, the relationship between the acceleration of the gas piston \( (\ddot{z}) \) and shaft can be shown to be

\[ \ddot{z} = \frac{A_{rod}}{A_{gp}} \dot{x} \]  

(3.24)

Combining equations (3.23) and (3.24), and solving for the head valve pressure gives

\[ p_{hv} = \frac{A_{rod} m_{gp}}{A_{gp}^2} \ddot{x} + p_g \]  

(3.25)

Combining equations (3.22) - (3.25), gives the total head valve pressure

\[ p_{hv} = \frac{A_{rod} m_{gp}}{A_{gp}^2} \ddot{x} + p_g + \frac{A_{gp} L_g}{A_{gp} L_g - A_{rod} \dot{x}} \]  

(3.26)

The total valve flow rate, \( Q \), is derived from the area of rebound side of the piston times the input velocity.

\[ Q = A_r \dot{x} \]  

(3.27)

The total flow rate is made up of individual flow rates, including: flow through the valves \( (Q_v) \), the bleed orifices \( (Q_b) \), and leakage past the piston. Previous work suggests that the leakage past the piston is insignificant, so all of the piston flows in this model are accounted for in equation (3.28) [29,30].

\[ Q = Q_v + Q_b \]  

(3.28)

Previous work also suggests that losses across the piston orifice are insignificant for a variation in the orifice area of less than fifty percent of the original area [29,30]. Using
this assumption, the orifice pressure difference can be eliminated from the model, and the pressure difference across the valve will be the same as the pressure difference across the piston.

\[ \Delta p_o = \Delta p_v = \Delta p \]  

(3.29)

The deflection of the shim stack (y) is determined from a free body diagram of the shim stack, shown in Figure 4-3. The summation of the applied forces yields equation (3.30) which includes the preload on the shim stack (F_{sp}), the pressure times the shim stack area (\Delta p A_v), the shim stack stiffness times the deflection (ky), and the momentum change in the fluid (F_m) (Figure 4-3). The fluid momentum term is found from the conservation of momentum across the valve, and is made up of the fluid density (\rho), the orifice area (A_0), the valve flow rate (Q_v), and an empirical flow coefficient (C_f).

![Free Body Diagram for Shim Stack Stiffness.](image)

\[ ky = \Delta p A_v + \rho \frac{Q_v^2}{A_0} C_f - F_{sp} \]  

(3.30)

The area which the fluid acts upon is assumed one half of the total annular area, due to the area the piston orifices act on the shims [29,30]. The results from Lang [16] showed that Bernoulli’s equation could be used to describe unsteady flow through a constant area. Lang used a dynamic discharge coefficient, C_D, rather than a steady-state discharge coefficient, to be able to apply Bernoulli’s equation to unsteady flow [16,29,30].
\[ Q_v = \frac{1}{2} \pi D_v y C_p \sqrt{\frac{2\Delta p}{\rho}} \]  \hspace{1cm} (3.31)

Bleed flow in the piston is adjusted with either bleed holes in the piston, or the bleed hole in the shaft. A technician adjusts the bleed orifice area in the shaft with the adjuster, located at the bottom of the shaft.

Applying Bernoulli’s equation again, the bleed orifice flow is calculated by

\[ Q_b = A_b C_p \sqrt{\frac{2\Delta p}{\rho}} \]  \hspace{1cm} (3.32)

The pressure variation across the piston is given by

\[ \Delta p = \begin{cases} p_c - p_r, & \text{compression} \\ p_r - p_c, & \text{rebound} \end{cases} \]  \hspace{1cm} (3.33)

During the damper’s compression stroke, the pressure in the rebound chamber is found by

\[ p_r = p_c - \Delta p \]  \hspace{1cm} (3.34)

The total flow rate through the head valve, \( Q_{hv} \), is the area of the shaft, \( A_{rod} \), times the velocity of the shaft. The parameters in the derivation below have the same meaning as before, but are distinguished as head valve parameters by a subscript “hv”.

\[ Q_{hv} = A_{rod} \dot{x} \]  \hspace{1cm} (3.35)

The flow through the head valve is comprised of the head valve bleed orifice flow, and the valve flow from the deflection of the head valve shim stacks.

\[ Q_{hv} = Q_{vHV} + Q_{bHV} \]  \hspace{1cm} (3.36)

Equation (3.37), for valve flow in the head valve, is derived the same as equation (3.31).
\[ Q_{ohv} = \frac{1}{2} \pi D_{ohv} y_{hv} C_{D_{ohv}} \sqrt{\frac{2\Delta p_{hv}}{\rho}} \quad (3.37) \]

Equation (3.38) gives the deflection of the shim stack in the head valve.

\[ k_{hv} y_{hv} = \Delta p_{hv} A_{shv} + \rho \frac{Q_{hv}^2}{A_{ohv}} C_{fhw} - F_{sphv} \quad (3.38) \]

Bleed flow in the head valve is symmetric, and is adjusted by changing the bleed jet to vary the bleed hole diameter. The bleed orifice flow in the head valve is described by

\[ Q_{bhv} = A_{bhv} C_{D_{bhv}} \sqrt{\frac{2\Delta p_{hv}}{\rho}} \quad (3.39) \]

Equation (3.40) gives pressure variation across the head valve.

\[ \Delta p_{hv} = p_{c} - p_{hv} \quad (3.40) \]

The final model contains two sets of four coupled nonlinear equations, four input equations, and the force balance equation to solve for the output force. Within the model, there is a switching condition to determine if the flow is only through the bleed orifice or through the valves as well. For just bleed flow, a known input into the model is used with equations (3.27) and (3.28) to solve for the bleed flow rate. With this, the pressure difference across the piston is solved with equation (3.32). From this, \( p_{c} \) is solved with equation (3.34) once \( p_{c} \) is determined from the flow equations for the head valve. To find \( p_{c} \), equation (3.26) and the known input are used with equations (3.35), (3.36), (3.39), and (3.40).

Valve flow starts with the same known input to the model, and equation (3.27) is used to solve equations (3.28), (3.30), (3.31) and (3.32). The solution to this gives the pressure variation across the piston (\( \Delta p \)). This along with equation (3.34) gives the pressure \( p_{r} \) once \( p_{c} \) is solved. The known input and equation (3.35) are used to solve equations (3.38), (3.39), (3.40). With this solution, the pressure variation across the head valve (\( \Delta p_{hv} \)) is known, and in combination with equation (3.26), the pressure in the
compression chamber is determined. With the pressure in the compression chamber, the pressure in the rebound chamber, the mass of the piston, and the friction force, equation (3.21) is used to calculate the output force of the damper. A program was written in Matlab using the fsolve function to solve the nonlinear equations in the model. Here, the model is developed to the same point that it was for its use in a International Mechanical Engineering Conference and Exposition (IMECE) paper and presentation. In this work [9], a parameter study was conducted to show the effect of parameters that a technician would tune, on the damper’s output force. Parameters include in this IMECE study were initial gas pressure, bleed orifice diameter, and shim stack stiffness at the main piston and head valve.

In order to tune the model for a wider range of frequencies, rather than just a single frequency, the model was tuned using the sinusoidal sweep data from Section 3.3. Plots of the 1 – 10 Hz input’s raw output force data are shown in Figure 4-4, and the filtered versus raw data is shown in Figure 4-5. The data was filtered with a fourth order Butterworth filter with a break frequency of 50 Hz. The raw data verses the filtered data for the other segments is similar to this data segment, so the other segment’s raw data is not included.

