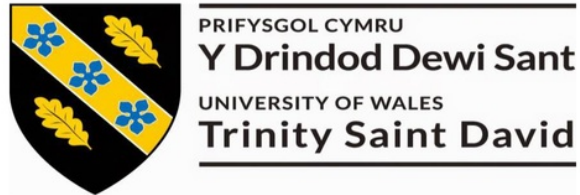


A Study into the Damping Performance and Transmissibility Characteristics of WP's Cone Valve System

by Amy Cosford

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Amy Cosford
P150093
Module Code: M6X00417
Supervised by Dr Owen Williams
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This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has lead to the award of a degree.

SIGNED:

Amy Cosford

DATED:

30/04/2019

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Abstract

Alternative methods of damping have been an ongoing development within performance suspension design. The WP Cone Valve damping system is one which has recently been introduced to the field of motorcycle suspension, and little has been exposed regarding the performance benefits and damping characteristics of such a system. This dissertation explores both the physical attributes that each system variable offers to damping force properties, as well as presenting methodology for the determination of perceived ride comfort.

A damper dynamometer was used in the testing phase to gather data on the forces produced by compression and rebound circuits. Changes in cone valve angle and spring stiffness resulted primarily in alterations to compression damping, whereby the gradient of mid-to-high velocity damping forces increased with stiffer spring rates and steeper angled cone valves. From the raw data output by the damper dynamometer, damping ratios were determined as a function of velocity.

Represented by a single-degree-of-freedom model, magnitudes of displacement and acceleration transmissibility were then calculated as a product of road input frequency and damping ratio. Informed by research conducted in the literature review and 3D graphs of transmissibility characteristics, the author was able to determine a baseline cone valve configuration that theoretically provided the closest ideal properties of ride comfort.

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1. Introduction

Damping performance of the modern suspension era has primarily been due to a valve and piston assembly inside a telescopic fork design, whereby a number of thin metal shims are stacked atop a bored-through piston fixed to a damper rod. In early designs the damper rod was a simple hollow rod with drilled orifices allowing the passage of oil, however with the performance increase of modern motorcycles, suspension and handling must be able to keep up, therefore improved methods of controlling the dynamic characteristics must be developed. Typically, oil flows through the piston and is inhibited by a number of shims arranged in a tapered stack, whereby the flow is choked until the velocity and resultant force is sufficient to bend the shim stack and allow increased oil flow. Piston designs and shim stack configurations vary between manufacturers, but this is known to be the standard performance setup.

1.1 Aims

In 2016 WP Suspension released a set of telescopic forks to fit parent company KTM's off-road motorcycle range, with a new patented damping design named the *Cone Valve*. The innovative design claimed to offer an almost infinite amount of tuning possibilities and was rid of the complexities of trial and error shim stack tuning. It is the aim of this project to thoroughly investigate the specific internal mechanisms controlling the damping, and measure damping performance of the cone valve design using a damper dynamometer. With the additional WP tuning kit there are 3 optional cones of differing slope angle, as well as 10 optional spring rates. Therefore the damping properties of the system fitted with each respective cone will be analysed alongside the effects of increasing or decreasing spring rate.

1.2 Stages

The project itself will be split into two significant stages. Firstly, the investigation phase whereby the forks will be stripped down to their individual components to understand the specific internal mechanisms that control damping. The forks will then endure intense testing on a SPA damper dynamometer, with results of each cone valve configuration plotted graphically with respect to force, velocity, and displacement. It is the objective of this stage to examine certain combinations of valve configuration and conduct an analysis of the effects of cone angle and spring rate respectively on the overall damping performance across the range of available compression and rebound adjustment.

The WP Cone Valve dynamometer data will subsequently be compared against data from standard WP forks taken out of a Husqvarna TE250i. Both pairs of forks are interchangeable and therefore offer an insight into the differences between standard production WP forks and the cone valve system.

Secondly, the post-processing stage using mathematical methods to understand how damping force characteristics influence the perceived ride comfort. Since ride quality is a strongly subjective matter dependent on rider's personal preferences, it cannot be confidently said that there exists a singular ideal suspension damping configuration that provides the best performance. Rather, properties of damping force can be analysed in regard to the effect that various cone valve combinations have on the movement and accelerations experienced by the sprung mass and rider. These characteristics will be determined by assessing how the transmissibility of the system changes with each variation of valving.

2. Literature Review

As a new and alternative damping method, the cone valve forks have experienced little external investigation, and as such limited information is available regarding how the system works. However studies have been conducted into the performance of various damping systems, namely one by Neal et al (Neal, et al., 2015) which discussed the critical aspects of tuning dampers for production vehicles. The paper categorised performance into ride and handling attributes such as stability and dynamic motion control, damper design aspects, and tuning principles. It summarised that an ideal damper should employ digressive force-velocity properties, shown in Figure 1, such that low-speed velocities are capable of maintaining control under dynamic pitch and roll conditions, whereas high-speed damping forces are minimised to avoid a harsh ride. This conforms with the generalised approach to modern damping tuning as suggested by Thede et al (Thede & Parks, 2010).

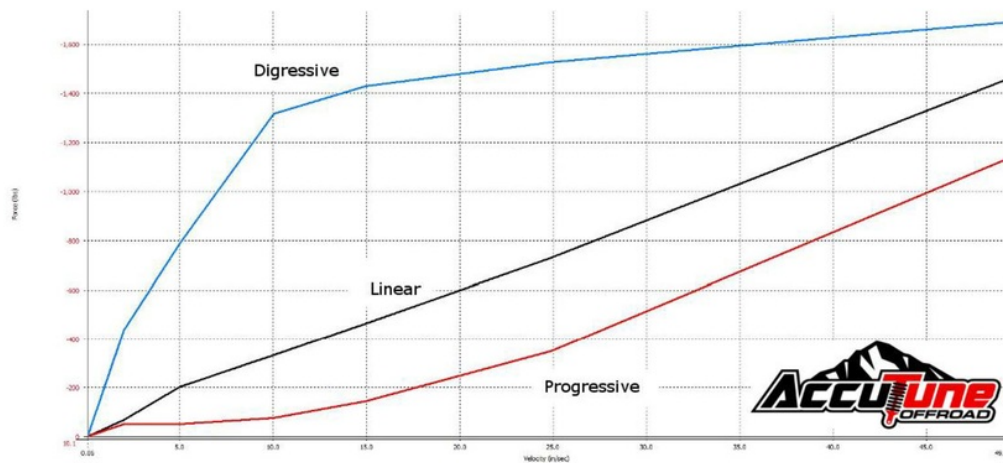


Figure 1: Force-velocity characteristics of Digressive vs Linear vs Progressive Damping Properties (Raker, 2017)

However the paper also suggested that opposing damping forces of compression and rebound should be balanced whereby neither force dominates. It demonstrated that balanced damping offers a smoother transition of suspension reversal, desirable for ride comfort. This theory differs from what is employed in modern damper design where rebound circuits tend to offer higher damping forces than compression.

Balike et al's study into damper asymmetry (Balike, et al., 2010) assessed this hypothesis and investigated the significance of biased damping on a quarter-car model in response to bump and pothole road excitations. It concluded that a higher ratio of rebound to compression damping resulted in the attenuation of peak sprung mass accelerations under bump excitations, though yielded higher accelerations when experiencing pothole excitations. The optimal damping parameters differed

considerably for either scenarios and the study thusly concluded that damper design necessitates a compromise between damping for both inputs.

Balike's study didn't ultimately state whether a higher ratio of rebound damping force was beneficial, just that the system operates in a complex manner and damper design must attempt to provide sufficient control for both compression and rebound circuits. Both Balike's and Neal's studies investigated damping performance when applied to four-wheeled vehicle systems. Though suspension obviously plays a very significant part in such vehicles, the importance of suspension for two-wheeled motorcycles is paramount to their stability, steering ability, load transfer and ride characteristics.

A study by Mahindra Two Wheelers Ltd took a holistic approach to ride dynamics in regards to the transmissibility behaviour of motorcycle suspension over road input frequencies (Shaikh, et al., 2013). Transmissibility is described as the ratio of road input frequency to output frequency, such that low values indicate a low transfer of oscillations and therefore increased ride comfort. The study looked into the influence of damping properties, namely damping ratio, on the displacement transmissibility characteristics of a quarter-car model system representing a rear suspension unit. Utilising these mathematical methods in conjunction with statistical analysis of logged suspension data, it concluded that the benefits of higher damping ratios at low-frequency road inputs should be combined with lower ratios at high-frequency inputs. A crossover point shown on figure 2 could be determined based upon the statistical analysis which highlighted the frequencies most experienced on varying road surfaces, such that after this point the system should exhibit lower damping ratios, correlating with the characteristics of a digressive damping profile.

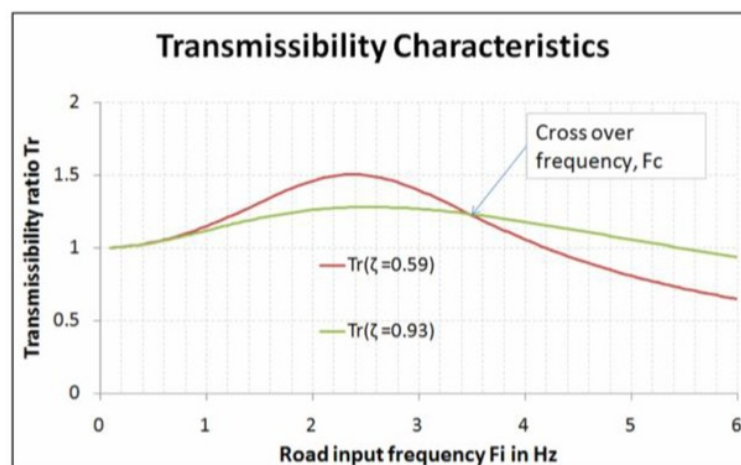


Figure 2: Transmissibility Characteristics (Shaikh, et al., 2013)

Shaikh's analytical methods back up what is proposed by Neal that digressive damping curves offer increased ride comfort, and also provides a method in which the change-over in damping ratio could be calculated.

What was missing in Shaikh's study was the effect that damping ratio has on the acceleration transmissibility. Cossalter implied that for ride comfort the transference of accelerations to the sprung mass is more significant, as the sensation of comfort is linked to the forces felt through the body (Cossalter, 2006). He outlined that accelerations are amplified at low-frequency inputs when damping ratio takes a lower value, however an ideal damping ratio can be determined whereby these accelerations are minimised. The process of this theorised ideal damping ratio will be further discussed in the relevant subsection of this report.

The above research, primarily that of Sheikh and Cossalter's work, provides a benchmark for which the analysis of the damping properties provided by WP's cone valve forks can be conducted. The force-velocity graphs taken from dynamometer testing can be thoroughly analysed both directly in terms of the efficiency of the various damping circuits, as well as the post-processed damping ratio profiles and transmissibility characteristics.

In order for a full understanding of the post-processing stage it is necessary to recognise the principles behind the calculations.

2.1 Quarter-Car Model

The quarter-car model is used in a number of studies to traditionally conceptualise how the suspension systems are connected. Suspension functions to control a vehicle under dynamic loads such as braking, accelerating and cornering with use of a helical spring and hydraulic viscous damper. The conventional dynamic model of a motorcycle is commonly referred to as a half-car model, with degrees of freedom allowing for vertical displacements in the Z-direction as well as rotational pitching displacements around the centre of gravity. Since effects of rear suspension are negated in this study a quarter-car model can instead be introduced with 2 degrees of freedom.

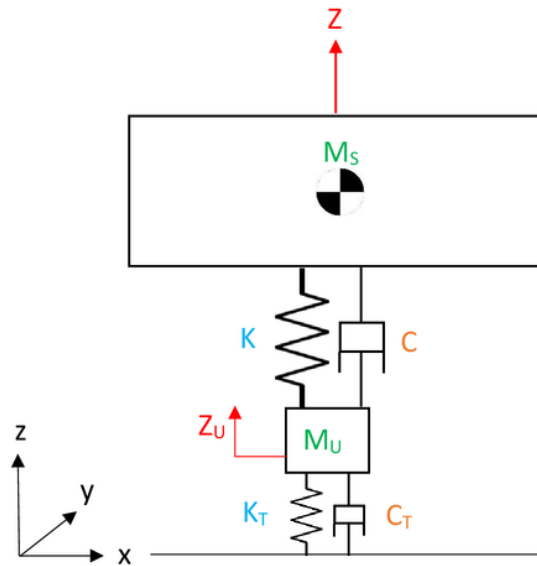


Figure 3: Quarter-Car Model

M_s = Sprung mass (kg)

M_u = Unsprung mass (kg)

C = Hydraulic damping constant (Ns/m)

C_T = Tyre damping constant (Ns/m)

K = Helical spring rate (N/m)

K_T = Tyre spring rate (N/m)

Z = displacement of sprung mass (m)

Z_u = displacement of unsprung mass (m)

However the effects of tyre deflection and the influence of the unsprung mass pose complications in determining transmissibility characteristics. Therefore it was decided to simplify the above model to a singular degree-of-freedom (DOF) model to make unsprung mass and tyre properties negligible, such that an equation of motion can be determined from the mass, spring rate and damping constant.

