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Exploration of Characteristic curve in FOX Float 3 shock dampers to expedite shock damp tuning.

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Exploration of Characteristic curve in FOX Float 3 shock dampers to expedite shock damp tuning.

An Honors Thesis submitted in partial fulfillment of the requirements for Honors in Mechanical Engineering

By

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Under the mentorship of Dr. Aniruddha Mitra

ABSTRACT

The shock absorber is an integral part of a vehicle suspension system and has a strong influence on its performance, especially in the case of motorsports. It is important to study the force versus velocity relationship, commonly known as the characteristic curve of the shock absorber both during compression and rebound. Vendor-supplied characteristics often reflect the behavior of the shock absorber in a particular setting. However, during the installation, the settings inside the shock absorber are adjusted to increase the human comfort level and performance of the vehicle. This may change the characteristic curve of the shock. The available data and direct comparison of different tune-up settings are limited. In an ideal model, the force is directly proportional to the velocity. However, in literature, except for the steep linearity at the relatively small section closer to the origin, the characteristic curve is often found to follow a regression model with an offset in the form of $F= a + bv^c$, where F is the force generated at the shock and v, the velocity, a, b, c are the regression parameters. In the current research, three Fox Float 3 are tested at their factory conditions to assess the relationships between the force and the velocity. Also, several shock position settings inside two of those shocks are tested to develop a mathematical model. A predominant linear trend has been observed for all the cases. Future work will involve tracking these parameters throughout their operational life cycle.

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KEY WORDS: Shock Absorber, Characteristic curve, Damping, Strain gauge, Linear potentiometer

INTRODUCTION:

Shock absorbers are an integral part of any vehicles suspension system. In performance applications these absorbers' main goal is to increase tractive effort between the vehicles ground and the racing surface. In military and heavy vehicle applications the purpose is to provide damping effort against extreme conditions allowing the vehicles to traverse the ground plane at a quicker and safer mode. In passenger vehicles the shock absorber's purpose is to provide a balance between ride comfort and performance by damping road irregularities and normal operating conditions. These absorbers even have applications of non-vehicle type such as heavy industrial machinery which needs damped vibration for fatigue or more consistent operating conditions. In all these applications the shock absorber must provide a median between the sprung and unsprung weight of the vehicle or machine. The Fox Float 3 is an air over hydraulic shock absorber that is used in a variety of vehicle applications. The shock comes standard on a sever snowmobile models and was recently incorporated into Fox's BAJA SAE program where it grew in popularity as the shock is easy to use in theory, light weight, and easy to package. The damping, however, is not a user-friendly adjustment available on these shocks as they are monotube shocks. A twin tube shock or a monotube shock encompassing a bypass valve would be easier to tune due to external adjustment making it so that there is no need for the user to discharge the gas and disassemble the shock to adjust. The Fox Float 3 does not possess these features presumably because of its original equipment manufacturer

(OEM) application on snowmobiles, one tune typically covers its original intent. If, however, a model could be made from experimental data to tune this shock then the adjustment could be done in a few, if not a single, adjustment saving time and money. Because this is a monotube shock, the main passage for hydraulic mineral oil to flow and for damping force to be developed exists inside the main orifices of the piston (main valve) these are tuned by shims that are seated on the top and bottom of the piston perpendicular to the theoretical flow of the oil and are secured with a nut attached to the end of the piston rod. By removing, adding, changing thickness, or changing diameter, these shims affect the damping characteristic curve (Force vs. velocity). Several different models have been made for shock absorbers and there are many starting points to this research. Some absorbers can be modeled solely off theoretical relationships between interactions inside the absorber. Many of these methods are valid to certain limitations whether it be speed, frequency, or displacement of the shock absorber.

LITERARY SEARCH:

It is important to know the setting for which dampers are used in a racing application to begin to understand the required testing conditions, parameters, and models to observe the shock in. *B. Warner* [1] provided an in depth, analytical investigation of suspension dampers for these applications. In his article he discusses a multitude of subjects including conventual testing methods, experimental methods, and specific manufacturer damp settings and responses.

The very basis of shock absorbers and their broad level purpose is to dissipate energy in an efficient way and disconnect a mass, which is meant to be protected from sometimes harsh input, from a mass that is in contact with that input. *D. Narkhede et al* [2] explore the nonlinear behavior of these dampers. The authors cite Maxwell visco-elastic model which involves a spring in series with a dashpot as the correct model corresponding to a broad range of applications. The authors then set up a loading test using custom shock testing machine that encompassed a weight of 98.1kN mounted on rollers to strike a damper to generate a half cycle sine wave signal. One of the conclusions they came to during testing was that the damper exponent α and coefficient Cw remain constant through the shock tests.

Shock absorbers are an integral part of describing a vehicle's performance in every aspect from how the vehicle handles to how much power the vehicle can get to the ground plane. Shock absorbers are a huge piece of the heart of a vehicle's suspension system and their adjustability can drastically change the characteristics of the vehicle. Shock absorbers typically use internal parts to create damping force of mineral oil usually varying in weight. Developing mathematical models that can predict damping is an integral part of being able to speed up the tuning process to achieve desirable damping force without endless trial and error. *K. Reybrouck* [3] develops a model that accounts for various areas of the shock absorber such as blow off springs, temperature, and pressure loss. The author compares his model figure 1 to experimental data via a shock dynamometer.

$$F_{damping} = \frac{F_{leak} \cdot F_{blow\text{-}off}}{\sqrt[K_{tr}]{(F_{leak})^{K_{tr}} + (F_{blow\text{-}off})^{K_{tr}}}} + F_{port}$$

Figure 1: K. Reybrouck

There are many complex ways to build easy adjustability to shock absorber. One way is through magnetic adjustments. *D. Więckowski1 et al.*[4] explore three magnetically adjustable shock absorbers with pre-defined signal (damping) increments. The method for collection involves sensors for force, and piston stroke to derive a force v. velocity diagram, and force v. displacement. The authors also model several important characteristics of these three shock absorbers including hysteresis, step response, and the damping itself. Some information from their research can be seen in figure 2.

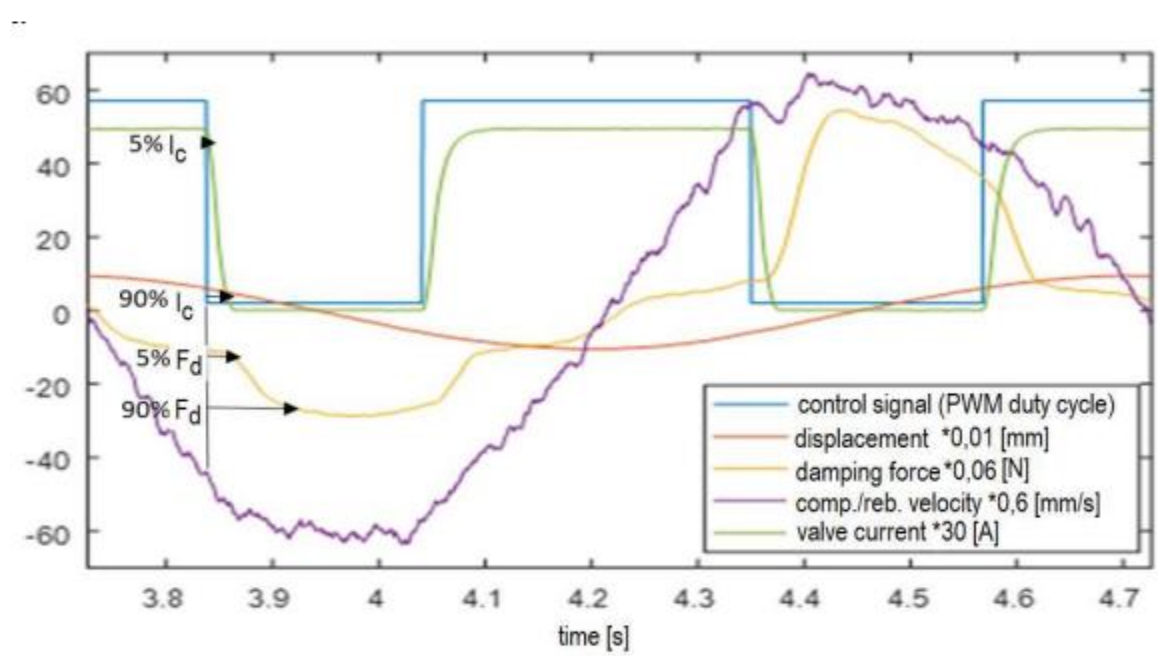

Figure 2:D. Więckowski1 et al. [4]