![Figure 4-4: Experimental Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz](image-url)
Figure 4-5: Experimental Data and Filtered Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz

With the code running correctly in Matlab, the model was compared to experimental data and the model parameters were tuned to the data. Model tuning is discussed more in Section 4.2, but before moving on to discussing that, one problem that arose with the model needs to be presented. With a correctly tuned model, the model matches the experimental data up to around 5 Hz. This would be acceptable if we were only concerned with damper behavior at lower frequencies, but the premise of this paper is to consider how the damper’s behavior changes as the input frequency increases. At frequencies higher than 5 Hz, the model gives a good representation of the backbone of the force vs. velocity lissajou, but does not capture the hysteresis in the data, shown in Figure 4-6 and also Figure 4-7. The root mean square (RMS) error and average absolute percent error for the experimental output force data vs. the model output force data is shown in Table 4-1. Equation devolvement for the RMS error and the average absolute percent error is presented in Section 4.2
Table 4-1: Experimental Data vs. Model without Warping Filter

<table>
<thead>
<tr>
<th>Section</th>
<th>RMS Error, lbs. Force</th>
<th>Average ABS % Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 - 10 Hz</td>
<td>14.00</td>
<td>5.22</td>
</tr>
<tr>
<td>40 - 50 Hz</td>
<td>86.34</td>
<td>39.4</td>
</tr>
<tr>
<td>90 - 100 Hz</td>
<td>192.79</td>
<td>101.1</td>
</tr>
</tbody>
</table>

Looking back at the model, it was determined that not incorporating the compressibility of the fluid, temperature effects, and the effects of cavitation/gas trapped in the hydraulic fluid caused the model to have a limited sensitivity to the input frequency. Due to time constrictions on this project, we determined that rather than incorporating all of the hysteretic effects into the model, that adding a filter to simulate these effects would suffice for the parameter study which this model was developed for.

Figure 4-6: Experimental Force Data vs. Model Force Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz
One way to incorporate the effects of hysteresis into a damper model is to add lag to the system. This can be achieved with a first order filter with filter parameters tuned to experimental data. The problem of just adding a first order filter is that the lag is distributed equally throughout the whole range of the model’s output force. To incorporate hysteresis into the desired parts of the model’s output force, a warping filter was developed. This filter consists of a warping function, a first order lag, and an inverse warping function, Figure 4-8.

Figure 4-8: Warping Filter which Adds Hysteresis to the Model’s Output Force

Figure 4-9 shows a force vs. velocity plot with an idealized output force backbone curve $F(v)$. This representation leaves out the knee portion of the curve described in Chapter
Three. For this, force vs. velocity lissajou, the nose section is between $v = 0$ and $v = v_o$, and the slope section of the lissajou occurs when $v$ is greater than $v_o$. The warping function is developed with $v_o$ (the nose velocity), $F_o$ (the output force at the nose) and $m$ (the angle of the slope in the slope region). The parameters described are for a warping function which is symmetric in compression and rebound. The warping function used in the final model considers asymmetric lissajous, and incorporates $v_o$, $F_o$, and $m$ in both compression and rebound. The blue axis in Figure 4-9 shows the magnitude of the warping function. When the velocity is between $-v_o < v < v_o$, the gain of the warping function is unity, otherwise it is a decreasing value depending on the lag in the warping filter. The development of the warping function continues on the next page.

![Diagram](image)

**Figure 4-9:** An Ideal Output Force Backbone $F(v)$ is Shown in Black, and the Warping Function $g(v)$ is Shown in Red
When \(-v_o < v < v_o\),

\[ g(v) = 1 \quad (3.41) \]

For \(v_o < v\),

From Figure 4-8

\[ z = F_o = g(v)y \quad (3.42) \]

The equation for \(F(v)\) beyond \(v_o\) is

\[ F - F_o = m(v - v_o) \quad (3.43) \]

Since \(y = F\), rearranging Equations (3.42) and (3.43) gives

\[ g(v) = \frac{F_o}{F} = \frac{F_o}{F_o + m(v - v_o)}, \quad g(v) > 0 \forall v \quad (3.44) \]

The equations for \(-v_o > v\) are solved for in the same manner. Figure 4-9 shows an idealized force vs. velocity lissajou, but the points \((v_o, m, \text{and } F_o)\) can easily be extracted from the experimental output force data, and consequently a backbone emerges, Figure 4-10. With the warping function complete, the warping filter is added to the model’s Matlab code, and the parameters are tuned to the experimental data to incorporate hysteresis into the model’s output force, Figure 4-11 through Figure 4-13.
**Figure 4-10:** Warping Filter Parameters Found from Experimental Output Force Data for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz

**Figure 4-11:** Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz
**Figure 4-12:** Experimental Force Data, Model Force Data, and Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz

**Figure 4-13:** Experimental Force Data, Model Force Data, and Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz
4.2 Model Tuning

Of the twenty-eight model parameters, four parameters for the main piston and the head valve (k, F_{sp}, C_{D}, and A_b) proved difficult to measure directly, both in rebound and compression. A program was written in Matlab utilizing the fmincon function to fit these parameters to experiential data, by minimizing the error between the experimental data and the model prediction. After completing the optimization coding, and using it to tune the model, it was found that it was faster to change the parameters manually. This not only gave results faster, but also was useful in observing the effect the parameters have on the force output of the model. This method was used for the rest of the model tuning.

The model and warping filter parameters were tuned for three segments, 1 – 10, 40 – 50, and 90 – 100 Hz, for the parameter study to follow in the Section 4.3. The tuned model is compared with the experimental output force data, and is shown in the figures that follow. Figure 4-14 shows the error (F_{data} – F_{hyst}) for the 1 – 10 Hz segment, and the model’s output force versus the experimental output force. The RMS error, Equation (3.46), was used to compare the difference in output force between the experimental data and the model.

\[
Error = F_{data} - F_{hyst} \quad (3.45)
\]

\[
Error_{RMS} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} Error_i^2} \quad (3.46)
\]

The RMS error for this data segment is 6.52 lbs force. The RMS errors for all of the segments are presented in Table 4-2. Figure 4-15 shows the force vs. displacement plot, and the force vs. velocity plot for the experimental and modeled output force for the 1 – 10 Hz case.
Figure 4-14: Error vs. Time and Experimental Force Data, and Model Force vs. Time for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz

Figure 4-15: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 1 to 10 Hz

Figure 4-16 shows the error for the 40 – 50 Hz segment and the model’s output force versus the experimental output force. The RMS error for this data segment is 7.96 lbs force. Figure 4-17 shows the force vs. displacement plot and the force vs. velocity for the experimental and modeled output force for the 40 – 50 Hz segment.
Figure 4-16: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 40 to 50 Hz

Figure 4-17: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 40 to 50 Hz

Figure 4-18 shows the error for the 90 – 100 Hz segment and the model’s output force versus the experimental output force. The RMS error for this data segment is 7.60 lbs force. Figure 4-19 shows the force vs. displacement plot and the force vs. velocity plot for the experimental and modeled output force for the 90 – 100 Hz segment.
Figure 4-18: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 90 to 100 Hz

Figure 4-19: Experimental Force Data vs. Model Force Data with Hysteresis for a Sinusoidal Sweep with Constant 9.425 inches/s Peak Velocity, and a Frequency Range from 90 to 100 Hz

The average absolute percent error was calculated using Equation (3.47).

\[
\text{Average Absolute Percent Error} = \frac{\text{Avg} \left( \left| \text{Error} \right| \right)}{\text{Avg} \left( \left| F_{data} \right| \right)} \times 100
\]  

(3.47)
Table 4-2: Experimental Data vs. Model Output Force with Warping Filter

<table>
<thead>
<tr>
<th>Section</th>
<th>RMS Error, lbs. Force</th>
<th>Average ABS % Error</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 - 10 Hz</td>
<td>8.47</td>
<td>3.3</td>
</tr>
<tr>
<td>40 - 50 Hz</td>
<td>9.91</td>
<td>4.4</td>
</tr>
<tr>
<td>90 - 100 Hz</td>
<td>8.87</td>
<td>4.5</td>
</tr>
</tbody>
</table>

All of the model parameters can be found in Appendix C, parameters which were tuned for the different frequency segments are shown in Table 4-3. It is important to note that one parameter set which accurately captures the damper’s output force over the entire 1 – 100 Hz range was not identified.