2.1.1 Equations of Motion

Newton's First Law of motion states that an object at rest remains at rest (Newton, 1729), and as such the forces acting upon a system undisturbed must be equal in magnitude and opposite in direction. Applying this theory to the 1DOF model, the deceleration force of the systems mass due to gravity must be equal to the opposing forces of the spring and damper systems, such that:

$$-ma = cv + kx \quad [1]$$

$$\therefore ma + cv + kx = 0 \quad [2]$$

This can subsequently be written in the form of a second order differential equation whose roots determine the overall characteristics of the suspension system as a function of time.

$$m \frac{d^2x}{dt^2} + c \frac{dx}{dt} + kx = 0 \quad [3]$$

For a homogenous second order differential equation whereby the result is equal to zero and all coefficients are constants, if one is to make the systems position equal to the exponential function such that:

$$y = e^{mt} \quad [4]$$

Where t is time and m is a secondary variable, it can consequently be derived that:

$$\frac{dy}{dt} = me^{mt} \quad [5]$$

and

$$\frac{d^2y}{dt^2} = m^2 e^{mt} \quad [6]$$

Therefore, equation [3] can be written as:

$$a \cdot m^2 e^{mt} + b \cdot me^{mt} + c \cdot e^{mt} = 0 \quad [7]$$

Where constants a, b, and c, represent the mass, damping constant, and spring rate, respectively. This can be further simplified, thus:

$$e^{mt}(am^2 + bm + c) = 0 \quad [8]$$

Ascertaining the roots of the quadratic equation nested in equation 8, commonly referred to as the auxiliary equation (Grayling, 2014), enables the determination of the specific damping properties of the system in regard to the critical damping value. If m is found to have real and different roots, the system is overdamped with and equation 9 applies:

$$y = Ae^{m_1 t} + Be^{m_2 t} \quad [9]$$

Where A and B are constants dependent on the starting conditions of the system. Real and equal roots of m signify a critically damped system with a damping ratio equal to 1, such that:

$$y = (A + Bt)e^{mt} \quad [10]$$

Finally, complex roots indicate an underdamped system, therefore:

$$y = e^{\alpha t}(A\cos(\beta t) + B\sin(\beta t)) \quad [11]$$

Where the resultant roots are equal to

$$y = \alpha \pm i\beta \quad [12]$$

(Tseng, 2006)

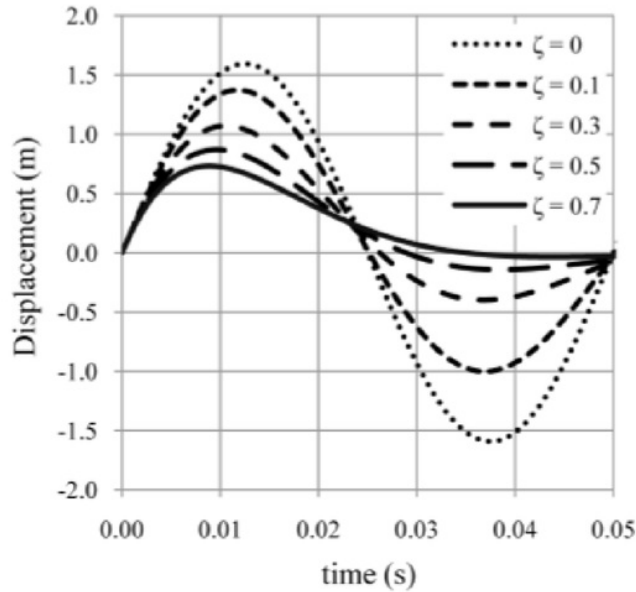


Figure 4: The effect of varying damping ratio on system displacement plotted against time (Sweeney, 2011)

Figure 4 shows the time response of a system with various damping ratios, given the Greek symbol zeta ζ . Values of zeta less than one correlate with an underdamped system, whereas above 1 represents the reaction of an overdamped system. At a zeta value of zero the system has no damping force and simply resonates at its natural frequency.

2.1.2 Critical Damping

Equation 10 represents the solution for a suspension system that is described as critically damped. A system with critically damped properties is one which when perturbed returns to its equilibrium in the shortest time and without any oscillation. Its value is given by the equation:

$$c_{crit} = 2\sqrt{km} \quad (\text{Cossalter, 2006}) [13]$$

Substituting this into the quadratic equation gained from equation 8, where k and m are the constants c and a, respectively, it can be proved that the outcome produces real and equal roots. Equation 13 can be rewritten as:

$$c_c = 2m \sqrt{\frac{k}{m}} \quad [14]$$

$\sqrt{\frac{k}{m}}$ is the natural frequency at which a system will oscillate with the absence of an external damping force, and is given the symbol ω , therefore critical damping can be said to be:

$$c_c = 2m\omega \quad [15]$$

Magnitude of damping force is often described as a ratio to critical damping, whereby a damping ratio of 1 is known to be critical. It is given the symbol zeta ζ and the formula for which is seen below.

$$\zeta = \frac{c}{c_{crit}} \quad [16]$$

Which can be rewritten as:

$$\zeta = \frac{c}{2m\omega} \quad [17]$$

$$\therefore c = 2m\omega\zeta \quad [18]$$

From equation 18 a new equation of motion can be rewritten as follows:

$$m \frac{d^2x}{dt^2} + 2m\omega\zeta \frac{dx}{dt} + kx = 0 \quad [19]$$

Dividing through by mass,

$$\frac{d^2x}{dt^2} + 2\omega\zeta \frac{dx}{dt} + \frac{k}{m}x = 0 \quad [20]$$

$$\frac{d^2x}{dt^2} + 2\omega\zeta \frac{dx}{dt} + \omega^2x = 0 \quad [21]$$

Hence, the quadratic gained from equation [8] can be written as:

$$m^2 + 2\omega\zeta m + \omega^2 = 0 \quad [22]$$

Using the quadratic equation:

$$m = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \quad [23]$$

$$m = \frac{-2\omega\zeta \pm \sqrt{(2\omega\zeta)^2 - 4\omega^2}}{2} \quad [24]$$

$$m = \frac{-2\omega\zeta \pm \sqrt{4\omega^2\zeta^2 - 4\omega^2}}{2} \quad [25]$$

$$m = \frac{-2\omega\zeta \pm 2\omega\sqrt{\zeta^2 - 1}}{2} \quad [26]$$

$$m = -\omega\zeta \pm \omega\sqrt{\zeta^2 - 1} \quad [27]$$

$$m = \omega \left(-\zeta \pm \sqrt{\zeta^2 - 1} \right) \quad [28]$$

If $\zeta = 1$,

$$m = \omega(-1 \pm \sqrt{1 - 1}) \quad [29]$$

$$m = -\omega \quad [30]$$

However a natural frequency cannot be negative therefore:

$$m = \omega \quad [31]$$

The solution thus has only one answer and therefore the roots are proven to be real and equal.

2.2 Frequency

Methods of damping are necessary in order to control the vertical displacement of the system as it is perturbed. Left undamped and the system would oscillate at its natural frequency until it returns to equilibrium. A motorcycles natural frequency ω_n , commonly referred to as ride frequency, is dependent on the spring rate, k , and mass of the system, m , given by equation 32. It is common for manufacturers to recommend spring rates based on a rider's body mass with an approximate ride frequency in mind such that the motorcycle behaves as desired.

$$\omega_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}} = \text{Hz} \quad [32]$$

Suspension inputs are also commonly characterised with a value of frequency which is dependent on both the velocity and displacement of the system, calculated by equation 33.

$$\omega = \frac{v}{\pi d} \quad [33]$$

From this it can be determined that frequency increases as the displacement gets smaller or as the velocity gets larger. However damping force is primarily a product of velocity rather than frequency, and therefore at a given frequency the damping doesn't have a singular value. Hence it is necessary in the post-processing stage to produce 3D graphs of damping characteristics as a function of both velocity and frequency.

2.2.1 Transmissibility Ratio

Transmissibility represents the ratio of the magnitudes of displacement or force transmitted to a structure by an exciting force such as road irregularities. As explored by Cossalter and Shaikh, transmissibility plays a crucial role in ride feel and the perceived harshness of road textures and undulations. By applying the principles of harmonic oscillation to the previously derived equation of motion it can be said that according to figure 5:

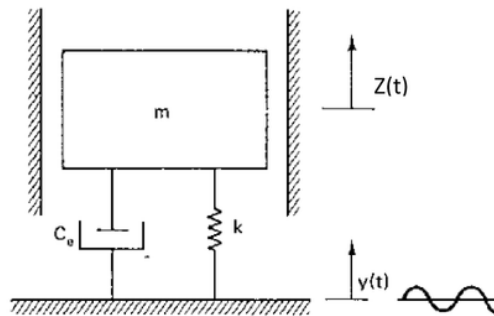


Figure 5: Transmissibility (Chan, 2008)

$$y(t) = y_o \sin \omega t \quad [34]$$

Whereby y_o is the amplitude of imposed motion, and ω is the natural frequency of the system. From equation [2], considering the system is affected by road irregularities gives an equation of motion equal to:

$$m\ddot{z} + c\dot{z} + kz = c\dot{y} + ky \quad [35]$$

Thus combining equations 34 and 35:

$$m\ddot{z} + c\dot{z} + kz = c(\omega y_o \cos \omega t) + k(y_o \sin \omega t) \quad [36]$$

In form equal to that found in equation 21, this can be rewritten as:

$$\ddot{z} + 2\omega_n \zeta \dot{z} + \omega_n^2 z = \underbrace{\frac{f_{0c}}{2\omega_n \zeta \omega y_o} \cos \omega t}_{z_{pc}} + \underbrace{\frac{f_{0s}}{\omega_n^2 y_o} \sin \omega t}_{z_{ps}} \quad [37]$$

Thus the steady-state solution is the superposition of the two sine and cosine solutions.

Sine Solution

To solve first for the sine part where:

$$\ddot{z} + 2\omega_n \zeta \dot{z} + \omega_n^2 z = f_{0s} \sin \omega t \quad [38]$$

For an inhomogeneous second order differential equation whereby the equation is not equal to zero, the general solution can be written in the form:

$$z = \underbrace{\tilde{z}_c}_{\text{complimentary function}} + \underbrace{z_p}_{\text{particular integral}} \quad [39]$$

(Grayling, 2014)

The complimentary function is that which is the solution to the homogenous form of equation 38 equal to zero. This can be solved using the same process shown in section 2.1.1 whereby the auxiliary equation is gained. For the particular integral, where the solution to the inhomogeneous equation is either a sine or cosine function, it is said:

$$z_p = A \cos \omega t + B \sin \omega t \quad [40]$$

Applying this to equation 38 it can be rewritten that:

$$z_{ps} = A_s \cos \omega t + B_s \sin \omega t = Z_s \sin(\omega t) \quad [41]$$

Whereby Z_s is the magnitude at which the sprung mass' motion is imposed, and the constants are equal to:

$$A_s = \frac{-2\zeta\omega_n\omega f_{0s}}{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2} \text{ and } B_s = \frac{(\omega_n^2 - \omega^2)f_{0s}}{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2} \quad [42]$$

(Institute of Engineering: SUT, n.d.)

Cosine Solution

The same process can be followed for the cosine part of the equation where:

$$\ddot{z} + 2\omega_n\zeta\dot{z} + \omega_n^2 z = f_{0c}\cos\omega t \quad [43]$$

Therefore:

$$z_{pc} = A_c\cos\omega t + B_c\sin\omega t = Z_c\cos(\omega t) \quad [44]$$

Where:

$$A_s = \frac{(\omega_n^2 - \omega^2)f_{0c}}{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2} \text{ and } B_s = \frac{-2\zeta\omega_n\omega f_{0c}}{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2} \quad [45]$$

(Institute of Engineering: SUT, n.d.)

Thus the addition of both the sine and cosine solutions gives a total solution for the amplitude of vertical displacement of the sprung mass, Z_0 .

$$Z_0 = \sqrt{\frac{f_{0c}^2 + f_{0s}^2}{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2}} \quad [46]$$

Therefore:

$$Z_0 = \sqrt{\frac{(2\omega_n\zeta\omega y_0)^2 + (\omega_n^2 y_0)^2}{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2}} \quad [47]$$

Taking y_0 and ω_n outside of the square root:

$$Z_0 = \omega_n y_0 \sqrt{\frac{(2\zeta\omega)^2 + \omega_n^2}{(\omega_n^2 - \omega^2)^2 + (2\zeta\omega_n\omega)^2}} \quad [48]$$

Simplifying again:

$$Z_0 = y_0 \sqrt{\frac{(2\zeta \frac{\omega}{\omega_n})^2 + \frac{\omega_n^2}{\omega_n^2}}{\left(\frac{\omega_n^2 - \omega^2}{\omega_n^2}\right)^2 + \left(\frac{2\zeta\omega_n\omega}{\omega_n}\right)^2}} \quad [49]$$

$$\frac{Z_0}{y_0} = \frac{(2\zeta \frac{\omega}{\omega_n})^2 + 1}{\sqrt{\left(1 - \frac{\omega^2}{\omega_n^2}\right)^2 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2}} \quad [50]$$

Therefore making r equal to the frequency ratio an equation for transmissibility can be gained describing the magnitude of displacement from an input frequency to the sprung mass.

$$T = \frac{Z_0}{y_0} = \frac{(2\zeta r)^2 + 1}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}} \quad [51]$$

The above equation is the basis of how the damper dynamometer data will be post-processed and interpreted to gain an understanding of how damping ratio affects the displacement of sprung mass.

3. Cone Valve Fork Operation

Common in certain types of telescopic forks, the right and left legs of the WP cone valve forks serve completely different purposes. The right leg operates both compression and rebound viscous damping circuits, whereas the left leg contains two gas chambers. The left leg is referred to as the “sprung” leg, though contains no physical spring. Instead it makes use of compressed air to act as the spring. As a result there is no preload adjustment or recommended spring rates and front suspension stiffness is entirely dependent on the air pressures of the negative and positive chambers.

Damping is controlled by the right leg with a cartridge-type fork design whereby damping occurs in a chamber sealed off from the stanchion and outer fork leg. The cartridge threads into the top of the outer fork leg and a shaft, henceforth referred to as the damper rod, protrudes from the bottom of the cartridge housing and threads into the bottom of the fork foot, meaning any oscillation of the wheel results in an equal oscillation of the damper rod. Inside the cartridge, the other end of the damper rod is home to the cone valve assembly.

The fork cap/cartridge housing has a separate central assembly which is removed when the internal components of the cartridge must be accessed. This central assembly can be removed using a specific WP fork cap tool to reveal what is herein referred to as the base valve. After draining the oil the cone valve and damper rod assembly can then be removed from the cartridge housing.