Modeling of a shock absorber for confidence in predictability beyond what is experimentally measured is an important part of shock absorbers. By developing models for specific shock absorbers, adjustments can be made with reduced testing time. *Y. Cui, et al* [5] discuss testing on a Mazda CX-7 shock absorber to develop a nonparametric model for the shock and discuss the outcome. The authors used a single post shaker table for testing which produced a standard looking damping graph (force v. velocity) even at high speed. The authors tested shocks at up to 1.3m/s (51.18in/s) which they claim to represent road irregularities.

The main damping adjustment for most shock absorbers is done through values with various methods of providing resistance to fluid motion. The main piston value is a characteristic all shock absorbers share and is adjustable most times through thin plates stacked on top of the value called shims. *D. Buczkowski et al.* [6] perform CFD analysis using Ansys Fluent v 19.0 to explore different shock settings in hopes to reduce tuning timeline hold up by optimizing flow though main valve design in the software before building a physical model. The authors concluded that the next step would be to test an additional piston design.

Cavitation is a phenomenon that occurs when air pockets are present somewhere within the fluid and can cause undesirable shock behavior and rough vehicle conditions. *P. Czop et al.* [7] investigate shock modeling and design variation for increased performance with an emphasis on cavitation effect. Based on CFD analysis the authors modified the main valve body to gain more desirable effects. They tested this through a hydraulic test rig enacting a known force on the absorber and measured force. V displacement. The authors quantified this by mapping the number of cycles the shock (figure 3) was able to maintain its damping force at and found that their modified piston was successful in outlasting the base model.

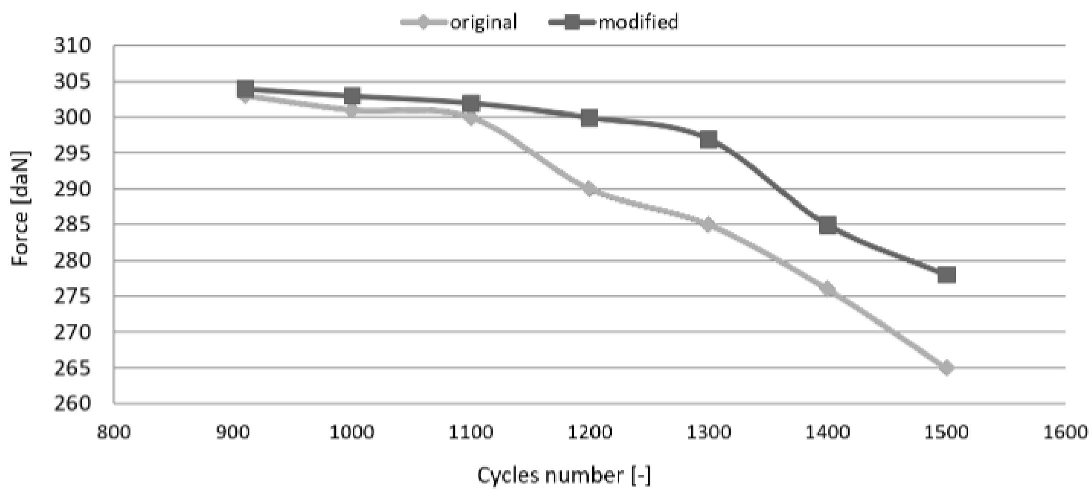

Figure 3: P. Czop et al. [7]

Damping lag is another undesirable effect that shocks can have, it occurs as the entering compression.

P. Czop, et al. [8]explore the causes of damping lag via mathematical analysis and simulation. The group uses a ratio derived from multiple theories to find X which acts as the ratio between the mass in bubbles occurring in the system over the mass of emulsion. Using the constant X, they can graph displacement v force with a noticeable damping lag shown in figure 4.

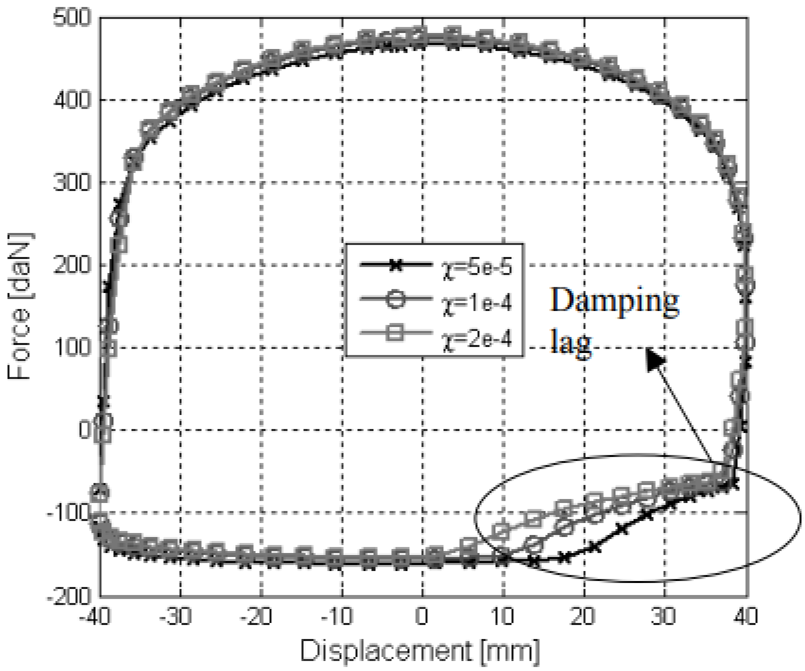

Figure 4: P. Czop et al. [8]

Pressure loss is an undesirable effect caused by friction in the small gap between the shim and the main valve. *L. Schickhofer et al.* [9] investigate ways to lessen pressure loss and tune the absorber. The authors use multiple variants of ansys that are put together along with mathematical models to arrive at theoretical points where the most pressure loss is found. The location that the authors found is highlighted in figure 5. In this journal, the authors also acknowledged oil viscosity differences and temperature variation.