Table 4-3: Tuned Parameters for 1 – 10 Hz, 40 – 50 Hz, and 90 – 100 Hz

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Frequency Range</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>k,</td>
<td>1 - 10 Hz</td>
<td>lbs/in</td>
</tr>
<tr>
<td>k,</td>
<td>40 - 50 Hz</td>
<td>lbs/in</td>
</tr>
<tr>
<td>k,</td>
<td>90 - 100 Hz</td>
<td>lbs/in</td>
</tr>
<tr>
<td>A_b,</td>
<td>1 - 10 Hz</td>
<td>in²</td>
</tr>
<tr>
<td>A_b,</td>
<td>40 - 50 Hz</td>
<td>in²</td>
</tr>
<tr>
<td>A_b,</td>
<td>90 - 100 Hz</td>
<td>in²</td>
</tr>
<tr>
<td>v_o,</td>
<td>1 - 10 Hz</td>
<td>in/s</td>
</tr>
<tr>
<td>v_o,</td>
<td>40 - 50 Hz</td>
<td>in/s</td>
</tr>
<tr>
<td>v_o,</td>
<td>90 - 100 Hz</td>
<td>in/s</td>
</tr>
<tr>
<td>F_o,</td>
<td>1 - 10 Hz</td>
<td>lbs</td>
</tr>
<tr>
<td>F_o,</td>
<td>40 - 50 Hz</td>
<td>lbs</td>
</tr>
<tr>
<td>F_o,</td>
<td>90 - 100 Hz</td>
<td>lbs</td>
</tr>
<tr>
<td>a</td>
<td>1 - 10 Hz</td>
<td>1X10⁻².⁷⁵</td>
</tr>
<tr>
<td>a</td>
<td>40 - 50 Hz</td>
<td>1X10⁻².⁸²</td>
</tr>
<tr>
<td>a</td>
<td>90 - 100 Hz</td>
<td>1X10⁻².⁷⁷</td>
</tr>
<tr>
<td>m,</td>
<td>1 - 10 Hz</td>
<td>195/10</td>
</tr>
<tr>
<td>m,</td>
<td>40 - 50 Hz</td>
<td>50/10</td>
</tr>
<tr>
<td>m,</td>
<td>90 - 100 Hz</td>
<td>60/10</td>
</tr>
<tr>
<td>m,</td>
<td>1 - 10 Hz</td>
<td>350/10</td>
</tr>
<tr>
<td>m,</td>
<td>40 - 50 Hz</td>
<td>50/10</td>
</tr>
<tr>
<td>m,</td>
<td>90 - 100 Hz</td>
<td>60/10</td>
</tr>
</tbody>
</table>
4.3 Model Study on Parameter Variation and its Effect on Damper Output Force as Frequency Increases

With the model tuned for the 1 – 10, 40 – 50, and 90 – 100 Hz frequency ranges, it is desired to use the tuned parametric model to predict how model parameters affect the amount of hysteresis in the damper’s output force as frequency is increased. To see how a parameter affects the amount of hysteresis in the physical damper, we would test it with the parameter at one value over a range of frequencies, say 1 – 100 Hz. Then, we would increase (or decrease) the value of the physical parameter and test the damper again. Comparing the amount of hysteresis in test A versus test B would show this parameters influence on the amount of hysteresis in the damper’s output force.

If our model had one set of parameters for the entire 1 – 100 Hz frequency range, the virtual test would be conducted in a similar manner, with the testing done by a computer rather than the damper dynamometer. Since our model does not allow for this, testing to look at the model parameter’s influence on hysteresis was conducted using a different approach. The idea was to use the three parameter sets for the 1 – 10, 40 – 50, and 90 – 100 Hz frequency ranges, and vary a given parameter between 75 percent and 125 percent of that parameter’s tuned value. The force difference for the upper and lower percent change would be recorded, and these force differences would be compared for the three frequency segments. Similar to the testing situation described in the previous paragraph, observing an increase in the force difference would relate to an increase in hysteresis caused by the change of the varied parameter. So as not to incorporate any unwanted effects into the model’s output force, experimental data was not used as the model’s input for this parameter study. A sinusoidal input was coded into the model, and the frequencies chosen for the inputs were the lower frequencies of the tuned parameter sets. For the 1 – 10 Hz parameter set, the single sinusoidal input frequency was 1 Hz. The other frequencies this parameter study considered were 40 and 90 Hz, and the corresponding parameter set was utilized.

With a method to evaluate a parameter’s influence on the amount of hysteresis in the dampers output force, a parameter study was conducted, and the results are shown in
Figure 4-20 through Figure 4-27 and Table 4-4 through Table 4-6. The model in this study was used for an extensive parameter study in [9], so there was already an idea of what parameters were likely to influence the damper’s output force the most as frequency is increased. The parameters chosen for this study are also parameters which can be changed on the physical damper, which will allow the results of this model study to be validated with experimental data. The three parameters chosen for this study include: initial gas pressure, friction, and fluid density.

For each of the input frequencies, the RMS difference between the output forces will be calculated for the nominal parameter versus the changed parameter. These values, along with the percent difference, will be shown in tables at the end of each parameter study.

\[
\text{Difference} = F_{100\%} - F_{75\%} \quad \text{Difference} = F_{100\%} - F_{125\%}
\]  

\[
\text{Difference}_{RMS} = \sqrt{\frac{1}{N} \sum_{i=1}^{N} \text{Difference}_{i}^2}
\]

The percent difference is calculated and shown as follows. \(D_{\text{max}}\) is the maximum absolute difference between the two lissajous for one cycle.

\[
\text{Percent Difference} = \frac{D_{\text{max}}}{\max(|F_{100\%}|)} \times 100
\]
4.3.1 Initial Gas Pressure Variation in the Model

Figure 4-20 shows the force vs. velocity plots, and Figure 4-21 shows the force vs. displacement plots for the three frequency segments as the initial gas pressure is varied. The force vs. time plots are shown in Figure 4-22. The results are summed up in Table 4-4, which shows the RMS difference, and the percent difference between the different parameter levels.