3.1 Base Valve

The base valve is comprised of a spring, piston, compression and rebound shim stacks, and an internal floating piston (IFP) all affixed to the base valve rod and inner cap. The spring tension between the fork cap and base valve seal keeps the oil inside the cartridge under pressure thus preventing the cavitation of oil and air. 5 different spring rates are available to the base valve assembly, but as standard the system is fitted with a 25N/mm spring.

As the system is compressed and the damper rod enters the cartridge, oil travels up through the base valve piston and compression shim stack, into the space between the piston and IFP. The base valve spring then compresses to take up the volume of oil displaced by the damper rod. Low-speed compression damping can also be adjusted through the base valve in the form of a classic needle valve adjuster.



Figure 6: WP Base Valve

3.2 Cone Valve

The damper rod and cone valve assembly sits inside the cartridge between the base valve and the bottom of the cartridge. The lower end of the damper rod extrudes outside of the cartridge and threads into the bottom of the fork foot, hence allowing the rod assembly to reciprocate through its stroke inside the cartridge at the same rate as fork velocity. Below the piston sits an adapter sleeve housing a cone shaped mechanism, preloaded by a spring. The stiffer the spring, the more force is required for the cone to displace and allow oil through. WP offer 3 different angled cones at 11° , 15° and 18° , in addition to 10 different spring-rate options ranging from 25N/mm to 240N/mm giving a total of 30 different valve configurations. Oil flow is thus restricted by both the steepness of the cone and the stiffness of the spring.

A secondary cone adapter is also made available from WP with an additional oil bleed hole. The effect of this adapter will be investigated and analysed as part of the testing stage of this project, though the adapter without the bleed hole will be used for the majority of testing to avoid introducing too many variables into how damping is controlled.



Figure 7: Cartridge, Base Valve, and Cone Valve Assemblies

4. Damper Dynamometer Testing

4.1 Initial Experiment Design

Due to the vast number of variables within the design of the damping system it's necessary to develop a baseline method in which to test the desired combinations. Of most importance is the cone valve itself and thus tests are focused around the effects of this assembly.

Test	Cone °	Cone Spring Rate N/mm
1	18	100
2		25
3		240
4	15	100
5		25
6		240
7	11	100
8		25
9		240

Table 1: Initial Cone Valve experiment

The initial experiment design followed the process as outlined above in table 1. It was decided that each cone will be tested with the 100N/mm spring that comes installed as standard, as well as two other spring rates available: 25N/mm, the softest available option, and 240N/mm which is the stiffest cone valve spring rate available.

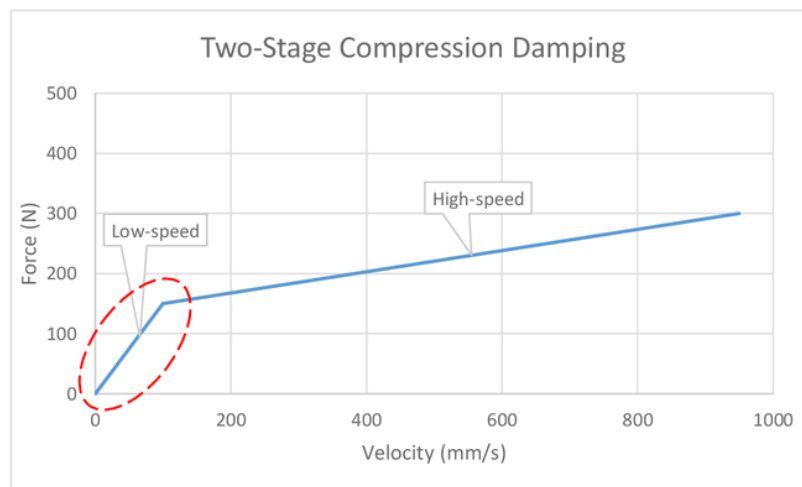
4.1.1 Predicted Outcomes

Before the experiment is conducted it is hypothesised that maximum compression and rebound damping would be achieved with the combination of 11° cone and 240N/mm spring. The 11° cone is the most acute of available valves and therefore offers the greatest restriction of oil flow. This combined with the stiffest spring and thus providing the most resistance, it is assumed that damping force would be of greatest magnitude.

With comparable logic it is therefore theorised that the least acute angled option, the 18° cone, combined with the softest spring of 25N/mm would provide the least resistance and therefore the least aggressive damping force.

4.2 Experiment Methodology

A damper dynamometer is extremely useful for testing suspension and allows the user to analyse the damping characteristics of any system. For racing applications it's important to understand how the system is valved to react to different dynamic scenarios. Most performance suspension systems designed around shim-stack damping can generally be described as having two-stage damping: a low-speed stage and a high-speed stage. The low-speed stage applies to relatively low suspension velocities such as upon acceleration or braking whereby the low flow rate of oil does not generate a force large enough to deflect the shim stack. Instead flow is choked through the needle valve damping circuit.



Graph 1: Two-Stage Compression Damping Example

From $F = cv$ damping force is proportional to velocity, therefore as the speed increases the force generated becomes large enough for the shims to flex and allow an increased volume of oil flow. As a result the damping force becomes less choked and develops a digressive characteristic after the shims open. This area after the knee in the graph is referred to as the high-speed damping stage whereby the system experiences high velocities due to harsh track conditions such as bumps and undulations.

4.2.1 Dynamometer Settings

Testing suspension systems is also important in gathering information regarding the range of adjustability of the system, thus enabling quicker decision-making when making track-side changes. For appropriate results the system should be tested at realistic real-world speeds and at a stroke similar to the available fork travel. Motocross forks can be subject to high suspension velocities due to landing off-road jumps or accelerating over ripples, however the mechanical limitations of the damper dyno restricts the maximum frequency to 3Hz at the greatest stroke of 100mm.

Thus the conditions remaining constant for the tests will be as follows:

Stroke	100mm
Test type	Multi-speed
Frequency	1-3 Hz
Steps	0.1 Hz
Oil	Silkolene 5W

Table 2: Experiment Constants

Multi-speed tests allow for a better resolution of cycle-averaged damping force graphs and thus allow for more precise post-processing. A single speed run would generate forces for just a single cycle of compression and rebound at the selected frequency, whereas a multi-speed test conducts multiple cycles increasing in frequency by 0.1Hz after each revolution of the dynamometer crank. This method enables the analysis of how damping characteristics change between cycles and the differences between low-speed cycles and high-speed in order to better understand the function of the cone valves.

4.2.2 Oil Effects

Oil viscosity is a major component of suspension damping and is therefore kept constant throughout the experiment. Viscosity is a property that determines a fluids resistance to shear stress and fluid friction (Elger, et al., 2013). As the temperature of the fluid increases its viscosity decreases and the oil can be known to break down causing a loss in damping. It was therefore decided to replace the oil for each test to prevent the chance of previous testing having an adverse effect on the damping of the subsequent test.

4.2.3 Test Method

For each combination of internal valving the test method must be consistent in order to later process and compare results. Manufactured performance suspension systems often have a recommended setting or operating range in which the system works best, however since this is not given by WP it is necessary to test the system throughout its whole range from the softest setting to the hardest.

For the compression adjuster, the softest setting is measured 30 clicks out from fully hard damping, whereas the rebound offers 34 clicks of adjustment. To avoid too much crossover in damping, the compression range will be tested with the least amount of rebound damping and starting at 30 clicks out. An additional 4 clicks of compression damping will be added after each damper dyno test. If the results of two successive tests show a large difference in damping characteristics the test will continue in intervals of 2 clicks.

After the compression range has been tested, the rebound range will be tested in the same manner with minimum compression damping.

5. Results Analysis

5.1 Initial Test Results

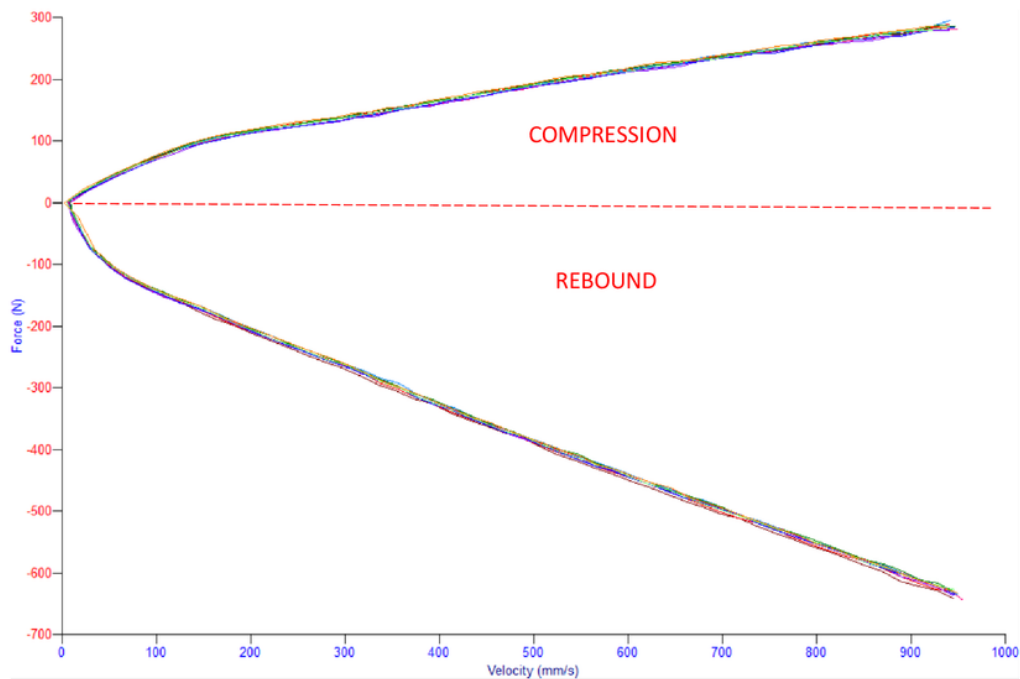
5.1.1 Pre-Test: 18° Cone, 100N/mm Spring, With Bleed Adapter

The first initial test was conducted with completely standard valving as sent from WP.

Cone °	18
Cone Spring Rate	100 N/mm
Base Spring Rate	25 N/mm
Adapter	Bleed hole

Table 3: Standard Valve Settings

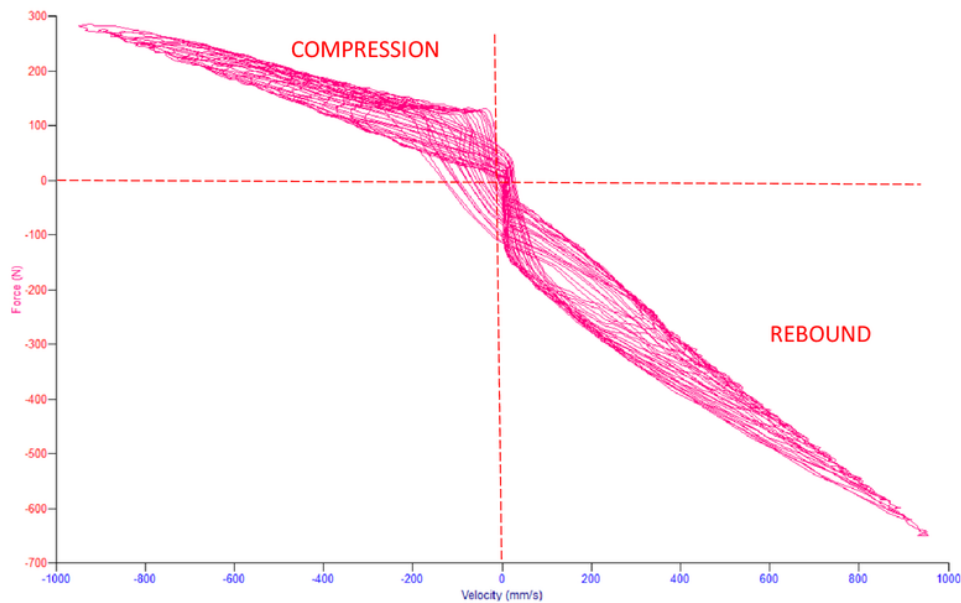
Suspension manufacturers tend to err on the side of caution and as standard usually opt for a softer set-up. The cone valve adapter used had a small bleed hole in the wall allowing for an additional passage for oil to escape through and thus reducing damping force. This however adds another variable into the equation and therefore is only used for the first initial test, after which it will be replaced with the secondary cone adapter without a bleed hole.



Graph 2: Compression adjustment range with 18-degree cone, 100N/mm spring, bleed adapter

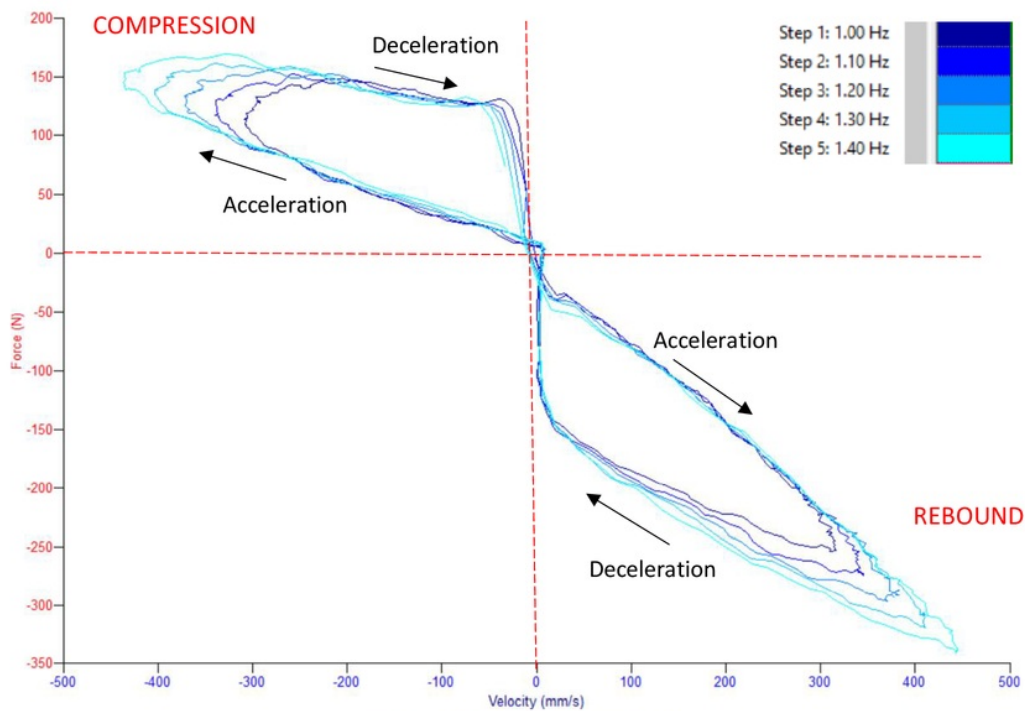
The results of the first test through compression adjustment range are shown on Graph 2. Values of force above zero correspond to damping force on the compression stroke whereas below zero represents rebound damping forces. The traces plotted on graph 2 are the average force values across the multi-speed run and are useful in simply determining the average effect of increasing or decreasing damping using the manual adjuster. The results show that there is very little change in damping characteristics as the compression adjuster approaches what should be maximum damping. This is unusual for a performance system however could be due to a blockage or fault within the internal mechanism.