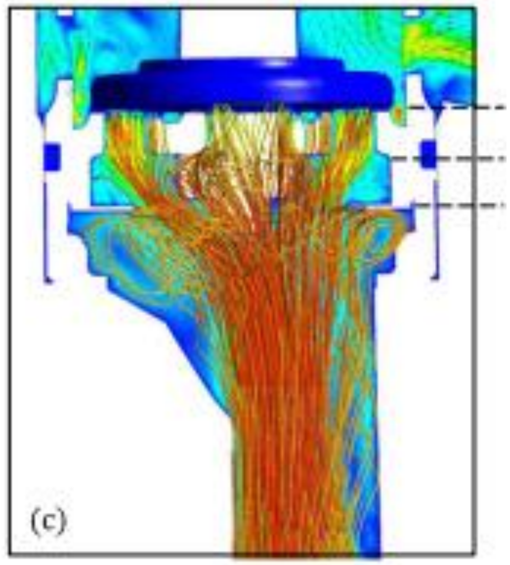

Figure 5: L. Schickhofer [9]

The viscosity of the fluid in a hydraulic damper affects the amount of damping force created by the fluid through the main and subsequent values. Temperature variation can cause a loss or increase in viscosity which can alter the properties observed in the damper. *Z. Hryciówa et al.* [10] the group use Instron 8802 electromechanical apparatus to test shock absorbers that had temperature variation. The shock was cooled with dry ice -40C and was tested until the shock reached a temperature of 83C during the test. The authors found that the damper coefficient drastically changed throughout the temperature range. This is due to the limitations on the pressure (main) value and change in the oil viscosity. Shown in figure 6, the force changes at each temperature for the velocity range. In this paper the apparatus used in the test was achieving velocities of 0.08m/s (3.1in/s) which is a notable point as most shock tests attempt to mimic road course conditions. In this test the comparison between damping forces and temperature is observed but no on road relationships could be drawn.

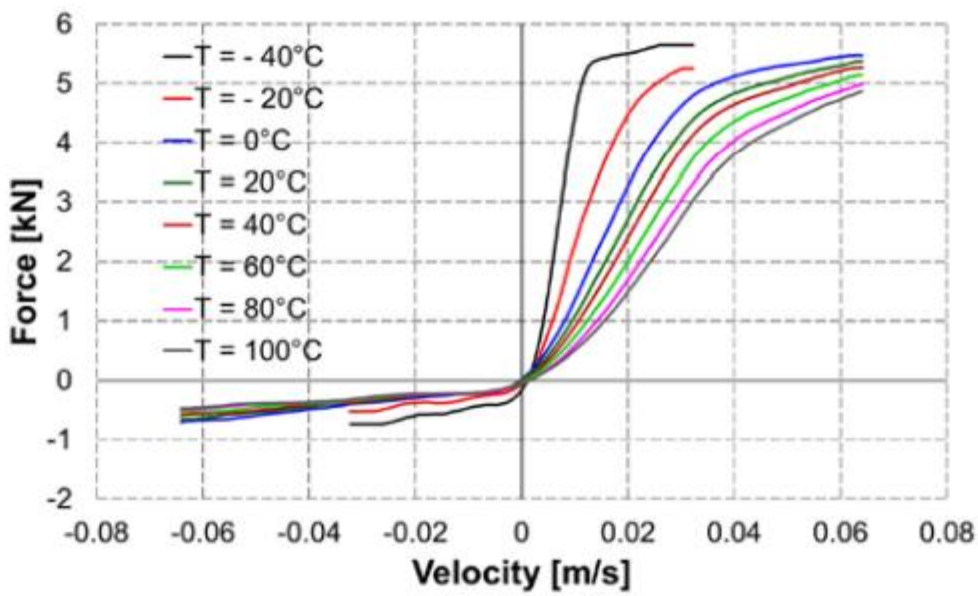

Figure 6: Z. Hryciówa et al. [10]

There are several ways to adjust damping force through the main valve of the shock absorber. Sprung valves that move out of the way with increased flow due to velocity are one way, the other is through shims used as mini cantilever beams. These shims have an associated spring rate that moves and returns at a linear rate. The shims are stacked on the top and bottom of the main valve in various thicknesses and diameters to develop a desirable damping rate for that side of the main valve. *P. Skačkauska et al.*

[11]investigate and verify a shim stack model. The group used experimental testing on a special electromechanical stand and obtained a peak velocity of 0.4m/s (15in/s). The

model developed by the authors includes several different equations derived from literature. An equation to note is the area through the main valve before and after the initial deflection of the stack occurs.

$$A_1 = 2\pi \cdot a_1 \cdot (y_1 - y_0) ,$$

Figure 7: Paulius Skačkauska et al. [11]

Another equation listed is a relationship between the effect between oil temperature and viscosity or the Vogel-Tamman-Fulcher type equation.

$$\eta = \eta_0 e^{\frac{E_0}{R(T - T_\eta)}} ,$$

Figure 8: Paulius Skačkauska et al [11]

Shock absorbers come in many different combinations depending on the tunability desired by the user and how that tunability is accomplished. Twin tube and Monotube shocks for example, are two different styles of shock that vary in tunability. The monotube shock is a single cylinder encompassing a main piston valve separating two chambers, one for compression and one for rebound directions. *Adrian Simms et al.* [12] explore damping on a monotube shock absorber. They claim that the Restoring Force Mapping Method, a popular method among shock modeling, was not a fitting model for monotube shock absorbers as they were non tunable. The authors chose to use an approach like Duym as his model factored in tunability which was a constraint in *Adrian Simms et al.* [12] research. After further developing the model, the author uses a quarter car model to obtain results and later adds in a pothole used by Jaguar tests which is shown in figure 9.

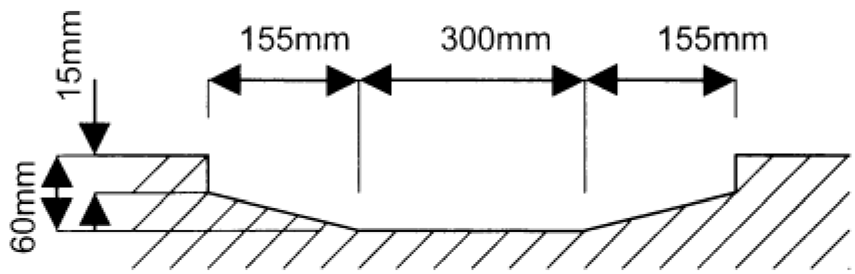

Figure 9: Adrian Simms et al. [12]

Frequency is a key variable when testing and characterizing a shock absorber. Damping force vs velocity ran are the main components of a characteristic curve, however frequency is a high level tunable characteristic capable and powerful enough to describe the shock absorber in detail. *D Kowalsk* [13] *et al* explore this trait as it is a useful way to detect and describe a shock based on nonlinear behavior. This is because in a road setting, the shock is subject to multiple, non-predictable, obstacles. This non liner behavior is part of the reason that shocks are difficult to describe with models. The authors mention a mathematical model made by Lang that worked for single signal excitation. The author's experiment first involved exposing the shock to two signal excitations using an MTS 831 machine and recording the response. The conclusion that *D. Kowalsk* [13] came to was that the higher frequency of the two sine wave inputs was the dominant one. The next test was to give the shock multiple sine wave inputs and observe behavior. The author found that a single sine wave model developed by Lang would correctly fit, but a multiple sine wave input behaved nonlinearly. The next step conducted by *D. Kowalsk* [13] was to fit a mid-size sedan with accelerometers and measure input frequency shown in figure 10 to compare to experimental lab data.