**Figure 4-20:** Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Initial Gas Pressure Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz
Sinusoidal Input, Constant Velocity, Frequency: 1 Hz, Varied Initial Gas Pressure

Sinusoidal Input, Constant Velocity, Frequency: 40 Hz, Varied Initial Gas Pressure

Sinusoidal Input, Constant Velocity, Frequency: 90 Hz, Varied Initial Gas Pressure

Figure 4-21: Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Initial Gas Pressure Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz
Figure 4-22: Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Initial Gas Pressure
Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz

Table 4-4: Difference in Output Force as the Parameter is Varied

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Initial Gas Pressure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Difference</td>
<td>75% vs. 100 %</td>
</tr>
<tr>
<td>Input</td>
<td>RMS Diff., lbs / % Difference</td>
</tr>
<tr>
<td>1 Hz</td>
<td>2.35 / 3.7%</td>
</tr>
<tr>
<td>40 Hz</td>
<td>1.62 / 2.6%</td>
</tr>
<tr>
<td>90 Hz</td>
<td>1.25 / 2.8%</td>
</tr>
</tbody>
</table>
4.3.2 Friction Variation in the Model

The force vs. velocity plots show the results as friction is varied, Figure 4-23. The force vs. displacement plots are shown in Figure 4-24, and the force vs. time plots are in Figure 4-25. The model’s response to a variation in friction is summed up in Table 4-5.

**Figure 4-23:** Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Friction Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz
Figure 4-24: Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Friction Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz
Figure 4-25: Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Friction Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz

Table 4-5: Difference in Output Force as the Parameter is Varied

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Friction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Difference</td>
<td>75% vs. 100%</td>
</tr>
<tr>
<td>Input</td>
<td>RMS Diff., lbs / % Difference</td>
</tr>
<tr>
<td>1 Hz</td>
<td>0.21 / -0.1%</td>
</tr>
<tr>
<td>40 Hz</td>
<td>0.19 / -0.1%</td>
</tr>
<tr>
<td>90 Hz</td>
<td>0.14 / -0.1%</td>
</tr>
</tbody>
</table>
4.3.3 Density Variation in the Model

Figure 4-26 shows the force vs. velocity plots, and Figure 4-27 shows the force vs. displacement plots for the three frequency segments as fluid density is varied. The force vs. time plots are shown in Figure 4-28, and the results are summed up in Table 4-6.

Figure 4-26: Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Fluid Density Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz
Figure 4-27: Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Fluid Density Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz.
Figure 4-28: Sinusoidal Input, Constant Velocity, Model Data Compared to +/- 25% Fluid Density Parameter Variation for Three Input Frequencies: 1, 40, and 90 Hz

Table 4-6: Difference in Output Force as the Parameter is Varied

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Fluid Density</th>
</tr>
</thead>
<tbody>
<tr>
<td>Difference</td>
<td>75% vs 100%</td>
</tr>
<tr>
<td>Input</td>
<td>RMS Diff., lbs / % Difference</td>
</tr>
<tr>
<td>1 Hz</td>
<td>34.95 / -22.3%</td>
</tr>
<tr>
<td>40 Hz</td>
<td>40.70 / -27.3%</td>
</tr>
<tr>
<td>90 Hz</td>
<td>33.66 / -24.3%</td>
</tr>
</tbody>
</table>
The results shown in Figure 4-20 through Figure 4-28, and Table 4-4 through Table 4-6 give a good indication of how changes in initial gas pressure, friction, and fluid density will affect a damper’s output force for different frequency segments. These results do not show any clear trends that can relate a change in a parameter to a change in the amount of hysteresis in the damper’s output force. So, this method does not give a good representation of a parameter’s effect on the amount of hysteresis in the damper’s output force for a given frequency. After reviewing the model, including the addition of the warping filter, it was determined that all of the effects of hysteresis are going to be accounted for with the warping filter. Any parameter change made after the model has been passed through the warping filter will not show that parameter’s true frequency dependence.

A discussion on another potential way to use this damper model, with the warping filter, to show a parameter’s influence on the amount of hysteresis in the damper’s output force is as follows. For clarity, this method will describe a variation in friction (Ff). This method begins with the same model as developed in Section 4.1 before the addition of the warping filter. At this point, we have a model which accurately predicts a parameter’s change, but lacks the dynamics associated with hysteresis in the damper’s output force. It is here where we would vary Ff by plus or minus twenty-five percent of the nominal value. After varying Ff, we would tune the warping filter parameters for the 75 %, 100%, and 125 % cases with the 1 – 10 Hz input. This would be repeated with the 40 – 50 Hz and the 90 – 100 Hz inputs. The difference in the warping filter parameters for these three inputs should show Ff’s contribution to the hysteresis in the damper’s output force as frequency is increased. This method will not be implemented in this study due to time restrictions, but it is left as future work in the further development of this model.

4.4 Results and Discussion

Based on previous models, a parametric damper model was developed to look at a parameter variation’s effect on the amount of hysteresis in the damper’s output force. Parameters which could be measured were recorded and used in the model. Equations
were developed that describe the flows within the damper, flows within the damper’s head valve, and the output force produced by these flows. The model was tuned to experimental data, and it was discovered that the model didn’t capture the dynamics seen in the data beyond 5 Hz. To accurately represent the damper’s response over a wider range of frequencies, a warping filter was added to the model to represent the hysteresis seen in the experimental data which the model didn’t capture. The updated model and warping filter were tuned to experimental data for 1 – 10, 40 – 50, and 90 – 100 Hz frequency segments. Using this tuned model, the output force differences were determined as initial gas pressure, friction, and fluid density were varied plus or minus twenty-five percent from their nominal values. The parameter study’s intention was to show the parameter’s influence on the amount of hysteresis in the damper’s output force. However, this method did not show any clear trends that would indicate a parameters influence on the amount of hysteresis in the output force. Another method to test a parameters influence on hysteresis in the output force was proposed but was not tested in this study. The next step to verify a parameter’s influence on hysteresis is with experimental testing, which will show the effect initial gas pressure, friction, and fluid density have on the amount of hysteresis in the damper’s output force.
5 EXPERIMENTAL EVALUATION OF PARAMETERS
HYSTERETIC EFFECTS

Previous literature [2,3,4] states that “cavitation has little contribution to the output force effects” (i.e. hysteresis). Increasing the initial gas pressure will increase the pressure on the hydraulic damper fluid, which in turn raises the point at which cavitation occurs and increases the gas spring effect. This parameter’s effect was tested experimentally by varying the initial gas pressure in the physical damper and exciting it at different inputs to study the amount of hysteresis in the damper’s output force. Another parameter investigated experimentally was friction. This parameter is tagged as a contributor to hysteresis by Yung [34,35,36]. Since the Roehrig damper dynamometer has the capability to measure the damper’s internal friction, this parameter’s contribution to hysteresis can be tested and measured in the lab as well. The final parameter that this experimental evaluation looked at is the type of damper fluid. Varying fluid type will not allow us to vary the fluid density, independent of a change in viscosity, but it will show the damper’s behavior as the fluid is varied.

5.1 Experimental Evaluation of Parameters Effect on Hysteresis in the Damper’s Output Force

For this experimental evaluation of selected damper parameter’s effect on hysteresis in the output force, we used the same constant velocity, single frequency sinusoidal input that was used in chapter three. Four frequencies from the eight tested in chapter three were chosen to show a parameter’s influence for a range of frequencies. The damper was tested with a sinusoidal input at 1, 4, 16, 64 Hz frequencies for all of the parameters studied.
5.1.1 Initial Gas Pressure Variation

Initial gas pressure in the damper is set by pressurizing the gas chamber with nitrogen through the Schrader valve, located at the top of the damper. The damper was kept connected to the dynamometer for all pressure adjustments. This was done to prevent any error associated with reconnecting the damper, and re-zeroing the load cell.