As in most motorcycle suspension systems, the damping is rebound biased, meaning rebound damping forces are higher than compression damping forces. On the compression stroke the damping system must only control the oscillation of the unsprung mass, whereas rebound damping is responsible for controlling the much heavier sprung mass of bike and rider weight.



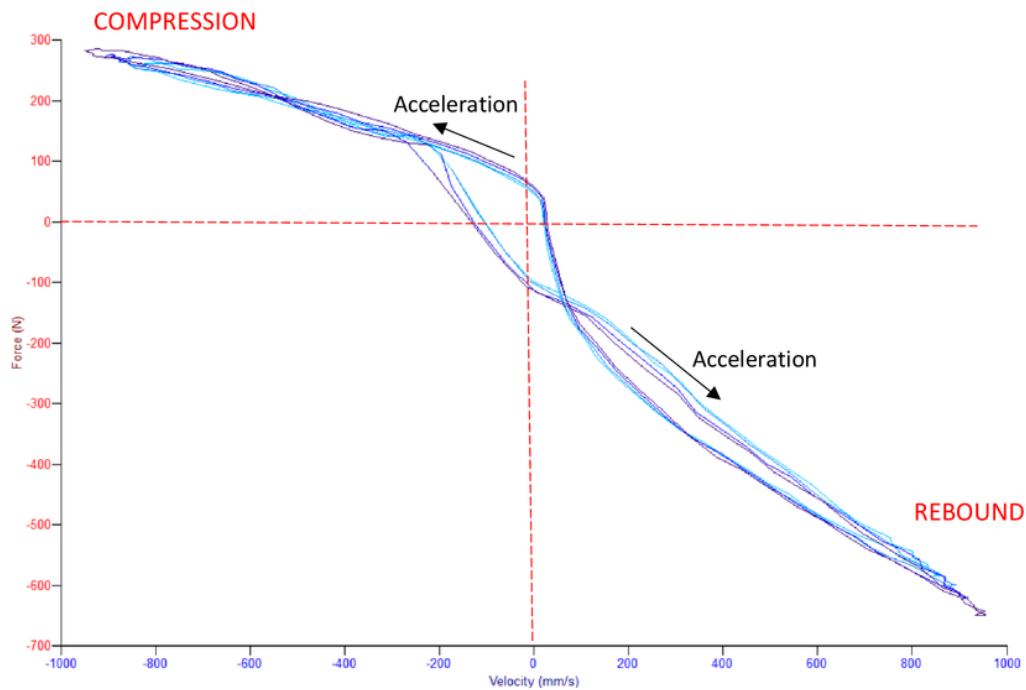
Graph 3: Complete Force vs Velocity Graph for Test 1

Graph 3 shows the results of each individual cycle from 1-3Hz plotted together. The upper left quadrant represents the entire compression stroke, whereby the peak force at peak velocity occurs midway through the stroke. The origin then represents the change of direction as the system begins the rebound stroke as shown in the lower right quadrant. This type of graph is extremely useful as it can be seen how the damping changes from low-speed oscillations to high-speed. It also shows damping characteristics on the initial acceleration of compression or rebound as well as the deceleration towards the end of the stroke.



Understanding how the damper dyno plots force against velocity is vital for accurate analysis of the two strokes. As shown on Graph 4, the load cell begins measuring force at the start of the rebound stroke as it begins to accelerate. Maximum force is recorded at maximum velocity, around midway through the rebound stroke, after which it decelerates towards full extension where velocity is zero and subsequently changes direction and begins acceleration of the compression stroke. After reaching maximum velocity the system again decelerates until the fork is completely compressed. The dyno then increases the next cycle's frequency by 0.1Hz and the process is repeated.

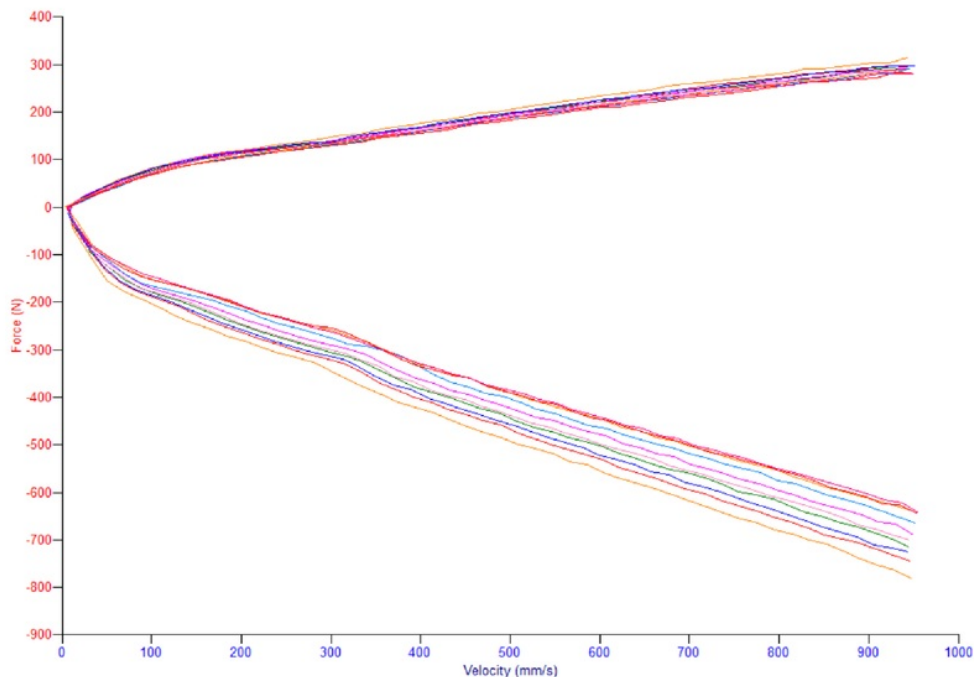
It can be seen on graph 4 that at low-frequency inputs both rebound and compression damping forces on initial acceleration are lower than their respective deceleration damping forces. Contrary to this, high frequency inputs as seen on graph 5 show very different damping characteristics. Rebound acceleration remains slightly lower but the hysteresis between this and deceleration is reduced. The compression stroke however shows higher rates of initial acceleration with little loss in damping on deceleration until nearing full compression.



Graph 5: Test 1 High Frequency cycles: 2.7-3Hz

The number of auxiliary variables that affect damping make it impossible to confidently make an assumption on the cause of the hysteresis between frequency cycles. Not only does the cone and spring rate influence damping, but the bleed hole in the adapter reduces damping force by providing an extra orifice for oil flow, in addition to the base valve compression and rebound shim stacks. This combined with low-speed compression and rebound creates too many damping circuits and thus future tests will eliminate certain variables.

5.2 Influence of Low-Speed Adjustors



Graph 6: Test 1 Rebound Adjustment range with 18-degree cone, 100N/mm spring, bleed adapter

The above graph 6 shows the rebound adjustment range from the first test, whereby damping force is increased as the low-speed adjuster is wound in, restricting flow through the circuit. Compared to the results from adjusting low-speed compression, the rebound adjuster seems to work as expected. The gradient of mid to high speed damping remains alike with parallel characteristics but is offset by the increased damping at low-speed. A slight increase in compression damping is also measured by use of the rebound adjuster, more than what can be seen using the compression adjuster showing evidence of some crossover effects in damping.

Disassembly of the fork and damping circuits revealed the base valve low-speed compression adjuster to be in mechanical working order, whereby as the external adjuster is turned the needle moves in tandem. This suggests that the lack of compression adjustment may be either deliberately designed by WP or a flaw in the overall system.

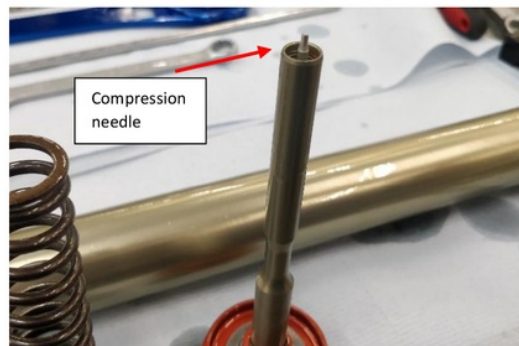
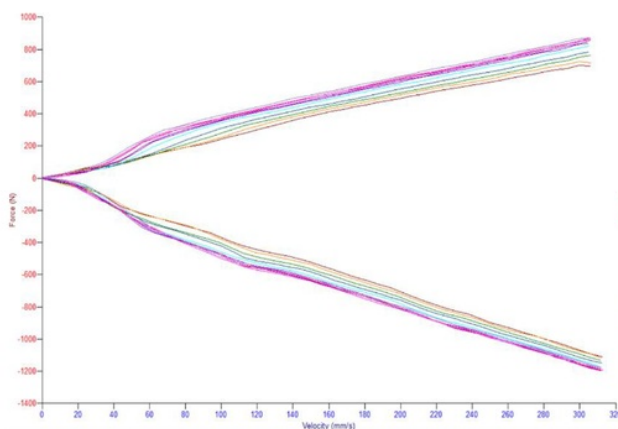


Figure 8: Base Valve Disassembly

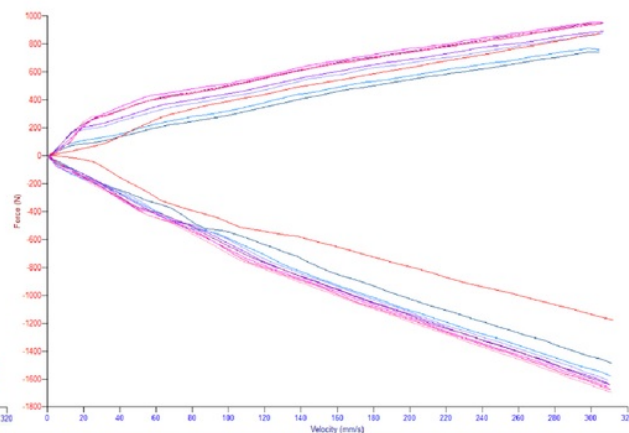
5.2.1 Crossover Effects

From experience, it is not uncommon in some modern performance systems for there to be little to no range of compression damping using the low-speed adjuster. Instead some systems have crossover effects such that the rebound damping adjuster has more of an effect on the compression damping. This isn't a desirable design; if road conditions or rider feedback imply a change in rebound damping is required, the additional influence on compression damping may be unwelcome and cause further handling problems.

Shown below is a side by side comparison of an Öhlins TTX-GP shock absorber for a Yamaha YZF-R6. Graph 7 on the left shows the measured damping from the compression adjuster, whereas on the right graph 8 is the results from the rebound adjuster. The red plot on graph 8 is a reference to the maximum compression damping force of graph 7. There are clear crossover effects for both damping circuits, drastically that of the effect on compression. Low-speed compression damping properties are radically changed by use of the rebound adjuster. These two graphs show the importance of testing suspension systems on a dynamometer. When all information is provided, decisions can be made in the field knowing the full extent and effect that the change will have.



Graph 7: Öhlins TTX-GP Shock- compression adjustment range

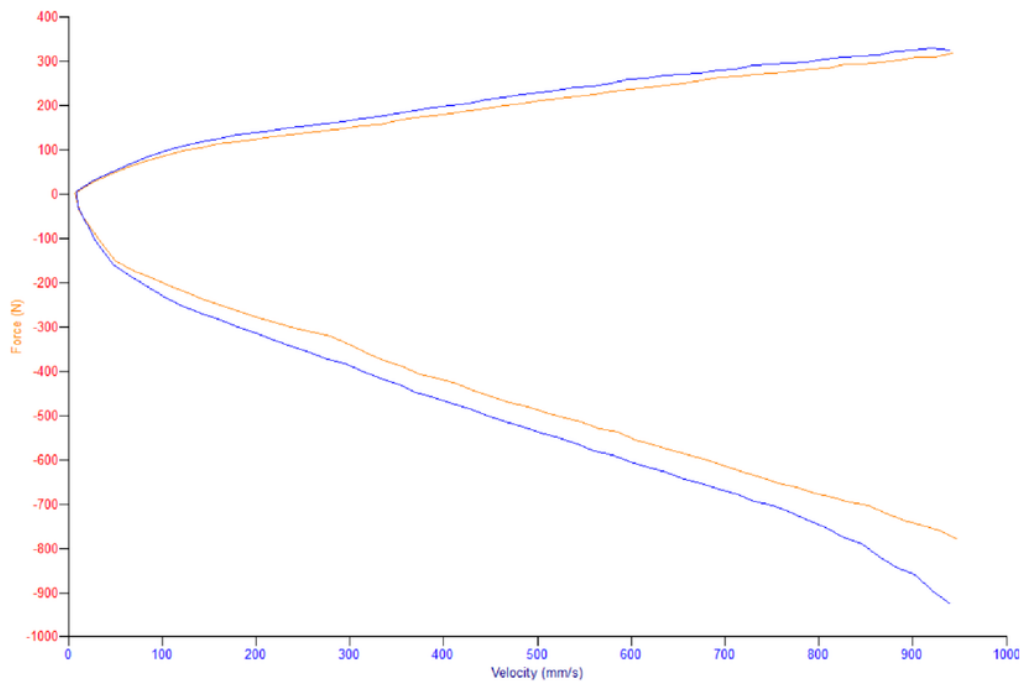


Graph 8: Öhlins TTX-GP Shock- rebound adjustment range

5.3 Influence of Auxiliary Variables

The influence of secondary variables on the damping requires isolation of the cone valve circuit. Hence the bleed adapter is swapped for the normal adapter without a bleed hole provided in the WP tuning kit. The impact of the base valve circuit will also be investigated throughout testing.