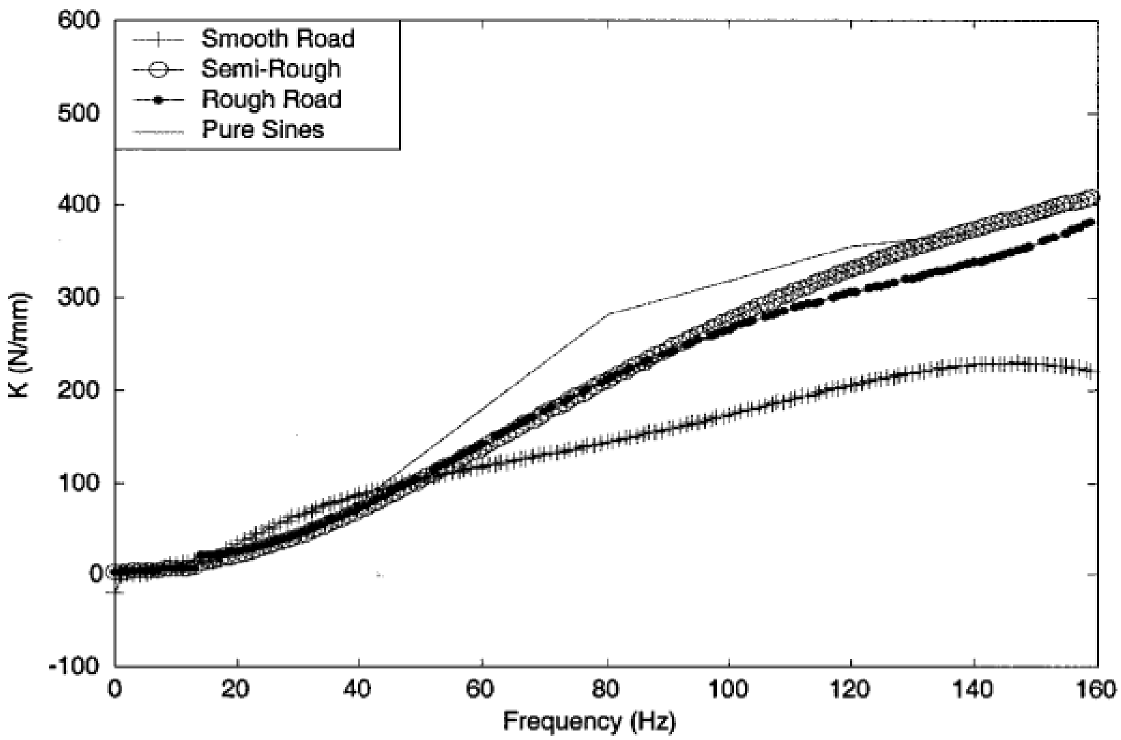

Figure 10: D Kowalsk [13]

This aspect of the research is unique as not many experiments make it to dynamic input testing. The authors found that higher frequency inputs and low amplitude inputs were not very accurate due to difficulty in reproducing those conditions, but most of the other scenarios were able to be predicted well.

Age has a significant effect on shock absorber performance and the characteristic curve. Shocks can be prone to failure in a few known areas. *C. Howard et al.* [14] attempt to observe the effects of age via a thermomechanical model to estimate work done through the shock as an indication of age. They cite the areas prone to failure as the rod main rod seal, internal damage, mechanical damage of body, breaking components, and losing gas pressure. The authors cite the most common failure as the main piston (main valve) seal, this directed the authors work towards finding a method of managing the failure of the piston seal. *C. Howard et al.* [14] propose a method using a calorimetry method to predict the temperature observed over time of a shock absorber on rough road and found their curve agreed with simulated data from a filtered quarter car model.

S.W.R. Duym is one of the highest cited authors on shock dampers. In this paper he explores modeling a shock damper that satisfies a multitude of requirements established by the INVEC consortium, a group of 7 well known car manufacturers. The model was proposed to be good to frequencies up to 30hz as *S.W.R. Duym* [15] mentions it is very difficult to simulate frequencies above that. His model encompasses a pressure model, flow model, friction model, and bumper (stops) contact model. After development, the model parameters were derived from a BMW front strut (series 7) using geometric characteristics and a Quansi-static compression test. The damper was then validated using a multitude of simulation software and with a RMS filter applied, the model came to around 5% error.

Flow through the main valve of the shock absorber is a complex problem when doing research surrounding a shock absorber. This is due to difficulty in modeling flow through the main valve because of factors such as pressure loss due to friction. *L. Lang* [16] highlights different methods used by authors to determine flow through the orifice of main valves. He makes mention of Stone who developed a model of flow through the piston that was nondimensional by dividing the valve exit and entrance terms encompassing pressure and diameter. This method was nonacceptable as it undershot the value by 10-20%. The next method *L. Lang* [16] makes mention of was Oki's method and was also not acceptable as there were no theoretical considerations. The authors settle on Steber and Romer's method as it agreed with predicted values very well. *L. Lang* [16] concludes in this chapter that the most suitable equations for simulation purposes would be those shown in figure 11 & 12.

$$m\ddot{y} + C_V\dot{y} + Ky + \begin{Bmatrix} K_c y, & \text{for } y < 0 \\ 0, & \text{for } y > 0 \end{Bmatrix} = \Delta p A_V + C_f\, \rho\, \frac{Q^2}{A_{in}} - F_{sp} \qquad (3\text{-}12a)$$

Figure 11: L. Lang [16]

$$F_V = \frac{A^2{}_{out}}{A_{in}}\, 2C_D{}^2 \Delta p$$

Figure 12: L. Lang [16]

A widely accepted way to test/validate the effects of tuning is to use a quarter car model. *Chris Boggs et al* [17] develop the quarter car study for performance vehicles. The quarter uses the nonlinear shock as the input to a linear model encompassing the road, a spring, the shock, and two masses. This system centralizes the force from the upward force from the road and the downward forces from aero. After the author developed the quarter car model equations, he was able to simulate this using a testing rig and observe the data from simulation in MATLAB vs the test rig. He found that the two models agreed with each other quite well.

Tunability of dampers is always a concern for most applications where dampers are used. This creates a heavy emphasis on bench testing as well as models designed to save companies time testing absorbers and install a close tune to the desirable one right away. The seven-post shaker test is another popular test used by race teams and automotive manufacturers to accomplish this expedited tuning. *H. Kowalczyk* [18]elaborated on the seven-post shaker model as well as the quarter car model to validate this testing. The author highlights the simplified quarter car model and explains the variables involved. One of the main characteristics of the quarter car model is the damping coefficient seen in figure 13 which is used to describe the nonlinear part of the damper.

$$R = \frac{1}{2} c_s (\dot{x}_t - \dot{x}_s)^2 + \frac{1}{2} c_t (\dot{x}_r - \dot{x}_t)^2$$

Figure 13: H. Kowalczyk [18]

The further support of the particle performance application of dampers is helpful to gain insight and reinforcement into the investigations of dampers and the benefit in the commercial and racing world. *C. Smith* [19] provides a widely acknowledged and accepted overview of race car adjustability which includes dampers. His book includes definitions for a wide range of shock absorbers, characteristics to look for in each, as well as the trade-offs in each one.

Shock absorber design is another subject area which can be helpful to gain knowledge in when modeling, validating, and testing dampers. *C. Smith* [20] provides the same great overview as he did with the tunable aspects of the damper in *C. Smith* [20] but applied to engineering concept which observe shock absorber selection and adjustment.

MATERIALS & METHODS:

Rebuilding the shock absorber is a complex process that must be done repeatedly for this research. This research uses Fox Float 3s which are monotube air over hydraulic shocks. The monotube, meaning the main valve is the only passage for air flow inside the absorber (figure 22)

The process begins with discharging air out of the air spring at the top of the shock, and sliding down the air chamber so it is no longer in the way. Then the nitrogen at the bottom of the shock absorber is discharged with a hypothermic needle. Once discharged, a modified Toyota passenger CAM tool (figure 14) was used to open the body of the shock via the two recessed pin holes on the main bearing. The main bearing assembly was pulled up and out of the shock absorber body and put into a vice so that the shims could be accessed. The nut on the bottom of the main valve(piston) was removed and then the shim stack could be modified and inspected.