A 25% increase in initial gas pressure from the nominal 50 psi did not show a considerable difference in the damper’s output force, so three initial gas pressure settings which the damper would likely see in service were chosen to perform this study. The initial gas pressure settings chosen were 50, 100, 150 and 200 psi. The Roehrig dynamometer has the capability of measuring the gas force at a desired damper displacement. The standard test programmed into the Shock 6.0 software measures gas force at a 0.5 inch displacement. Gas force was measured for each initial gas pressure, for all four of the frequencies tested. This gave four gas force values for each initial gas pressure setting. The four measured gas forces were averaged, and these averaged forces are shown in Table 5-1.

<table>
<thead>
<tr>
<th>$P_{gi}$, psi</th>
<th>Avg. Gas Force, lbs</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>15.77</td>
</tr>
<tr>
<td>100</td>
<td>33.37</td>
</tr>
<tr>
<td>150</td>
<td>48.10</td>
</tr>
<tr>
<td>200</td>
<td>64.70</td>
</tr>
</tbody>
</table>

Figure 5-1 shows the results of the variation in initial gas pressure at 1 Hz. As the initial gas pressure is increased, the force vs. velocity lissajou is shifted up. This result is expected, and is also captured by the damper model as seen in Figure 4-20 through Figure 4-22. Another result shown in Figure 5-1 is an increase in hysteresis as the damper’s initial gas pressure is increased. The hysteresis not only increases in the low velocity...
region, but the high velocity region as well. What makes this result interesting is that there was not an increase in hysteresis at the other frequencies tested, as initial gas pressure was increased. The 4, 16, and 64 Hz inputs have the same amount of hysteresis regardless of the initial gas pressure. To verify this visually, a constant value was subtracted from the entire 200 psi case’s output force data set to see how well it lined up with the other initial gas pressure cases. This essentially shifted the lissajou down by an amount close to the gas force. When the 200 psi case was compared to the 50 psi case, 60 lbs of force was subtracted from the whole range of the 200 psi output force. With this adjustment, the 200 psi case’s lissajou aligned with the 50 psi case’s lissajou for the 4, 16, and 64 Hz inputs, but not for the 1 Hz input. Figure 5-2 through Figure 5-4 show the variation in initial gas pressure for the 4, 16, and 64 Hz inputs. These cases show the same amount of hysteresis in the damper’s output force regardless of initial gas pressure.

Figure 5-1: Sinusoidal Input, Constant Peak Velocity, 1Hz Frequency, Initial Gas Pressure Variation
Figure 5-2: Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Initial Gas Pressure Variation

Figure 5-3: Sinusoidal Input, Constant Peak Velocity, 16 Hz Frequency, Initial Gas Pressure Variation
Besides the 1 Hz case, variations in initial gas pressure did not have an effect on the amount of hysteresis in the damper’s output force for the other frequency inputs. The increase in hysteresis around 1 Hz is more than likely due to stiction at the seal of the main piston. Based on this experimental variation in the damper’s initial gas pressure, the parametric damper model from chapter four is able to capture the effects initial gas pressure has on the damper’s output force, but the model does not show an increase in hysteresis as the initial gas pressure was increased.

### 5.1.2 Friction Variation

Minimizing friction is one of the main goals of all damper manufacturers, but looking for ways to reduce the internal friction of the damper was not one of the goals of this study. So rather than trying to reduce friction, in order to vary it, friction was increased for this experimental study. Figure 5-5 shows a cutaway of the main piston. The piston has a groove in it with an o-ring that pushes out on the seal to prevent fluid leakage past the piston during operation. The seal on the main piston, and the seal at the interface with the damper shaft are the main sources of friction within the damper. Using the Roehrig dynamometer, the seal drag force (friction force) was measured for the 1, 4, 16, and 64 Hz cases. Roehrig Engineering describes the seal drag test as follows: “The seal drag test
runs a triangle wave about a particular offset into the damper. The offset is the zero
displacement position for the test. The window size specifies the amount of displacement
on either side of zero (the offset) to perform the test. The test speed should be quite slow
[26].” For all testing conducted for this study, the window size for the seal drag test was
0.125 inches, the speed was 0.03 in/s and the offset was 3 inches. Speed and window
size were the default values of the dynamometer. Friction force was measured for the
four inputs and averaged. Average friction force was found to be 15.83 lbs. This number
will be considered the nominal friction force value. The damper was then tested at the
four frequencies, and the experimental data was collected. A description of a
modification to the main piston is described in the following paragraph.

![Piston](image)

**Figure 5-5:** Cutaway of Main Piston with out any Modification

The model study considered a 25% parameter variation; so increasing the friction by as
close to that amount was the goal. In order to increase the friction within the damper, the
side force on the seal had to be increased. This was accomplished by reducing the depth
of the groove where the o-ring rests with Teflon tape. Teflon tape was chosen because it
allowed the o-ring to sink into it some, which keep the o-ring from rolling in the seal’s
grove. Also, the Teflon would not disintegrate at high temperatures, and did not have the
potential to ruin any parts of the damper. Figure 5-6 shows a cutaway of the modified
piston. The Teflon tape is shown in red. It should be noted that the tape did not
completely fill the o-ring’s groove. There was about 1/16 of an inch left for the o-ring to
rest in.
Using the Roehrig dynamometer, friction force was measured for 1, 4, 16, and 64 Hz inputs. The friction forces were averaged, and this average friction force was found to be 18.82 lbs for the modified damper. The percent increase from the nominal value was 19%. This was less than the desired 25%. The addition of more Teflon tape could lead to the o-ring rolling in the piston seal groove, which could cause unwanted effects in the damper’s output force. To prevent this, the 19% increase in friction was deemed acceptable, and the damper was then tested at the four frequencies, and the experimental data was collected.

Figure 5-7 shows the damper’s output forces for the nominal versus the increased friction case for a sinusoidal input with a frequency of 1 Hz. This increase in friction has a minimal effect on the output force in rebound, and almost no effect in compression. The effect in rebound is not an increase in the amount of hysteresis in the dampers output force, but rather an increase in the slope of the curve. Without further testing, it would be rash to comment on why the increased friction only has an effect on the rebound portion of the lissajou.
A variation in the output force due to an increase in friction is shown in Figure 5-8 for the 4 Hz case. Again, the variation is predominant in rebound, but more variation in compression is seen compared to the 1 Hz case. This trend continues for the 16 Hz case, as seen in Figure 5-9.

Figure 5-7: Sinusoidal Input, Constant Peak Velocity, 1Hz Frequency, Internal Friction Variation

Figure 5-8: Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Internal Friction Variation
The most variation seen in this study was with 64 Hz input, Figure 5-10. This simple study suggests that friction does have an influence on the damper’s output force, and its influence is frequency dependent. Overall, the experimental testing exhibited more of a change in output force from a physical variation in friction, than the parametric model did for a virtual variation in friction. This suggests that the parametric model needs a better way to account for friction within the damper.

Figure 5-9: Sinusoidal Input, Constant Peak Velocity, 16 Hz Frequency, Internal Friction Variation

Figure 5-10: Sinusoidal Input, Constant Peak Velocity, 64 Hz Frequency, Internal Friction Variation
5.1.3 Fluid Type

The last parameter which was varied in the model study in chapter four was the damper’s fluid density. This parameter is easily varied within the model, but it proved to be more challenging to vary density in the physical damper without affecting the fluid’s viscosity. Nevertheless, the damper was tested with three different fluids. The effects of this fluid variation can not be directly related to the density variation in the model, but this variation will shed some light on the affect on the damper’s output force as the fluid type is varied.