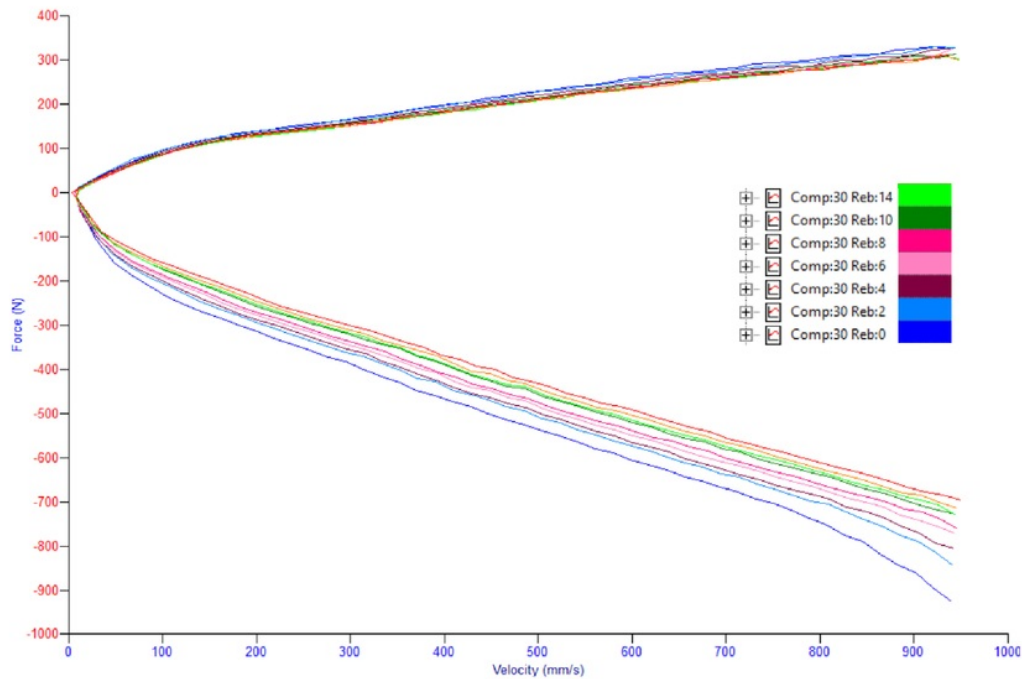
5.3.1 Bleed Adapter



Graph 9: Comparison of maximum damping with two different cone adapters

Graph 9 above shows a comparison of maximum damping forces after replacing the cone adapter. Compression damping is marginally increased, though the most notable difference is that of high speed rebound damping. As the velocity surpasses approximately 750 mm/s the force progressively increases. Both measurements were tested with the low-speed rebound adjuster all the way in, thus the bleed adapter evidently provides a type of blow-off relief to prevent such a rate of increase in damping at high velocities.

As shown on graph 10 the damping only shows this dramatic increase in high-speed damping when the adjuster is approaching its maximum point. At this setting the needle is most likely almost completely blocking the flow of oil through the low-speed orifice which results in such a difference in damping between this setting and 2 clicks out, represented by the dark blue and light blue traces respectively. The bleed adapter therefore diminishes this affect by allowing a passage of oil flow through the cone assembly.



5.3.2 Base Valve

The base valve is another variable aiding in both compression and rebound properties due to the shim stacks either side of the bored piston. However the rebound shim stack is unlike typical systems in that it contains a unique laser cut shim as shown in figure 10. This allows some amount of oil flow through the piston in both directions and it is assumed may not have the largest effect on damping compared to if it were a standard shim.



Figure 10: Base Valve Assembly

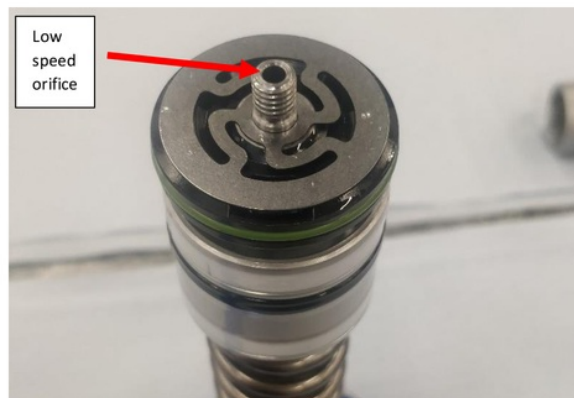
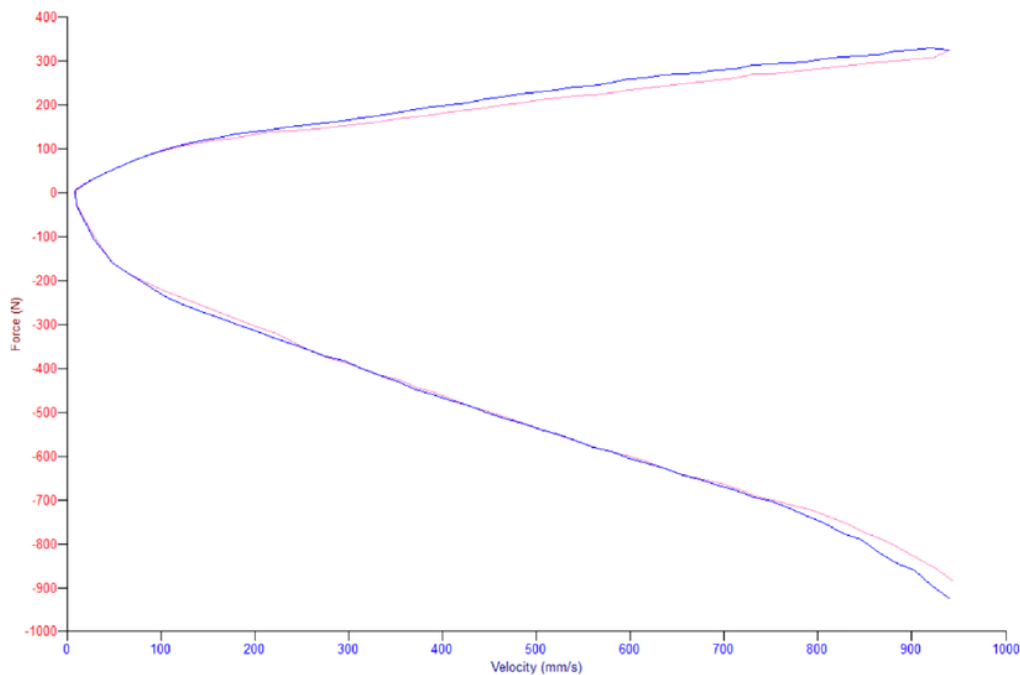


Figure 9: Unique Rebound Shim

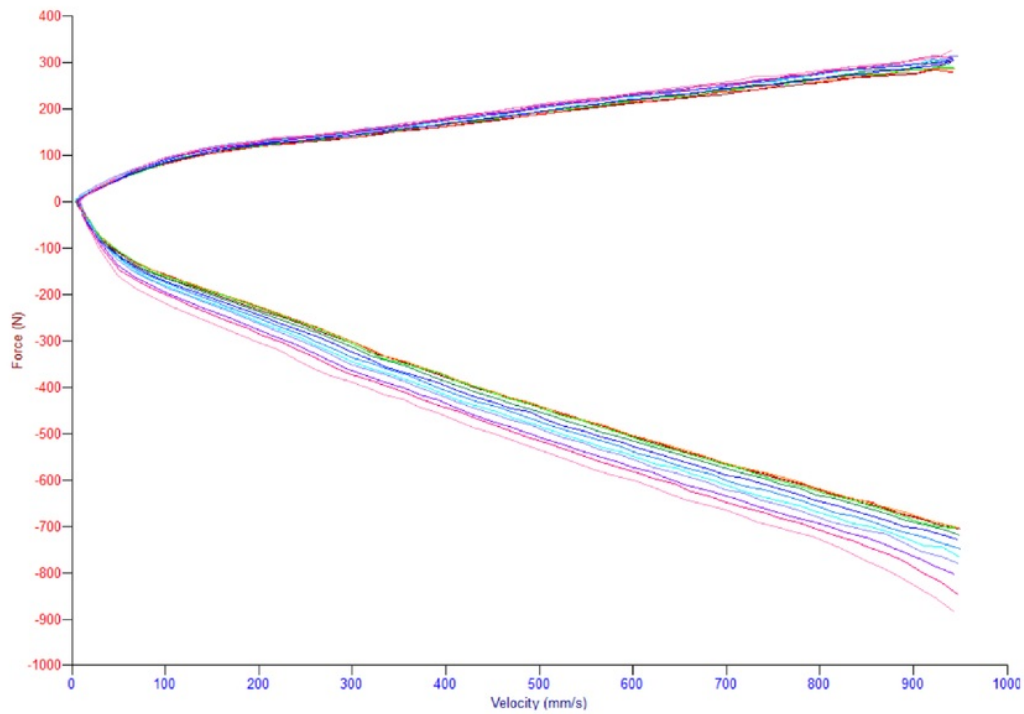
The compression shim stack on the opposite side is similar to a standard tapered shim stack and as the damper rod moves through the cartridge on the compression stroke, the silver floating piston compresses the spring to take up the volume of oil displaced by the damper rod. The upward acceleration of the cone valve assembly forces oil up through the base valve piston and low-speed circuit. It is therefore expected that the removal of this piston would decrease both compression and rebound damping.

The results of removing the base valve piston are shown on graph 11 below. As predicted the mid to high speed compression damping is slightly lower but not considerably. Since the low-speed compression circuit seems to actively make no difference, this remains unchanged. The rebound damping is only fractionally lower at high velocities where the force ramps up, below this the base valve piston has little effect on the rebound damping. Presumably the slotted shim is responsible for the lack of change in the rebound damping as it doesn't provide significant resistance against oil flow.



Graph 11: Comparison of maximum damping with and without base valve piston

Although the above results suggest the base valve piston plays an insignificant role in altering damping properties, its use will be tested along with the various combinations of cone valve and spring rate before making a conclusion on its influence. The complete range of rebound damping adjustment without the base valve can be seen on graph 12.



Graph 12: Range of rebound damping for 18 cone, 100 N/mm spring, without base valve

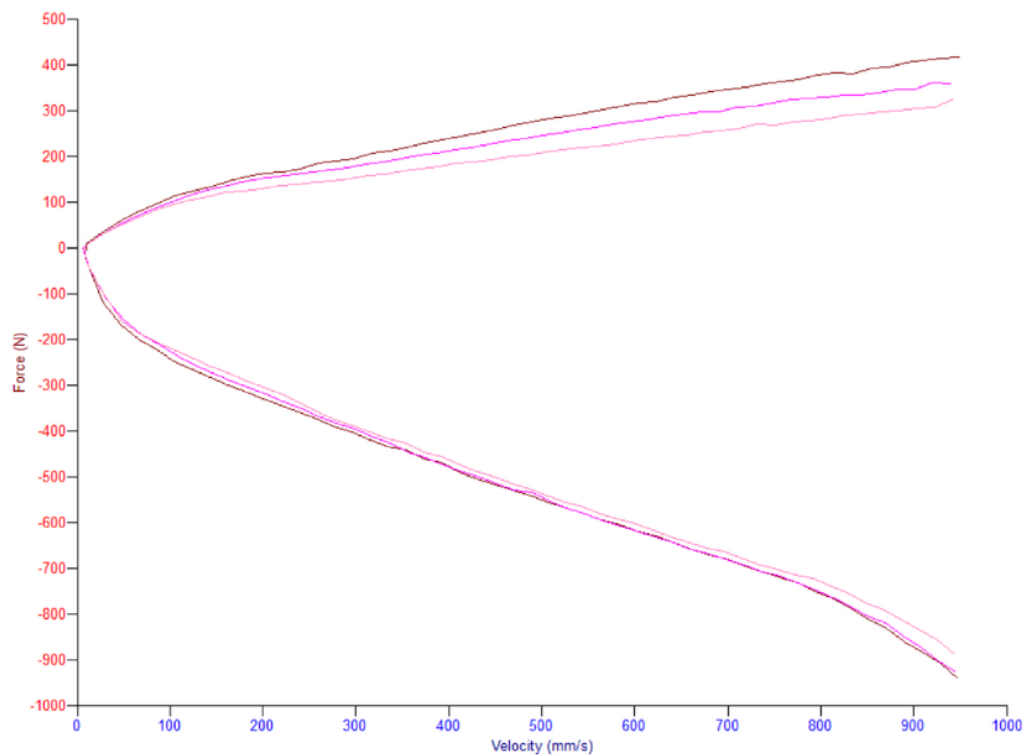
5.4 Influence of Cone Angle

The effect that the three different types of cone have on the system and the subsequent post-processing is one of the principal focuses of this study. The shallowest of cones has a slope angle of 11° from the vertical, followed then by the 15° and the 18° cones. As hypothesised in section 4.1.1 it is predicted that the 11° cone will offer the most resistance to oil flow and therefore highest magnitudes of damping, yet how this translates into the transmissibility of road inputs will be further explored in section 6.