Upon the arrival of the Fox Float three different shocks were tested on the test rig with factory tune-up inside to gain a baseline characteristic curve (figures 24, 25 and 26). A test plan was then written to achieve the data necessary to begin physical analysis of the dampers and achieve a mathematical model. (figures 20-22) The initial test plan was to remove shims of equal diameter and thickness in even intervals to observe the severity that the shims had on the damping curve, testing on the dynamometer after each adjustment. After experimentation, however, it was observed that the diameter and thickness could not be held as constant as there are not enough of the same shims in the factory tune to accomplish this. To combat not being able to keep same variables, the changes were tracked and kept as constant as possible.

This research utilizes a CTW RD3 shock dynamometer (figure 23). This testing rig encompasses a linear potentiometer for displacement (figure 19) and velocity derivation, a load cell to measure shock force (figure 17), and a thermocouple mounted on the linear rail to capture shock temperature. The purpose of a shock dynamometer is to capture the characteristic curve (force vs velocity) of the shock absorber. The shock is mounted on an adjustable top cross bar via a pin (figure 18). That crossmember is mounted on a set of linear rails. A load cell is a device that uses a series of strain gauges to differentiate resistance and derive a load. A linear potentiometer varies voltage linearly and sends the signal through a calibration equation to derive distance. A Thermocouple, like the load cell, varies resistance in the presence of temperature. The CTW software uses the time available through the program and the linear distance recorded from the linear potentiometer and takes the first derivative to calculate velocity. The dynamometer uses a scotch yoke style method for power delivery from the unit's AC motor to the position that the bottom of the shock is hooked to (figure 16). Scotch yokes are a more desirable way to achieve a vertical displacement motion from the motor's rotational displacement than a traditional crank dyno (figure 15). Dynamometers utilizing a crank style to drive the shock absorber can side load the dyno slightly causing a loss in accuracy throughout the test. To test the shock absorber on the dynamometer a test plan was created using CTW Probe software. The test plan entails a 10in/s warmup of the absorber to get the oil moving and slightly increase the temperature above its static temperature. The next part of the test is a rod force test. This test is primarily for if the spring is equipped onto the shock, for this test the spring was effectively removed from the shock by disconnecting the air chamber, however; this portion of the test program was left in to increase data accuracy by removing a small amount of the seal drag or any other position-dependent forces. The dynamometer does this by moving the shock to the 0-stroke position (perpendicular position of the shock absorber) which is the theoretically highest point of force developed on the dynamometer. The rig is programmed to read these forces statically so there are no position dependent forces to influence of damping present. The shock dynamometer then creates a force V. displacement diagram and effectively removes it from the data present posttest. Next, the dynamometer completes three cycles at three velocities and records the middle cycle of each one to minimize the effect of the dynamometer accelerating or decelerating to the desired velocity abnormally.

Figure 14: Modified Toyota CAM tool used to separate the main bearing and the body.

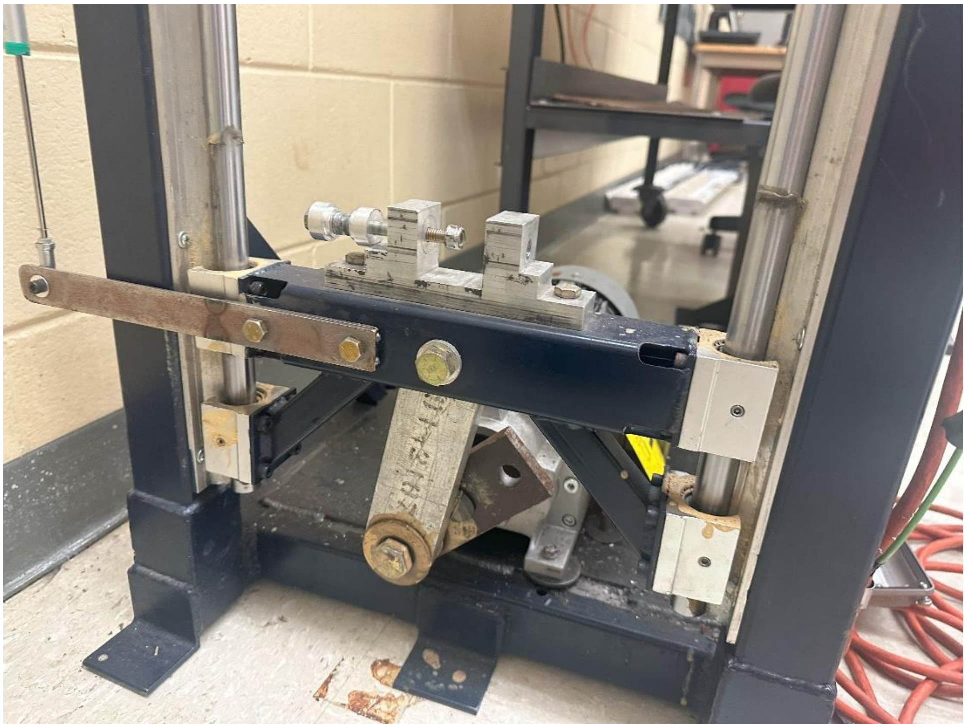

Figure 15: Traditional crank style dynamometer.

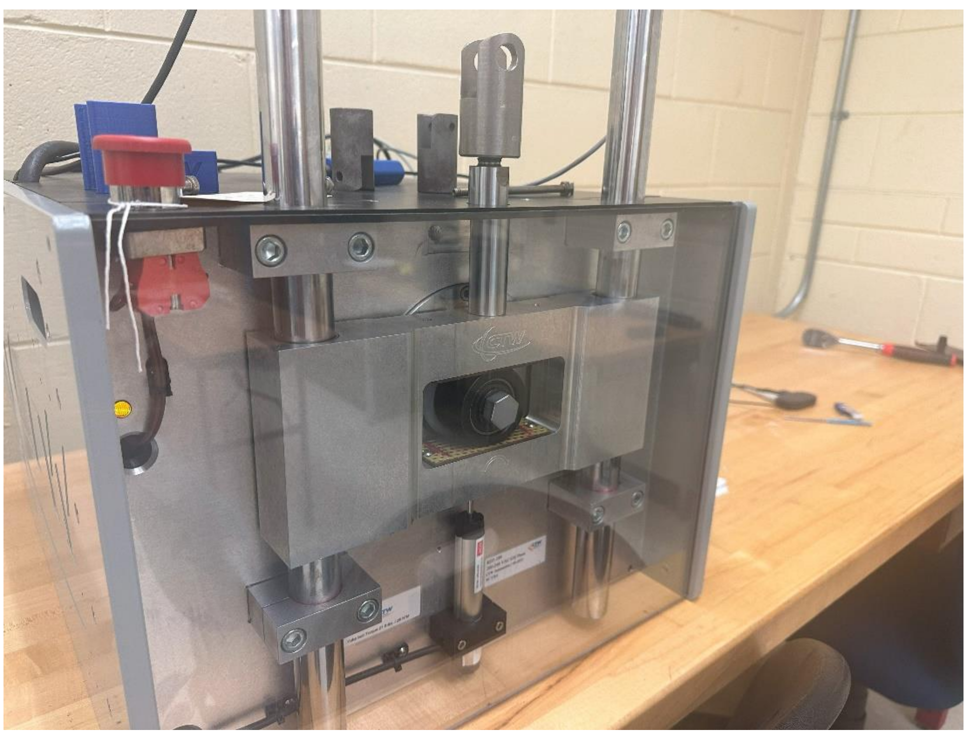

Figure 16: Scotch yoke on CTW dynamometer.