Table 5-2 shows the manufacturer’s listed specific gravity, calculated densities, and kinematic viscosities for two Society of Automotive Engineering (SAE) weights of damper fluid and water. Water was tested because it gives the correct density variation, and because it is rumored that some race teams have run races with water as their working fluid. The percent increase in density from 2.5 WT oil to water is 27.4%, which is close to the 25% variation in the model, but the large difference in kinematic viscosity does not allow for a fair comparison of these two fluids. This variation in fluid easily gives a variation in density, but comes with an undesired change in viscosity.

One step in the process of rebuilding a damper is to bleed out air trapped in the main piston when it is inserted into the damper oil, Appendix A. To further insure that all air was removed from the system for the fluid type tests, the fluid was vacuum degassed with

<table>
<thead>
<tr>
<th>Fluid Type</th>
<th>SG</th>
<th>Density, lbm/ft³</th>
<th>Kinematic Viscosity, ft²/s</th>
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<tr>
<td>2.5 WT</td>
<td>0.816</td>
<td>50.92</td>
<td>@ 100 °F 14.64 x 10⁻⁶</td>
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<tr>
<td>5.0 WT</td>
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<td>@ 100 °F 28.74 x 10⁻⁶</td>
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<td>Water</td>
<td>1</td>
<td>64.85</td>
<td>@ 100 °F 0.738 x 10⁻⁶</td>
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</tbody>
</table>
the setup shown in Figure 5-11. Here, a vacuum pump was used to pull -15 in Hg on the damper fluid to remove any remaining air trapped within the damper’s fluid.

![Vacuum Degassing of Damper Fluid](image)

**Figure 5-11:** Vacuum Degassing of Damper Fluid

Figure 5-12 shows the output forces for 1 Hz input, comparing the three fluid types. Although we can not use the test results with the water as the working fluid to validate the results from the model study, due to the difference in viscosity, it is interesting to note, that like the model, there is a large variation in the output force for an increase in density. Similar to the model, this variation is more pronounced in rebound than compression, which has to do with the lack of bleed in compression. Without further testing, it would not be good practice to comment on why the water’s effect on the damper’s output force is greatest for velocities between -10 in/s and -4 in/s in rebound, and between 10 in/s and 5 in/s in compression. Figure 5-12 through Figure 5-15 show the effect a change in damper fluid has on the damper’s output force.
Figure 5-12: Sinusoidal Input, Constant Peak Velocity, 1 Hz Frequency, Fluid Type Variation

Figure 5-13: Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Fluid Type Variation
After preparing the damper for testing the effect of a variation in fluid type, one question arises. How is the damper’s output force affected by damper preparation? For instance, what is the difference in the damper’s output force for a damper which is bled as instructed by the manufacturer, versus a damper which is subjected to vacuum degassing, and how do these two preparations compare to a damper which is not bleed at all? To find the answer to these questions, a damper with SAE 5 WT damper fluid was prepared, and tested first by bleeding as recommended by the manufacturer, then bleeding and
vacuum degassing, and finally was built without any consideration to bleeding or vacuum degassing. Figure 5-16 shows a close up of the compression side of the 1 Hz input. Looking at this force vs. velocity plot, we can see that vacuum degassing the working fluid leads to a higher damper output force, the next best preparation is bleeding the fluid, and the worst method of preparation was to not bleed or vacuum degas the fluid.

![Sinusoidal Input, Constant Velocity, Frequency: 1 Hz, Varying Fluid Preparation](image)

**Figure 5-16:** Sinusoidal Input, Constant Peak Velocity, 1 Hz Frequency, Fluid Preparation Variation

Figure 5-17 through Figure 5-20 show the effect fluid preparation has on the lissajou in the force vs. velocity plots, and the force vs. displacement plots for the 1, 4, 16, and 64 Hz. inputs. Although the variation between fluid preparations is relatively small, around 10 lbs, it should be noted that this study was conducted for only one velocity. Higher velocity, increased temperature, and the damper’s input all have the potential to increase the variation between the damper’s output force of the differently prepared fluids.
Figure 5-17: Sinusoidal Input, Constant Peak Velocity, 1 Hz Frequency, Fluid Preparation Variation

Figure 5-18: Sinusoidal Input, Constant Peak Velocity, 4 Hz Frequency, Fluid Preparation Variation
5.2 Result and Discussion

In Chapter Four, parameters with the potential to have an influence on the amount of hysteresis in the damper’s output force were tested with a parametric model. The parameter variation in the model showed their effects on the damper’s output force, but
were unable to capture the parameter’s contribution to the amount of hysteresis in the damper’s output force. These same parameters were then tested experimentally to show what the model could not. The parameters tested were initial gas pressure, friction, and fluid type/preparation. They were tested with a sinusoidal input at 1, 4, 16, and 64 Hz.

The first parameter tested was initial gas pressure. When this parameter was increased from the nominal 50 psi, there was little increase in the amount of hysteresis in the damper’s output force, in the 4, 16, and 64 Hz inputs. The 1 Hz input, however, showed an appreciable increase in the amount of hysteresis in the damper’s output force as the initial gas pressure was increased. This increase was a steady increase with the amount of hysteresis growing larger for each pressure increase from 50 psi. This evaluation shows that at frequencies around 1 Hz, initial gas pressure increases the amount of hysteresis in the damper’s output force as the initial gas pressure is increased. The experimental evaluation of a variation in initial gas pressure also showed that the parametric model of chapter four is able to capture the effects of initial gas pressure on the damper’s output force, but not the hysteretic effects.

The second parameter tested was friction within the damper. It was easier to increase the amount of friction in the damper, rather than find ways to decrease friction in the damper, so for this study a variation in friction is an increase in friction. The damper was tested at four frequencies without any modification to the damper. By increasing the side load on the main piston seal, friction within the damper was increased. Initially there was no variation in the damper’s output force in compression, and only a little in rebound, but as the frequency was increased, the variation in the damper’s output force increased in both rebound and compression. Although this study was limited, it still shows that friction does have an effect on the amount of hysteresis in the damper’s output force, and it increases with frequency. This study also shows that the parametric model does not do a good job of capturing friction within the damper. According to the model, there is no variation in the damper’s output force, which this study proves is not true.

The third, and last parameter varied was fluid type. Fluid type is not one of the damper model’s parameters, so this final study can not be directly related back to the model. The
parameter varied in the parametric model study in Chapter Four was density, but it proved difficult to vary density alone without also affecting the viscosity of the fluid. Fluid density and viscosity were considered for three different fluids, and their effect on the damper’s output force was tested. Changing the fluid from SAE 2.5 WT to SAE 5 WT increased the damper’s output force in compression. There was little change in the damper’s output force in rebound for the 1, 4, and 16 Hz inputs. In the 64 Hz input, the damper’s output force was increased in both compression and rebound. With water as the working fluid, the damper’s output force was increased from the SAE 2.5 and 5 WT fluids in both rebound and compression. This increase only occurred after a certain velocity, -4 in/s in rebound and 5 in/s in compression. In between these velocities there was change to the damper’s output force, but not as significant as the velocities beyond -4 in/s in rebound and 5 in/s in compression.