Test	Cone spring rate N/mm	Cone $^\circ$	Graph colour
1	100	11	Dark Red
4		15	Magenta
7		18	Pink

Table 4: Cone Analysis Tests

Referring back to table 1 in section 4.1, table 4 represents the respective results under analysis in this section, whereby spring rate is kept constant and the cone is the singular variable.



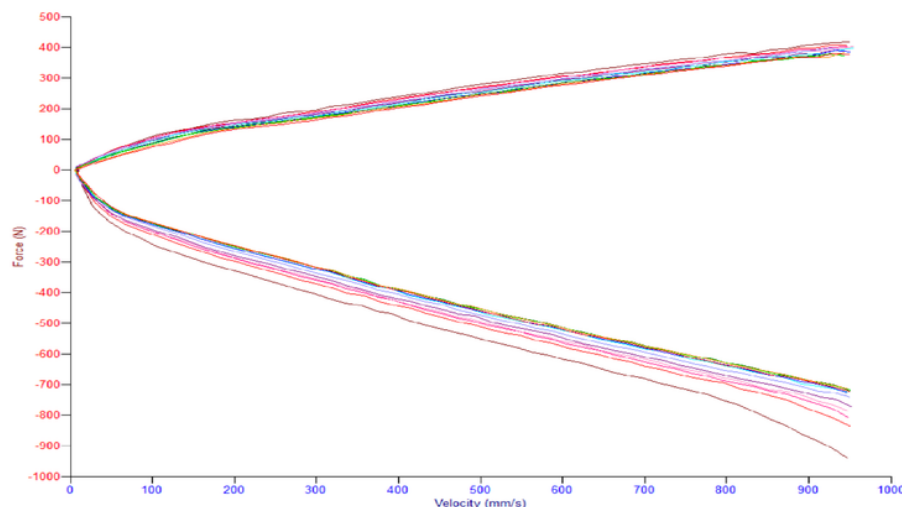
Graph 13: Comparison of maximum damping settings with three cone valve

Graph 13 shows a comparison of the measured damping forces at the maximum rebound damping setting for each cone. Since the cone valve is primarily to control compression damping, the changes in rebound damping are very minimal. However the 18° cone clearly exhibits rebound damping forces less than that of the 15° and 11° cones. This could potentially be due to the lack of installed preload of the spring against the cone valve allowing for the passage of oil up through the adapter on the rebound stroke. Approximately 0.5-1mm of free-play exists between the spring and cone assembly and the cone adapter which could be enough to allow oil flow and thus decrease damping. It's more noticeable on the 18° cone due to the shallower gradient allowing a larger volume of oil flow. The 15° and 11° cones show very similar rebound damping properties, the only difference being slightly decreased low-speed damping with the 15° cone.

Cone	Peak Compression Force (N)	Peak Rebound Force (N)
11	418.28	955.45
15	359.95	942.67
18	325.57	904.7

Table 5: Maximum compression and rebound forces for cone analysis

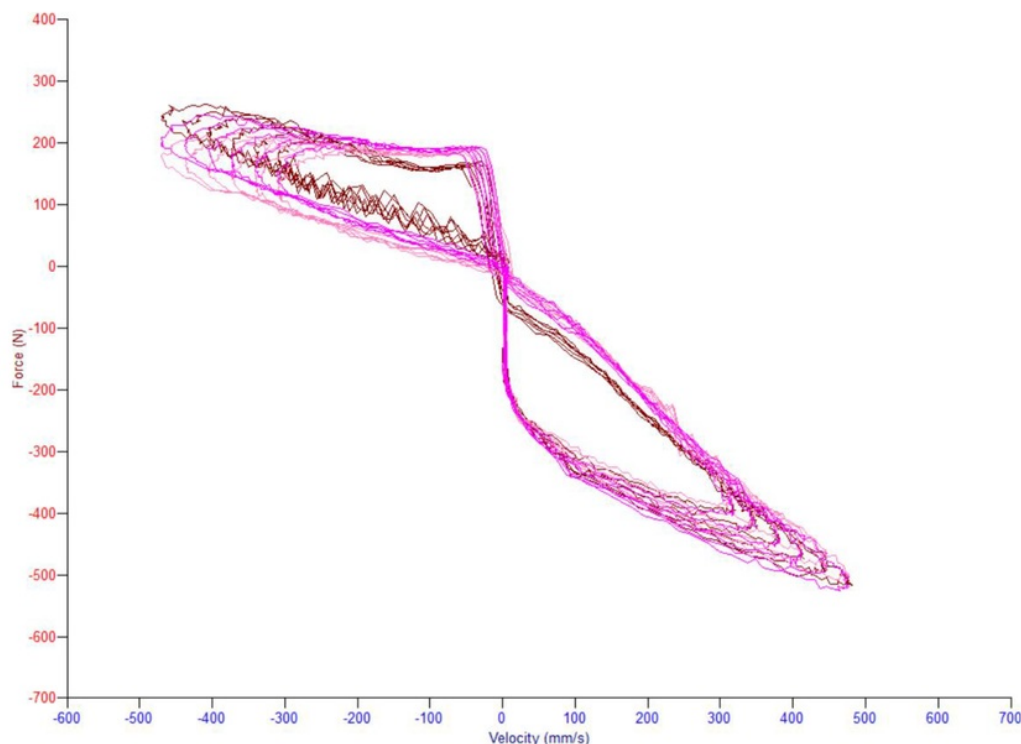
Compression damping forces are the most noticeable changes caused by changing cone, showing differences across low-speed and high-speed velocities. Unlike increasing the external rebound adjuster whereby the mid to high speed rebound damping profiles remain parallel but are offset from each other, changing cone alters the gradient of the mid-high speed compression damping profile. The 15° cone offers a 9.5% increase in compression damping force compared to the 18° cone which comes as standard. The 11° cone offers 22% more resistance than standard.



Graph 14: Rebound adjustment range for 11° cone, 100 N/mm Spring, without base valve

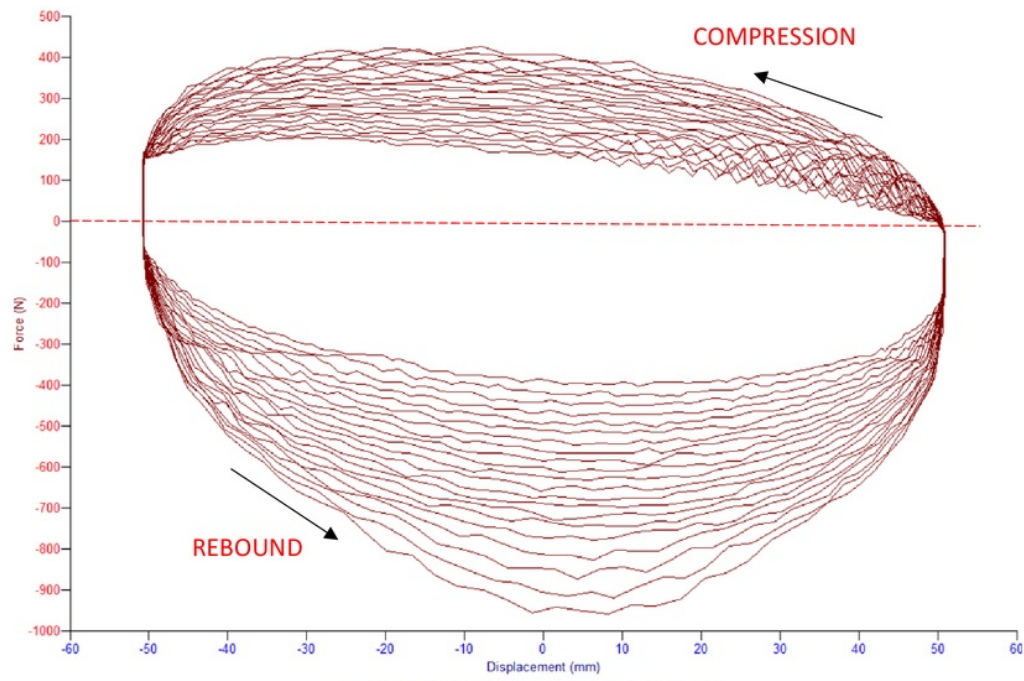
As a collective range it can be seen on the following graph 14 that the 11° cone displays similar characteristics to that of the 18° cone seen on graph 12, whereby damping profiles are gradually offset as the adjuster is wound in. The rebound offset between the two maximum traces is comparably larger than is seen with the 18° cone, however this will largely be due to the effect of the needle.

Interestingly, when isolating the low frequency cycles of each cone at maximum rebound settings, the 11° valve exhibits anomalous characteristics. Shown on graph 15, on initial acceleration of the compression circuit there are noticeable fluctuations in damping force that are not seen on the two other cone valves. Further examination of the results from preceding rebound settings revealed this to be a recurring characteristic for the 11° cone throughout the range of adjustment.



Graph 15: Low frequency cycles of the 11°, 15° and 18° cones with 100 N/mm spring, without base valve

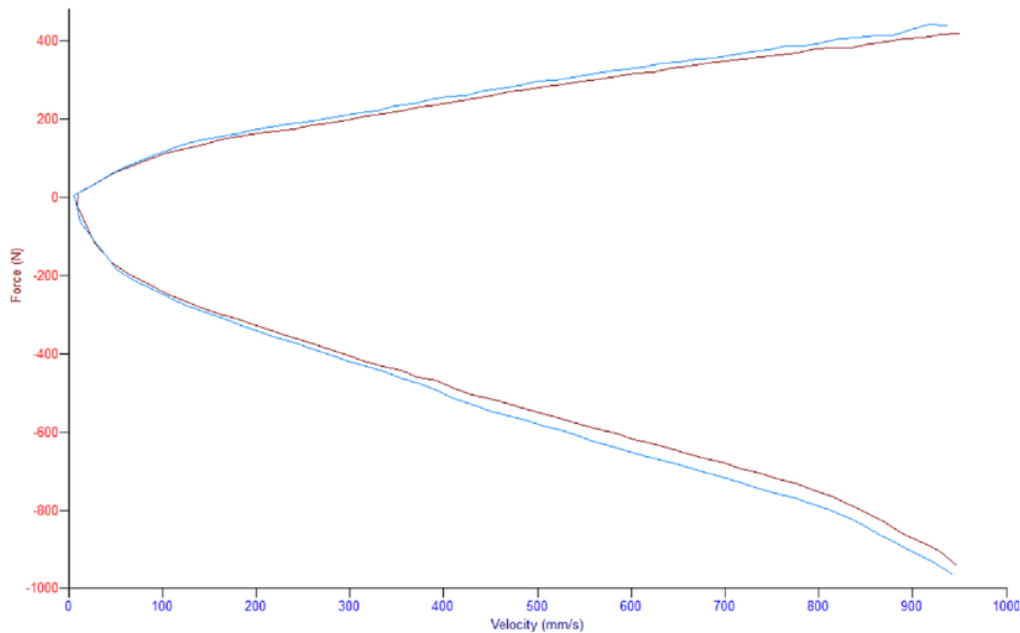
It is supposed that the low velocities and turbulence of oil flow combined with the steep gradient of the cone cause a “fluttering” effect of the cone against the adapter due to the free-play in the system, which can also be seen in the force-displacement graph 16. It appears that the fluttering tends to diminish as the damping force surpasses approximately 100N, which happens to coincide with the 100N/mm spring rate of the cone spring. As the cone valve begins to open damping becomes more stable and continues to increase as the system approaches maximum velocity.



Graph 16: Force-Displacement graph of 11 cone

5.4.1 Base Valve Effects

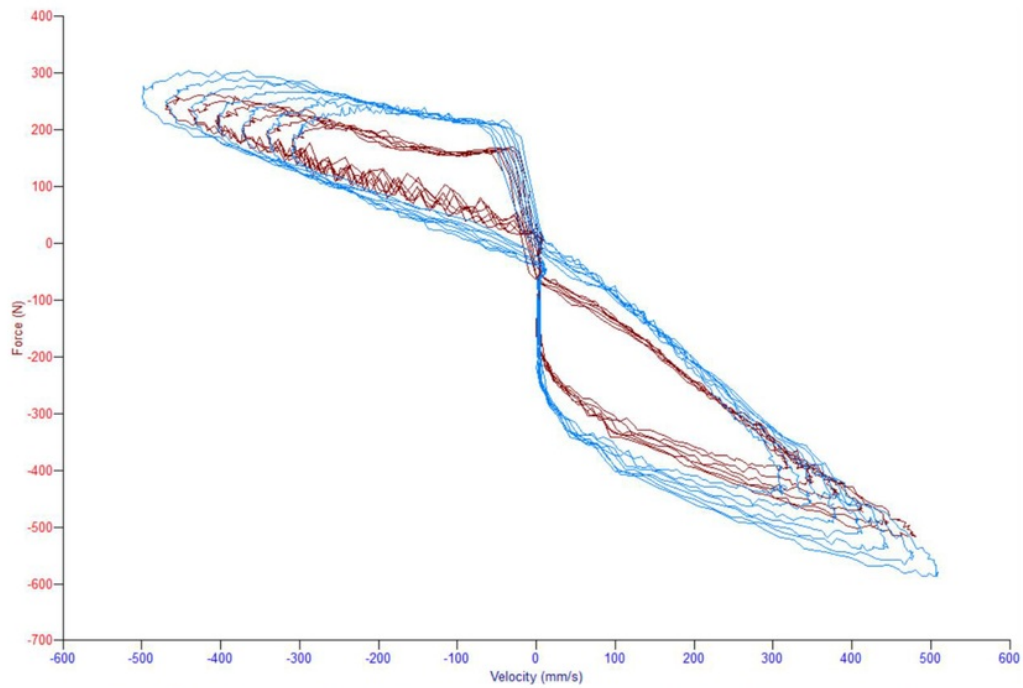
Since the damping characteristics of the 11° cone were irregular compared to the other two cones, it was hence decided to test the valve with the base valve piston back in the assembly. It was noted that with the 18° and 15° cones the base valve made minimal difference to damping properties, and looking at the average force-velocity graph for the 11° cone with and without base valve the same might be said.