Figure 17: S-beam type load cell on CTW dynamometer.

Figure 18: Top crossmember on CTW dynamometer.

Figure 19: Linear potentiometer on CTW dynamometer.

Shock 4 Factory Build

	OD (in)	Thickness
Thick	1.2275	0.08
Shim 2	0.7	0.01
Shim 3	0.8	0.01
Shim 4	0.9	0.01
Shim 5	1	0.01
Shim 6	1.1	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
Shim 10	1.3	0.008
Shim 11	1.3	0.008
Shim 12	Bleed	
Piston		
Shim 13	1.1	0.008
Shim 14	1.1	0.008
Shim 15	1	0.01
Shim 16	1	0.01
Shim 17	0.9	0.01
Shim 18	0.8	0.01
Shim 19	0.7	0.01
Washer		
Nut		

Build 1

	OD (in)	Thickness
Thick	1.2275	0.08
Shim 2	0.7	0.01
Shim 3	0.8	0.01
Shim 4	0.9	0.01
Shim 5	1	0.01
Shim 6	1.1	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
Shim 10	1.3	0.008
Shim 11	1.3	0.008
Shim 12	Bleed	
Piston		
Shim 13	1.1	0.008
Shim 14	1.1	0.008
Shim 15	1	0.01
Shim 16	1	0.01
Shim 17	0.9	0.01
Washer		
Nut		

Build 2

	OD (in)	Thickness
Thick	1.2275	0.08
Shim 2	0.7	0.01
Shim 3	0.8	0.01
Shim 4	0.9	0.01
Shim 5	1	0.01
Shim 6	1.1	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
Shim 10	1.3	0.008
Shim 11	1.3	0.008
Shim 12	Bleed	
Piston		
Shim 13	1.1	0.008
Shim 14	1.1	0.008
Shim 17	0.9	0.01
Washer		
Nut		

Build 3

	OD (in)	Thickness
Thic	1.2275	0.08
Shim 2	0.7	0.01
Shim 3	0.8	0.01
Shim 4	0.9	0.01
Shim 5	1	0.01
Shim 6	1.1	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
Shim 10	1.3	0.008
Shim 11	1.3	0.008
Shim 12	Bleed	
Piston		
Shim 13	1.1	0.008
Washer		
Nut		

Figure 20: Rebound (Shock 4) test plan/schedule.

Shock 1 Factory Build

	OD (in)	Thickness
Thick	1.2275	0.08
Shim 2	0.7	0.01
Shim 3	0.8	0.01
Shim 4	0.9	0.01
Shim 5	1	0.01
Shim 6	1.1	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
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Shim 11	1.3	0.008
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Shim 14	1.1	0.008
Shim 15	1	0.01
Shim 16	1	0.01
Shim 17	0.9	0.01
Shim 18	0.8	0.01
Shim 19	0.7	0.01
Washer		
Nut		

Build 1

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Shim 2	0.7	0.01
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Shim 6	1.1	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
Shim 12	Bleed	
Piston		
Shim 13	1.1	0.008
Shim 14	1.1	0.008
Shim 15	1	0.01
Shim 16	1	0.01
Shim 17	0.9	0.01
Shim 18	0.8	0.01
Shim 19	0.7	0.01
Washer		
Nut		

Build 2

	OD (in)	Thickness
Thick	1.2275	0.08
Shim 2	0.7	0.01
Shim 3	0.8	0.01
Shim 4	0.9	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
Shim 12	Bleed	
Piston		
Shim 13	1.1	0.008
Shim 14	1.1	0.008
Shim 15	1	0.01
Shim 16	1	0.01
Shim 17	0.9	0.01
Shim 18	0.8	0.01
Shim 19	0.7	0.01
Washer		
Nut		

Build 3

	OD (in)	Thickness
Thicc	1.2275	0.08
Shim 2	0.7	0.01
Shim 7	1.25	0.01
Shim 8	1.25	0.01
Shim 9	0.7	0.006
Shim 12	Bleed	
Piston		
Shim 13	1.1	0.008
Shim 14	1.1	0.008
Shim 15	1	0.01
Shim 16	1	0.01
Shim 17	0.9	0.01
Shim 18	0.8	0.01
Shim 19	0.7	0.01
Washer		
Nut		

Figure 21: Compression (Shock 1) test plan/schedule.

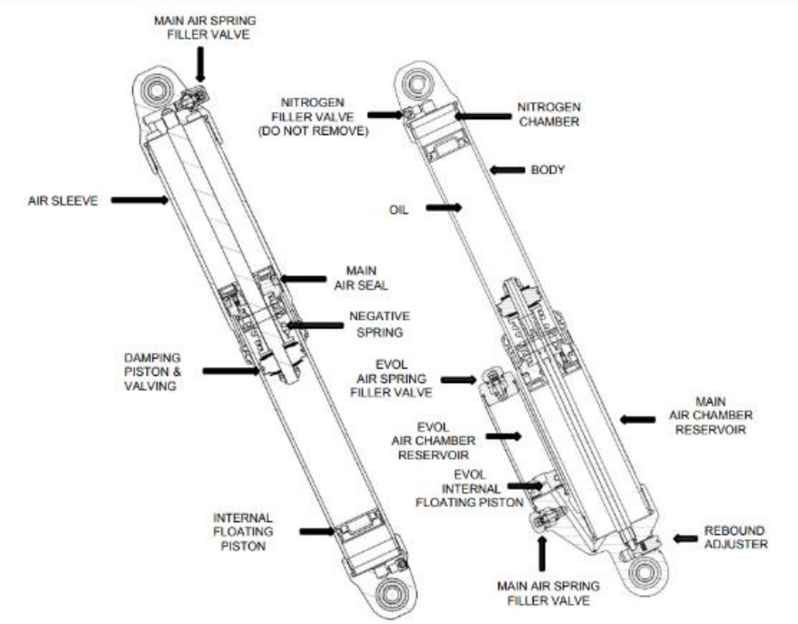

Figure 22: Fox Float 3 Diagram

Figure 23: CTW RD3 Dynamometer

RESULTS:

All results are presented in graphical form. Figures 24, 25 and 26 show the characteristic curve at the factory setting for three different shock absorbers. Figure 27 indicates the data for different settings at the compression side including the original factory setting for one of the shocks. Similarly, figure 28 shows the data for different settings at the rebound side including the original factory setting for a different shock. Linear trends are predominant for all the settings. Finally, figures 29 and 30 indicate the change in damping coefficient with respect to number of shims for both compression and rebound. Even there the overall linear trend is obvious from the R squared value.