Overall, the variation in SAE fluid did not show an increase in the amount of hysteresis in the damper’s output force as frequency was increased. One advantage to conducting tests with a damper model is that you can vary parameters which are difficult to vary physically. An increase in density in the model showed an increase in the damper’s output force in the model’s response. Water was 27.4 % denser than the SAE 2.5 WT fluid, but had significantly less viscosity. Interestingly enough though, water showed an increase in the damper’s output force, but further testing with additional fluids is needed to truly understand how viscosity and density effect the damper’s output force.

One final test while looking at fluid type was fluid preparation. For this study, the fluid was prepared and tested by bleeding as the manufacturer recommends, bleeding and then vacuum degassing the fluid, and testing the damper without bleeding or vacuum degassing the fluid. The testing results showed that bleeding the fluid then vacuum degassing it lead to the largest damper output force for a given velocity. This was determined to be the best fluid preparation method followed by bleeding as the manufacturer recommends, and the worst fluid preparation method was to not bleed or vacuum degas the fluid.
6 CONCLUSIONS AND RECOMMENDATIONS

This chapter presents the highlights of the physical and virtual testing conducted in this study. The objectives from chapter one will be reexamined, along with a discussion on how these objectives were met. Recommendations for future work will be the final discussion in this chapter.

6.1 Summary

Before the objectives from Chapter One are reexamined, a brief summary of this study will be presented.

- Chapter One described why this study is important, what it accomplished, and how it was done.

- Chapter Two gave the reader an understanding of the damper’s role on the vehicle, the parts of the damper, the damper’s operation and how the damper is tested.

- Chapter Three presented a fundamental look at the damper’s frequency dependence. This was accomplished first by limit and linear equation analysis, followed by experimental testing which included sinusoidal and sinusoidal sweep inputs.

- Chapter Four showed the development of a parametric damper model, the tuning of this model to experimental data, and the use of this model in a parametric study.

- Chapter Five presented three parameters linked to hysteresis in the damper’s output force. This chapter showed the initial gas pressure, friction, and the fluid type’s effect on the amount of hysteresis in the damper’s output force.
With the previous chapters summarized, we will look back at the objectives from Chapter One and tell how these objectives have been met.

The first objective of this study was to better understand the fundamental physical characteristics of race dampers in both low and high frequency regimens. Vary few studies have considered race dampers for testing purposes, and none of these studies using a race damper were conducted at high frequencies. The modern racing damper is a highly nonlinear device whose output force is dependent on more than just the velocity of the input. Because of this, it behaves differently depending on whether it is in a low or high frequency regimen. This objective was met through experimental testing and parametric model studies. The results in Chapters Three through Five provide results which give a fundamental understanding of the physical characteristics of the race damper.

The second objective was to reexamine the commonly held notion that a damper’s output force is a function of velocity. If you take a look at any undergraduate level dynamics book on dynamics or vibration, the damper’s output force is a linear function of velocity and a constant damping coefficient. The assumption that the damper’s output force is a function of only velocity makes the math easier, but fails to capture the true nature of the damper. In reality, any damper, especially a modern racing damper, is a highly nonlinear device that includes additional dynamics such as the gas spring effect from the gas chamber. The gas spring effect, and other contributors to hysteresis, makes the whole system’s output force behave dependent on more than just velocity.

Chapter Three shows how a linear model of a damper and spring in series is dependent not only on velocity, but frequency as well. This simple linear model was subjected to constant velocity sinusoidal inputs. Frequency is increased in the inputs from 1 to 64 Hz. As the frequency is increased, the damper behaved less like that expected of a damper, and began to behave more like a spring. This model study was confirmed with an experimental evaluation using the same inputs the simple linear model was subjected to, along with additional inputs at higher frequencies. Experimental result showed the same results as the model; an increase in input frequency caused the damper’s output force to
transition from the characteristic response of a damper to the characteristic response of a spring. This commonly held notion was further tested with a sinusoidal sweep input. The results from the sinusoidal sweep input were the same as the results from the sinusoid input and the results from the simple linear model.

The third objective of this study was to evaluate how the force characteristic of a damper changes from various combinations of displacements and frequencies, with velocity held constant. This objective closely relates to the second objective. When examining the notion that the damper’s force is a function of velocity, various combinations of displacements and frequencies, with velocity held constant, were explored. The outcome of this objective was the same as the second objective. With peak velocity held constant, a decrease in displacement will give an increase in frequency, and as frequency is increased, the damper’s output force transitions from that of a damper to that of a spring.

The fourth objective was to provide test methodologies which can be use for accurately evaluating damper characteristics at various frequencies. The test methodologies developed in this study kept peak velocity constant so that the damper’s frequency dependent behavior could be studied. The inputs which allowed the damper’s peak velocity to be kept constant were the sinusoidal input and the sinusoidal sweeping input. The sinusoidal input allows the user to change frequency, which changes displacement, while maintaining a constant peak velocity. The frequency is chosen at discrete points of interest. The other input used in the study was the constant peak velocity sinusoidal sweep. This input allows users to sweep over a range of frequencies, so that the effect of a wide range of frequencies can be seen.

The fifth and final objective of this study was to provide an analytical model, supported by test data for performing parametric studies for dampers. In chapter four a parametric model was developed, based on the Penske 7300 racing damper. This damper model expanded upon previous damper models to include a head valve, a feature that has been added to racing dampers in recent years. The model was tuned with experimental data, and was used for a parametric study in an IMECE paper, as well as the parameter study in chapter four.
6.2 Recommendations for Future Research

This study has laid the foundation for characterizing a race damper’s frequency dependent behavior. Up until now, little work has been done on passenger vehicle dampers at high frequencies, and even less work has been done on race dampers at high or low frequencies. So, there are some great opportunities to build upon this study to further the knowledge in the field of dampers. This section will present recommendations in a chapter by chapter approach.

The first chapter where future research could be conducted is in Chapter Three. Here one could:

- Conduct additional tests that include a random input which varies frequency and displacement, but keeps velocity constant. These additional tests should include a wider range of constant velocities.

- Rerun the tests conducted in Chapter Three with additional damper builds. These tests would include: changing the shim configurations, changing the piston type, changing the bleed, and changing the head valve configurations.

- Rerun the tests conducted in Chapter Three at different operating temperatures.

In Chapter Four a parametric model was developed to conduct parametric studies without the hassle of having to reconfigure the physical damper. Some of the improvements that could be made to the parametric model include:

- Rerun the parametric study conducted in Chapter Four by varying the parameter in the model before tuning the warping filter.

- The addition of more dynamic effects to the model. This should include: an improved friction model, adding temperature effects, and accounting for air entrapped within the working fluid. It would not be practical to add the effects of fluid compressibility to the current model, because many of the
equations are derived based on the assumption of an incompressible fluid. To consider this effect, one would be better off developing a new model.

Chapter Five presented an experimental study on initial gas pressure, friction, and fluid type’s effect on the damper’s output force. This testing only touches on all of the testing which could be conducted to see the effects parameter variation has on the damper’s output force. To expand upon the results from Chapter Five, additional testing should look at:

- A more in-depth look at fluid preparation’s effect on output force.
- Additional fluid types to test the working fluid’s effect on the damper’s output force.