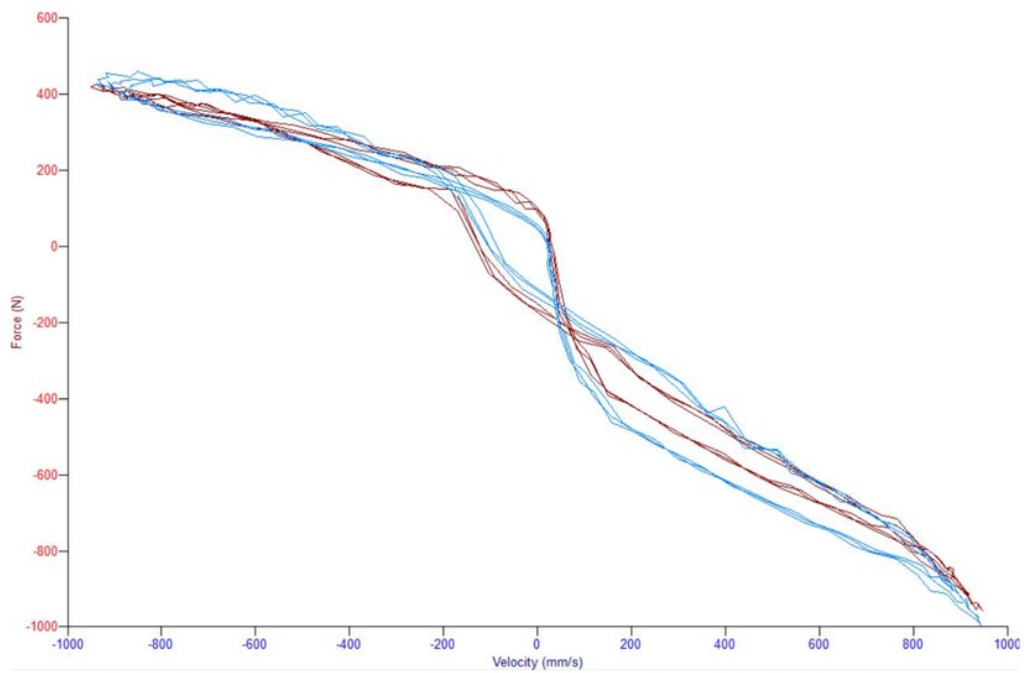
Graph 17: Force-velocity graph for 11° cone with and without base valve

However a closer look at the low frequency cycles shows a drastic change in damping properties as can be seen on graph 18. The previous abnormal behaviour of the fluctuation upon initial compression acceleration has been resolved and instead increases linearly as has become expected. Damping on the deceleration has also increased and thus the base valve has a more important role than initially assumed. For the 11° cone the base valve provides an additional restricted circuit for oil flow and the extra damping force it provides at low speed is enough to provide a stable response from the cone valve.

The difference in high frequency damping with the addition of the base valve is shown on graph 19 to mainly increase the deceleration damping phases. Other characteristics are kept consistent and thus it can be summarised that the base valve is efficient at providing both low-speed and high-speed damping support.



Graph 19: Low frequency cycles for 11 cone with base valve (blue) and without base valve (burgundy)



Graph 18: High frequency cycles for 11 cone with base valve (blue) and without base valve (burgundy)

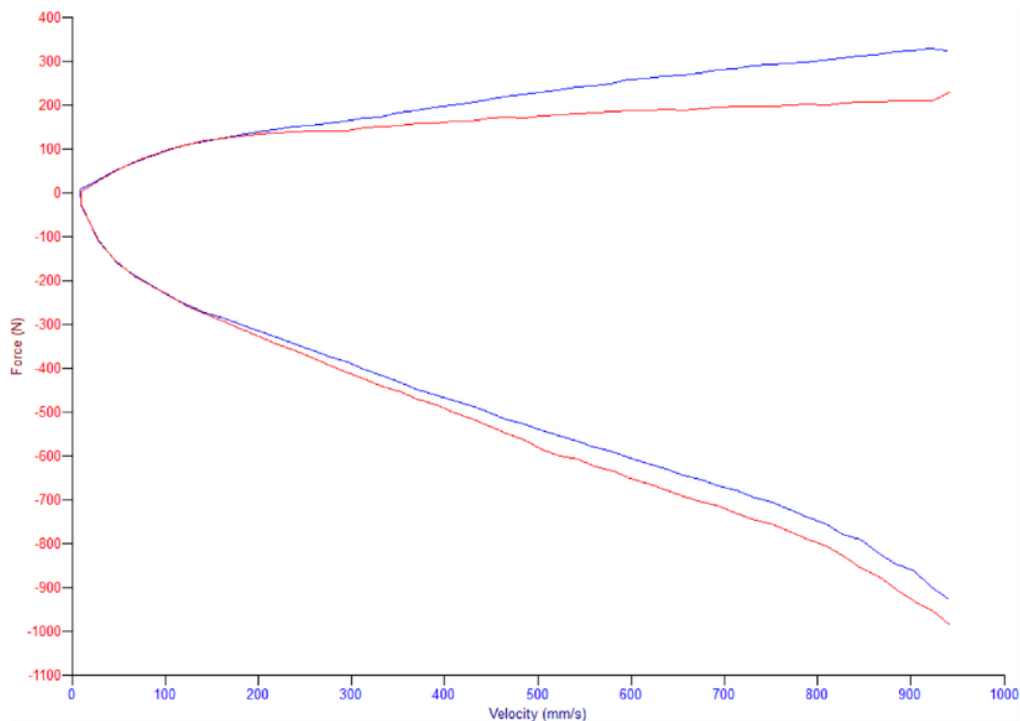
5.5 Influence of Spring Rate

Cone spring rate is the second variable under investigation and was expected to have more of an influence on damping than the cone angle. Out of the 10 available springs in the WP tuning kit, 3 were tested: the 100 N/mm which comes as standard, the softest available spring at 25 N/mm and the stiffest option at 240 N/mm.

Cone	Spring Rate N/mm	Peak Compression	Peak Rebound	Graph Colour
18	25	230	1000.06	Red
	100	327.66	942.76	Blue
	240	524.16	1018.23	Orange

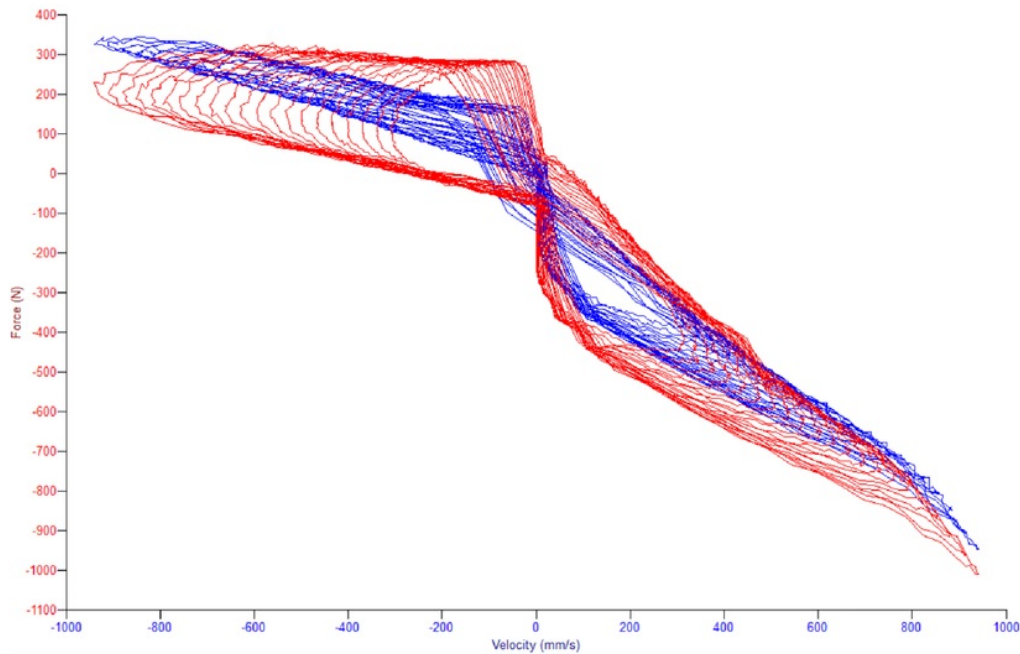
Table 6: Peak forces for 18° cone with differing spring rates and corresponding graph colour

Looking initially at the results of changing spring rate for the 18° cone, graph 20 shows the results of the 100 N/mm spring and 25 N/mm spring. Low speed compression damping is almost exactly the same, although the softer spring exhibits slightly more digressive properties than the harder option as velocity increases. The gradient of mid to high velocity damping characteristics is almost flat after the initial changeover for the softer spring with a maximum compression force measured at 230 N.



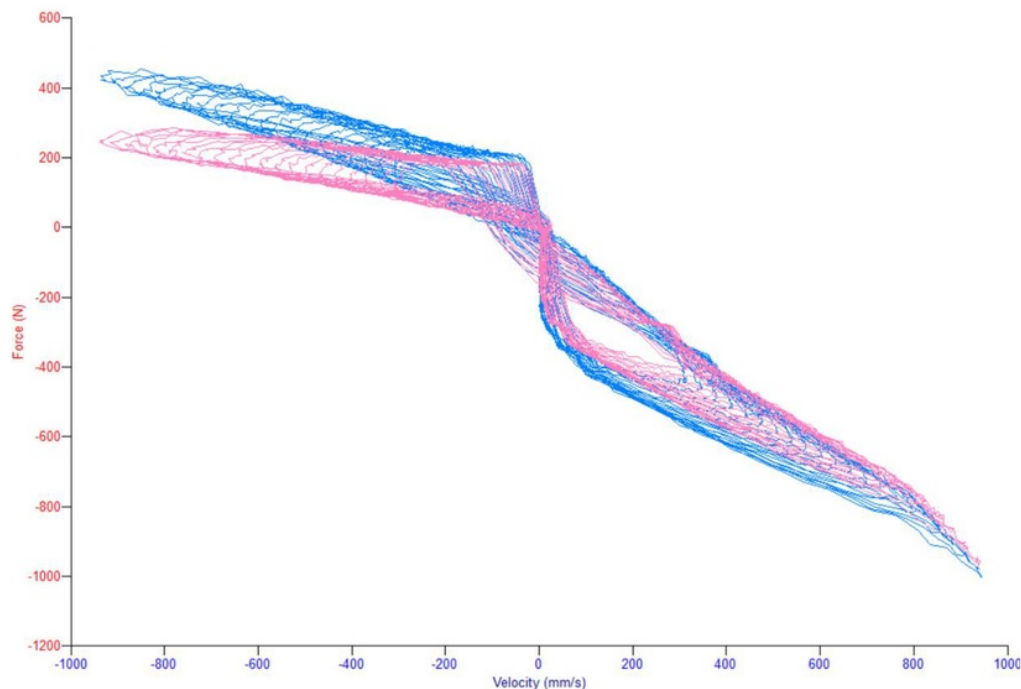
Graph 20: Average force-velocity graph for 18 cone with 100 N/mm and 25 N/mm springs

Unexpectedly however the high speed rebound damping is actually increased with the softer spring. Similar to compression, low speed characteristics are pretty equal with either spring, though at approximately the same velocity that compression changes over to a less steep gradient the rebound damping starts to increase.



Graph 21: Full force-velocity graph for 18° cone with 100 N/mm and 25 N/mm springs

Comparing the force-velocity graphs for both complete runs it can be seen that the damping characteristics are actually extremely different. Where it was previously thought that low speed properties are almost equal it is clear from graph 21 that this is not the case. The hysteresis between compression damping acceleration and deceleration is minimal with the 100 N/mm spring and damping characteristics, particularly at high velocities, are somewhat similar. However the hysteresis with the 25 N/mm spring is exaggerated, whereby damping force drastically increases as the system surpasses peak velocity, such that compression damping on the deceleration is actually higher with the softer spring. This does not correlate with what logic would suggest the effect of a softer spring would be and therefore prompted further investigation into the anomalous result.

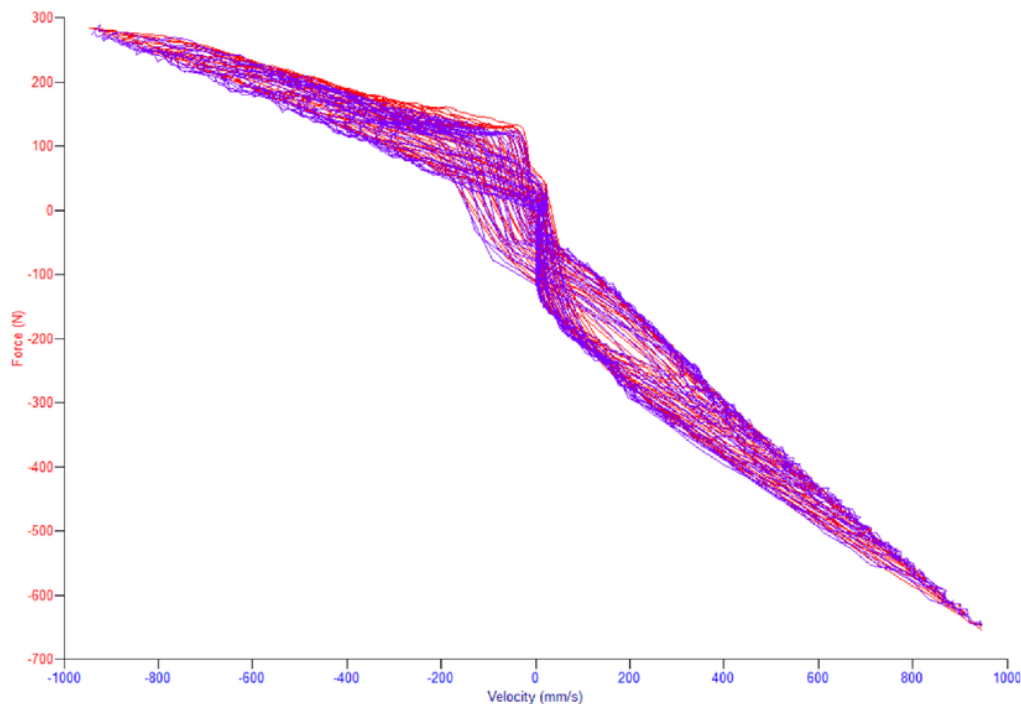


Graph 22: Full frequency Force-velocity graphs for 11° cone with 100 N/mm (blue) and 25 N/mm (pink) springs

Graph 22 shows the effect of changing spring rate when combined with the 11° cone. The hysteresis characteristics recorded with the 18° cone cannot be seen and as such imply that the anomaly is caused by something else.