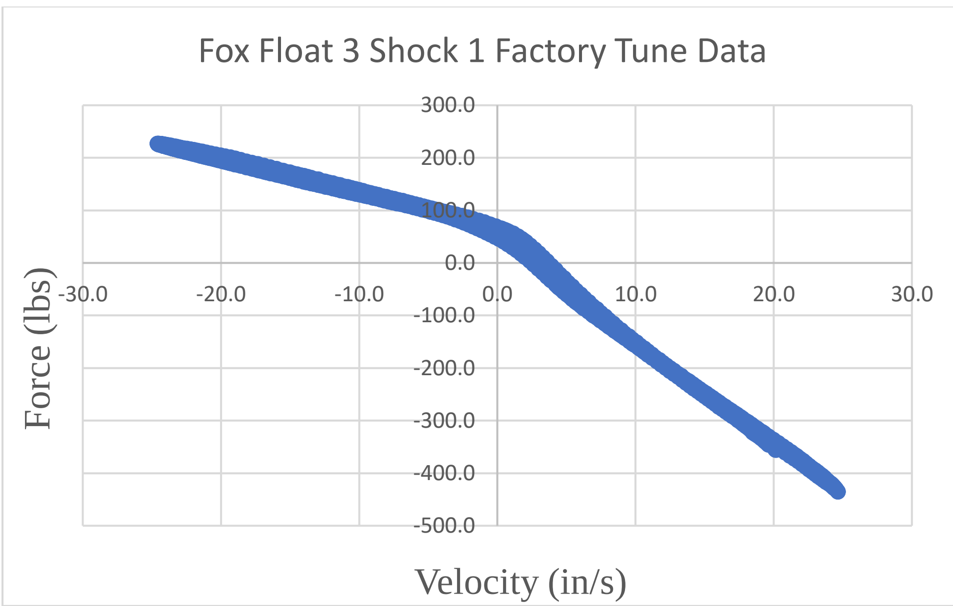

Figure 24: Fox Float 3 Shock 1 Factory Tune Data

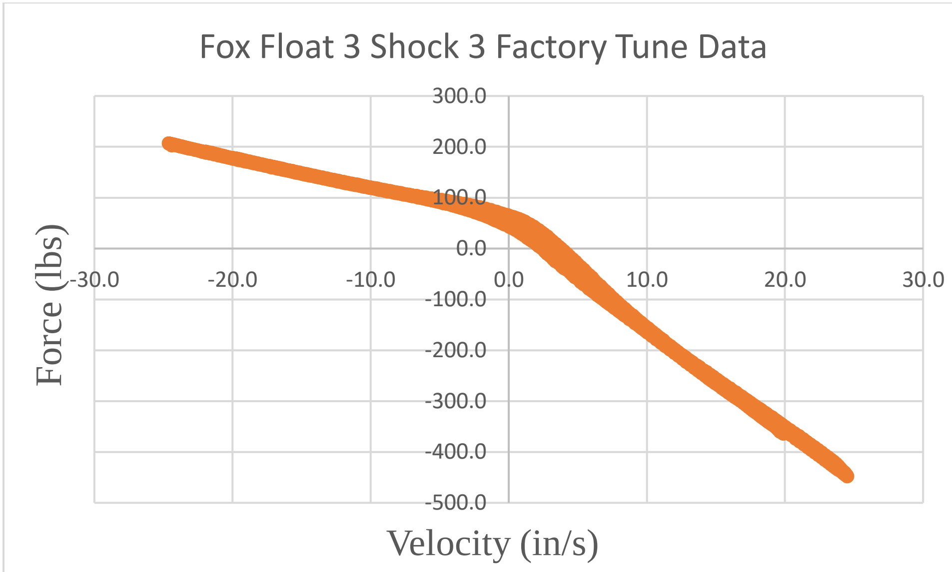

Figure 25: Fox Float 3 Shock 1 Factory Tune Data

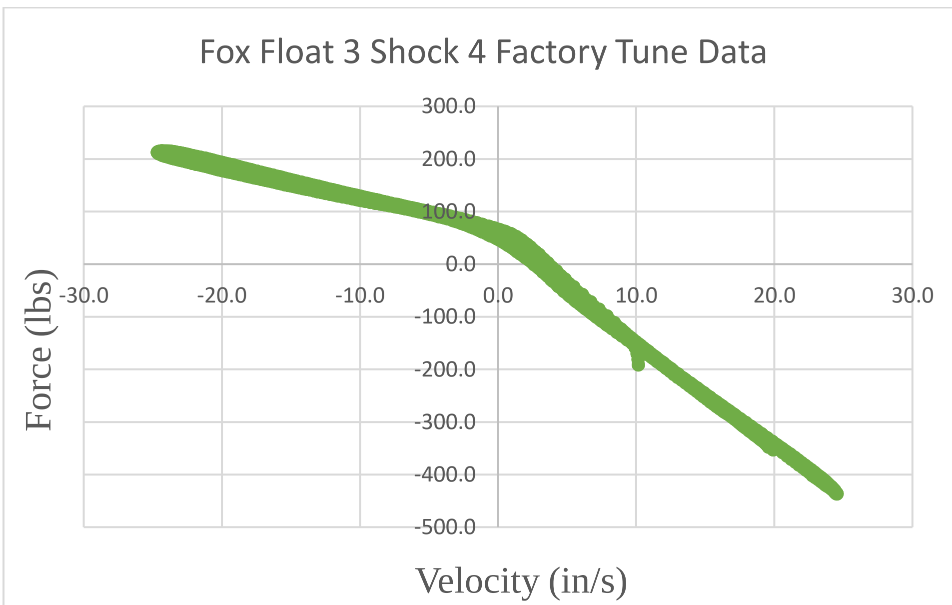

Figure 26: Fox Float 3 Shock 4 Factory Tune Data

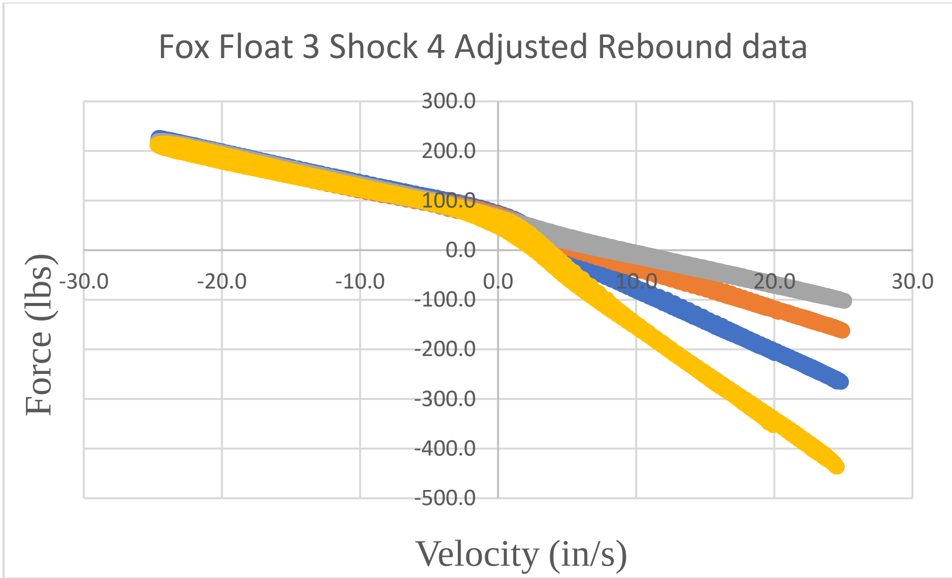

Figure 27: Fox Float 3 Shock 4 Adjusted Rebound Data

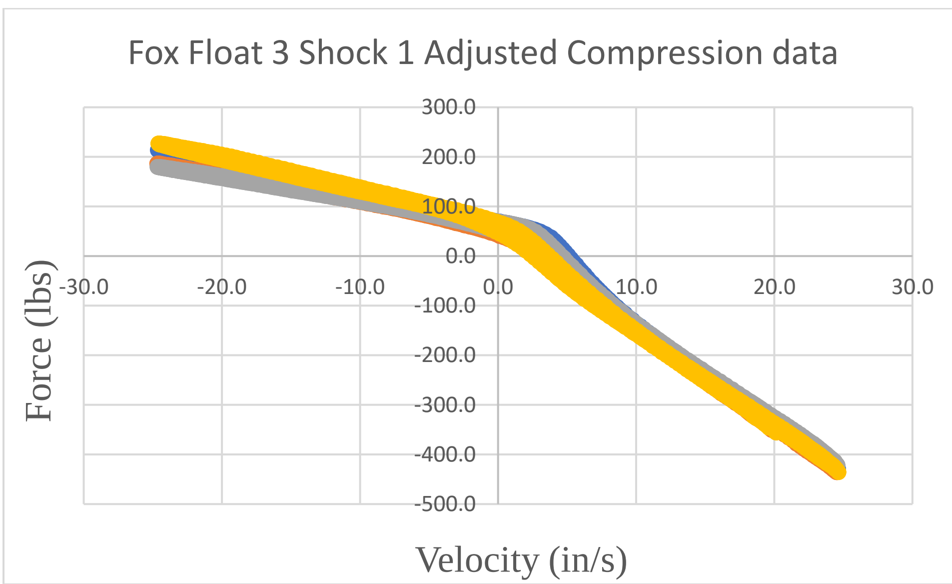

Figure 28: Fox Float 3 Shock 1 Adjusted Compression Data

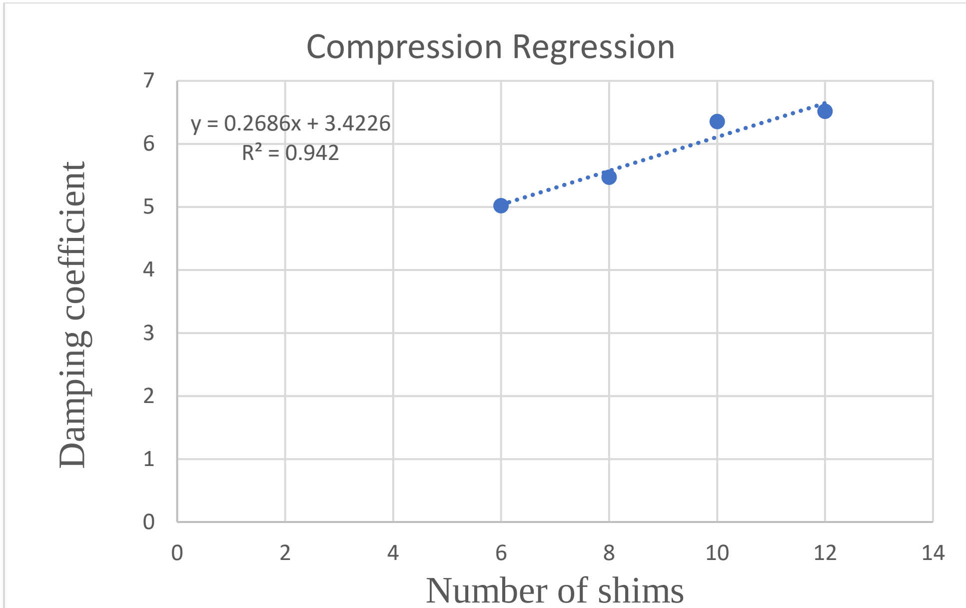

Figure 29: Compression Regression Analysis

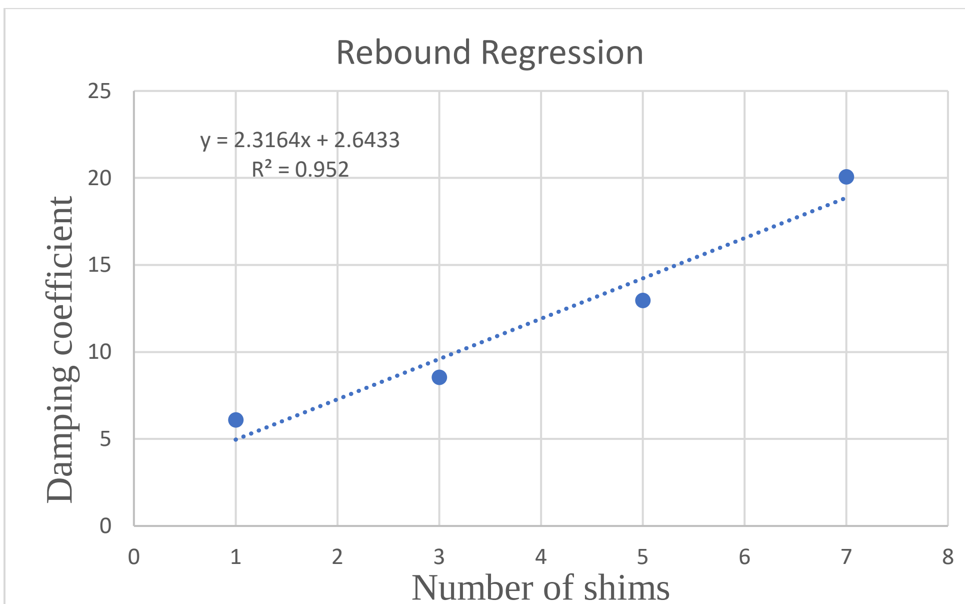

Figure 30: Rebound Regression Analysis

DISCUSSION:

To establish the baseline, three different shock absorbers were tested with the original company setting with twelve shims on the compression side and seven shims in the rebound side. Based on the regression analysis, both compression and rebound zones for all the shocks are quite linear with correlation coefficient close to 0.99. In case of compression, the average damping coefficient is 6.06 lb /(in/sec) with a standard deviation of 0.42 lb/(in/sec). In case of rebound, however, the average is 20.35 lb/(in/sec) with a standard deviation of 0.83 lb/(in/sec). Hence, the shocks show higher stiffness under rebound in factory condition.

Two shocks were selected for further investigation under different tune up settings which is accomplished by removing preplanned number of shims. One of them was used for conducting different settings under compression while the other one was used for rebound. The experimental results again indicate that irrespective of the level of tuning, shocks have linear characteristics curves, the force is linearly proportional to the velocity. In all cases, the correlation coefficients are close to 0.99. Hence the damping coefficient can be viewed as constant for a given setting. This nature is quite in contrast with some of the references where the characteristic