Beyond improving upon the tests conducted in Chapter Five, testing which looks at other parameters such as: temperature, friction at the gas piston, and preload on the shim stacks should be considered.
References


26. Roehrig Engineering Inc. Shock 6.0 and EMA Users Manuel


Appendix A: Damper Rebuilding Procedures

Rebuild Procedures for Penske 7300 Damper

Disassembly

1. Back the bleed adjuster to full soft and depressurize the damper.
2. Clamp the body eyelet in the damper vise and place the overflow ring on the body.
3. Unscrew the shaft bearing assembly and remove from damper body. Place the shaft assembly in the vise next to the body.
4. Dump oil from damper body in appropriate container and place damper body back in vise.
5. Remove ¾” nut to change piston assembly.
6. With the body sleeve and body wrench, unscrew the body from body cap to access the head valve.
7. Remove the head valve if necessary with the specialty socket.

Assembly

1. Press open the Dill valve on the body cap and press the gas piston all of the way down.
2. Fill the body cap with oil up to the threads.
3. If the head valve was removed, make sure it was rebuilt properly and torque the head valve nut to 160 in-lbs.
4. Install head valve plate and torque to 240 in-lbs.
5. Fill the damper body half full of oil, thread it back into the body cap, and tighten with the body sleeve and wrench.
6. With the fluid still in the body, pressurize the gas chamber to 150 psi.
7. Place damper back into the vise and fill with oil to the bottom of the threads.
8. Re-valve the piston if desired and place back on shaft with compression side facing down.
9. Torque the ¾” nut to 300 in-lbs.
10. Torque the bleed jet to 100 in-lbs.
11. Insert the shaft and piston assembly into the damper body and begin to work out the air bubbles trapped in the piston, by using 1” – 2” strokes. Move the shaft up and down a few times, making sure the two bleed holes in the shaft remain below
the fluid surface. Lightly tap with rubber mallet to assure all the air bubbles are gone.

12. Pull the shaft up until the bleed holes in the shaft remain just below the surface of the oil.

13. Top off (to the top of threads) with oil and slide the shaft bearing down to seat the o-ring into the shock body without moving the shaft.

14. Depressurize the gas chamber while asserting pressure to the shaft bearing and thread the shaft bearing into the damper body and tighten.

15. Pressurize the gas chamber to required pressure.
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<td>RR-16</td>
<td>Retaining Ring, 1.025 Spriroloc</td>
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<td>MO-8T</td>
<td>Monoball, 500 ID, Teflon</td>
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<td>3</td>
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<td>Assembly, 7900 Body Complete (No Monoball) (Includes Items 3-10)</td>
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<td>Air Valve, Port O-Ring, S.S.</td>
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<td>IU-04</td>
<td>Valve Core, 2000 psi</td>
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<td>Valve Cap, High Temperature</td>
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<td>O-Ring, 2-010, Buna 70</td>
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**AS-73CDBD**

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*Incomplete Part Number*
Appendix B: Damper Testing Procedures

Before testing, you need to decide which data card you want to use to collect data. If you are testing at frequencies below 50 Hz, use the PMD data card that is standard with the dynamometer. For tests above 50 Hz, you should use the Nservo data card. The steps that follow will tell how to set up each data card and will also tell the rest of the procedures for testing with the Roehrig dynamometer.

PMD Mode

1. Before turning on the dyno check to see if the PMD power cord is plugged into the power strip. If it is not, plug it in.

2. Disconnect the Nservo sub-D connector on the dyno and connect the PMD sub-D connector.

3. Turn on the power to the machine and then power up the computer.

4. Start Shock 6.0

5. Select the Hardware menu and select the data card sub menu. I will give a warning message, select yes.

6. Click on the PMD data card row and it will be highlighted in blue. Click ok.

7. When changing back and forth between PMD and Nservo, you have to change how the data is plotted. To do this, either press F12 or select the Edit menu and select the Preferences sub menu.

8. Once the Preferences is open, select the data tab. Here, you can adjust how you want to look at the data.

9. Skip the Nservo setup section and continue on to the damper setup section.
**Nservo Mode**

1. Before turning on the dyno check to see if the PMD power cord is plugged into the power strip. If it is, unplug it.

2. Disconnect the PMD sub-D connector on the dyno and connect the Nservo sub-D connector.

3. Turn on the power to the machine and then power up the computer

4. Start Shock 6.0

5. Select the Hardware menu and select the data card sub menu. I will give a warning message, select yes.

6. Click on the Nservo data card row and it will be highlighted in blue. Click ok.

7. When changing back and forth between Nservo and PMD, you have to change how the data is plotted. To do this, either press F12 or select the Edit menu and select the Preferences sub menu.

8. Once the Preferences menu is open, select the data tab. Here, you can adjust how you want to look at the data.

9. Continue to the damper setup section.
Damper Setup

1. With the data card selected and the machine turned on; connect the damper to the top clevis only.

2. Refer to the Roehrig Manual to set up tests.

3. With the test set up and ready to run, the test control panel should be displayed.

4. Zero the load cell by clicking on the zero load cell button on the test control panel.

5. Connect the damper to the lower clevis and tighten the top and bottom clevises.

6. With everything connected, testing can begin. Click the start test button on the test control panel to begin testing.

7. Refer to the Roehrig Manual for any other question regarding setting up the test, looking at data and exporting data.
Appendix C: Damper Test Data and Model Parameters

Sinusoidal Data

Sinusoidal Input, Constant Velocity, Frequency: 1Hz

Sinusoidal Input, Constant Velocity, Frequency: 2Hz

Sinusoidal Input, Constant Velocity, Frequency: 4Hz
Displacement, in
Force, lbs
Sinusoidal Input, Constant Velocity, Frequency: 8Hz

Displacement, in
Force, lbs
Sinusoidal Input, Constant Velocity, Frequency: 8Hz

Displacement, in
Force, lbs
Sinusoidal Input, Constant Velocity, Frequency: 16Hz

Displacement, in
Force, lbs
Sinusoidal Input, Constant Velocity, Frequency: 16Hz

Displacement, in
Force, lbs
Sinusoidal Input, Constant Velocity, Frequency: 32Hz

Displacement, in
Force, lbs
Sinusoidal Input, Constant Velocity, Frequency: 32Hz
Sinusoidal Sweep Data
Sinusoidal Sweep, Constant Velocity, Frequency Range: 10 Hz to 20 Hz

Sinusoidal Sweep, Constant Velocity, Frequency Range: 20 Hz to 30 Hz

Sinusoidal Sweep, Constant Velocity, Frequency Range: 30 Hz to 40 Hz
Sinusoidal Sweep, Constant Velocity, Frequency Range: 40 Hz to 50 Hz

Sinusoidal Sweep, Constant Velocity, Frequency Range: 50 Hz to 60 Hz

Sinusoidal Sweep, Constant Velocity, Frequency Range: 60 Hz to 70 Hz
Model Parameters:

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<tr>
<td>$k$</td>
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<td>lbs/in.</td>
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<table>
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<tr>
<th>Parameter</th>
<th>Compression</th>
<th>Rebound</th>
<th>Units</th>
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<tbody>
<tr>
<td>$A_v$</td>
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<td>0.16</td>
<td>in.$^2$</td>
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<tr>
<td>$C_f$</td>
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<tr>
<td>$F_{sp}$</td>
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<td>5.00</td>
<td>lbs.</td>
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<tr>
<td>$D_{v hv}$</td>
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<td>in.</td>
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<tr>
<td>$A_{bhv}$</td>
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