curve shows significant nonlinearity and in some cases power relationship. Four different tune up conditions were tried for each compression and rebound cycle including the initial factory setting. In each case the shims were removed in steps and the experiment was conducted. Although, the thickness of the individual shims varies a little with a mean of 0.1 inch, they can be treated identical all practical purposes with respect to the resistance they offer to the fluid flow from one chamber to the other (figure 27 & 28. The experiment reveals that the number of shims dictates the damping coefficient, and it is quite linear as well. In the compression side each shims adds 0.27 lb/(in/sec) to the overall damping coefficient. In the rebound side each shim adds 2.32 lb/(in/sec) to the overall damping coefficient. The data also indicates there are significant intercepts at zero, meaning even without any shims there are some resistances, nevertheless.

CONCLUSION:

The characteristic curve, force versus velocity, of shock absorber is an important aspect with respect to vehicle performance as well as human comfort angle. Identical model of three Fox Float shock absorbers were tested using the dyno available in the mechanical engineering laboratory at Georgia Southern University. They are also tested for different tune up setting by removing predetermined number of shims both in compression and rebound zone. As expected, the damping coefficient, the slope of the characteristic curve, is different for compression and rebound with rebound being stiffer. However, based on the data collected, it is quite apparent that the relationship between the force and the velocity is linear with high correlation coefficient. It has also been observed that the damping coefficient is proportional to the number of active sims, at least within the working range of the tune up settings.

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