How low-speed adjusters influence damping was previously discussed in section 5.2 and it was concluded that the compression adjuster made no noticeable difference to average damping properties. While this remains true, its impact on damping as a whole may not be as insignificant as first thought. The changes in damping in graph 21 can neither be ignored, nor explained by the logic of changing the cone or spring. It is also unlikely that a rebuild error caused such differences and hence triggers further investigation into the compression adjuster.

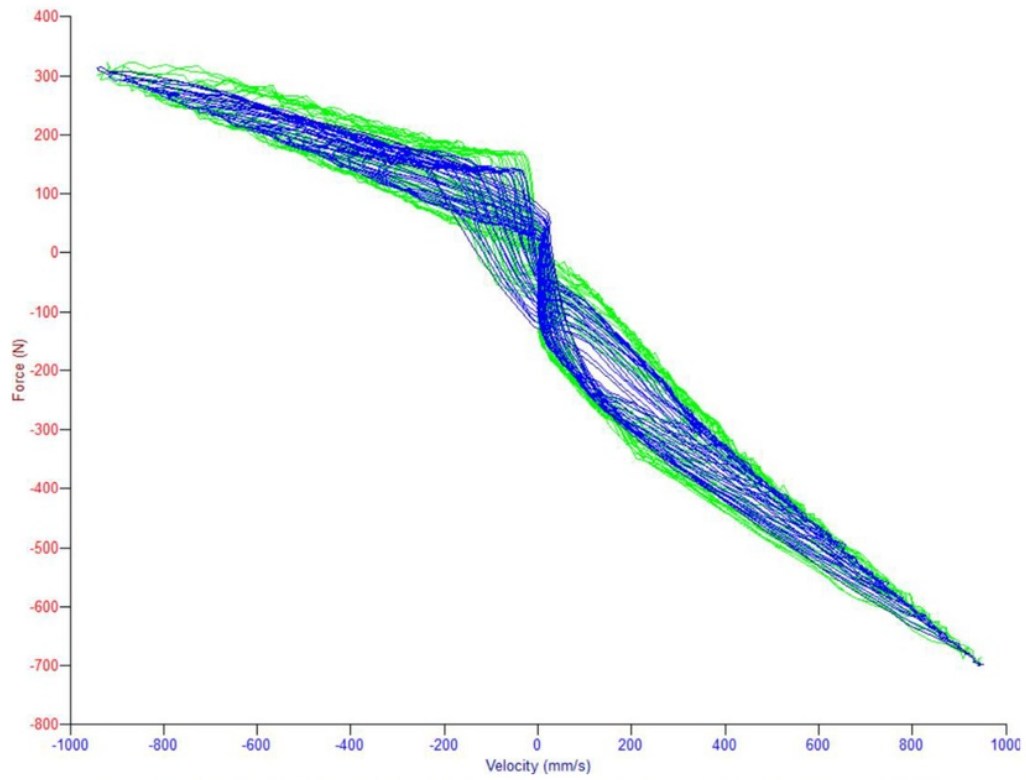
When the initial tests were conducted the range of both compression and rebound were tested before it was decided that since the compression adjuster had no effect there was no use in testing its range, and following tests only measured the effects of rebound adjustment. However the data retrieved from the first tests can be used to analyse whether something was missed in the full frequency force-velocity graphs.



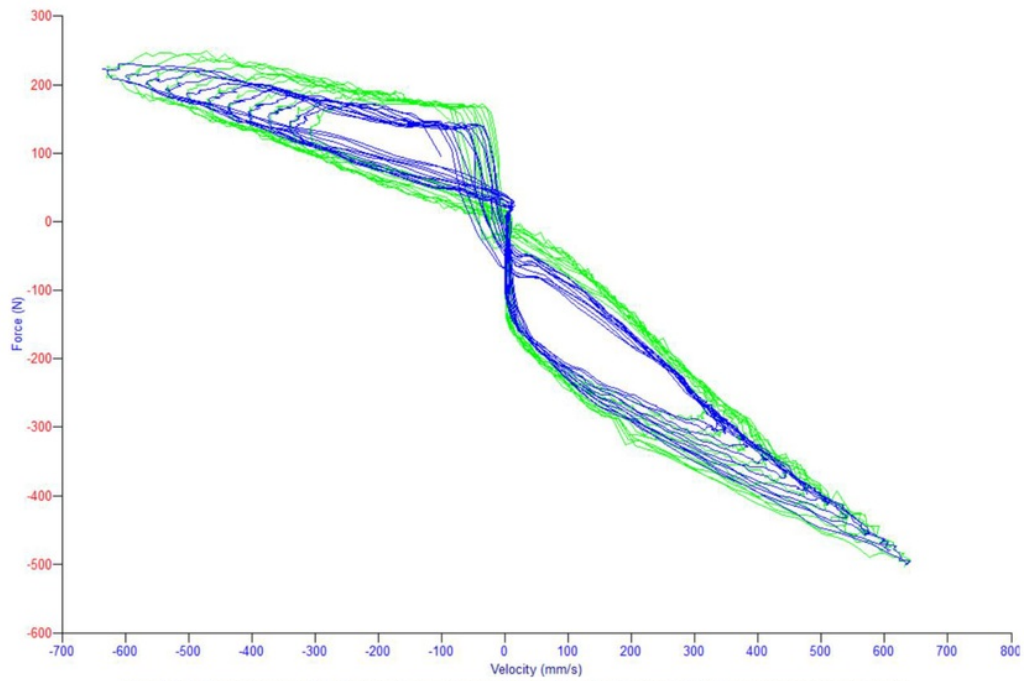
Graph 23: Full frequency runs for 18° cone, 100 spring, bleed adapter, min and max compression settings

Going back to the first test conducted with the bleed adapter, graph 23 shows no change in damping characteristics between the compression adjuster at its minimum and maximum settings, therefore causing the average graphs to show no changes. When the adapter was replaced for the option without a bleed hole, subsequent testing again showed no change in average damping and it was therefore determined that the adjuster had very minimal effect. What was overlooked was the effect the new adapter had on the full frequency graph.

Graph 24 shows the difference in damping across the full frequency range when the adjuster is at maximum and minimum settings, green and blue traces respectively, and explains why nothing can be seen on the average graph. Isolating the lower frequency runs shown on graph 25 shows a clearer view of how the damping is behaving at low speeds. Winding in the compression adjuster to its maximum point actually decreases acceleration damping; seemingly it forces more oil to pass through the cone and thus force is minimised at the beginning of the compression stroke. Upon deceleration however, the cone valve closing and seating against the adapter restricts the oil flow, in addition to oil being choked through the needle valve circuit creates the increase in damping as the system's velocity is decreasing.

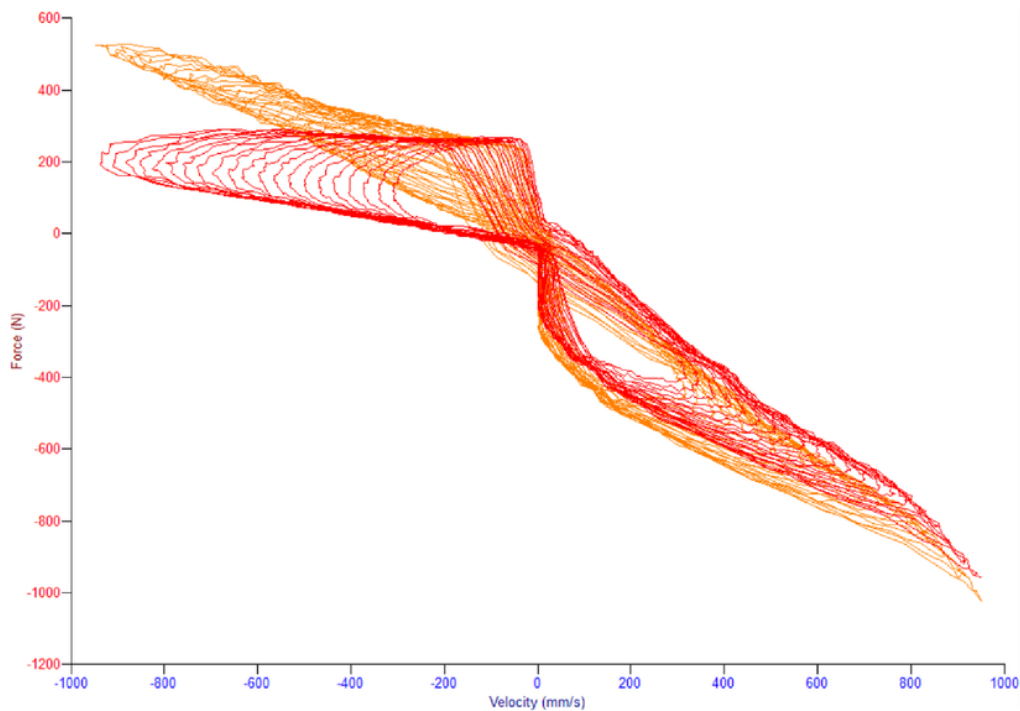


Graph 25: Full frequency force-velocity graph for maximum (green) and minimum (blue) compression settings



Graph 24: Frequencies 1-1.8Hz with maximum (green) and minimum (blue) compression settings

The characteristics presented by the previous two graphs correlate with the anomalous behaviours seen in the comparison of the 100 N/mm and 25 N/mm springs. Consequently it can be concluded that such behaviours with softer spring do not come as a result of cone angle or spring rate, but due to the low-speed compression adjuster: a variable previously assumed to be negligible. During rebuild phases between tests the compression adjuster must have mistakenly been altered, a change which thought bore no consequence to damping characteristics. Though this revelation poses some complications for the analysis of certain graphs, the more that is understood about how the system interacts with its collective circuits the better conclusions can be drawn.

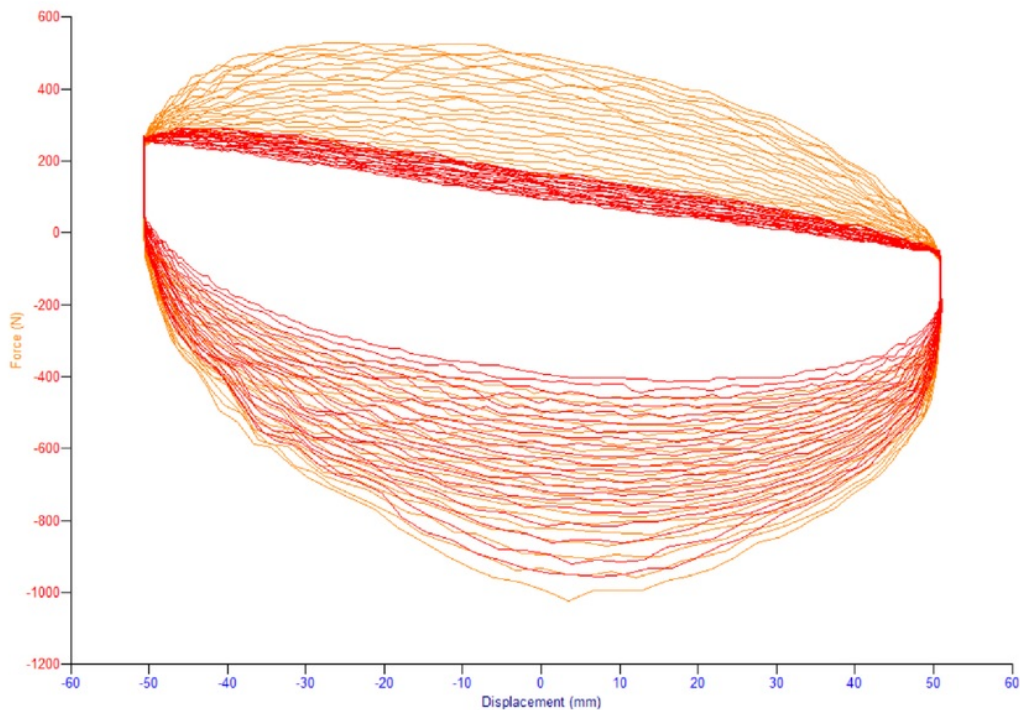


Graph 26: Full frequency force-velocity graphs for 18° cone with 25 N/mm spring (red) and 240 N/mm spring (orange)

It is unknown at what point and by how much the compression adjuster was changed, however the tests compared on graph 26 were conducted in sequence and looking at the traces it can be safely assumed that compression settings were equal.

The 18° cone with stiffer spring thus has an effect whereby the gradients of both acceleration and deceleration compression damping are drastically increased compared with the 25N/mm spring and therefore offers more high speed damping. It was noted by Neal that digressive properties are preferred in damping systems whereas the 100N/mm spring exhibits more linear characteristics. The effect of how these characteristics translate to the overall transmissibility of the system will be further explored in section 6.

Force as a function of fork displacement can be seen on graph 27. The differences in acceleration and deceleration damping can be seen at either end of the major-axis whereby the particularly skewed red trace shows that maximum compression damping occurs just before the change of direction onto the rebound stroke. Notably there's marginal differences in compression damping for low-speed and high-speed cycles with the softer cone valve spring, unlike that of the harder spring whereby as expected as the frequency increases as does the damping force. The fact that the 25 N/mm requires such little force for it to open a significant amount most likely explains why there is such little change in damping characteristics.

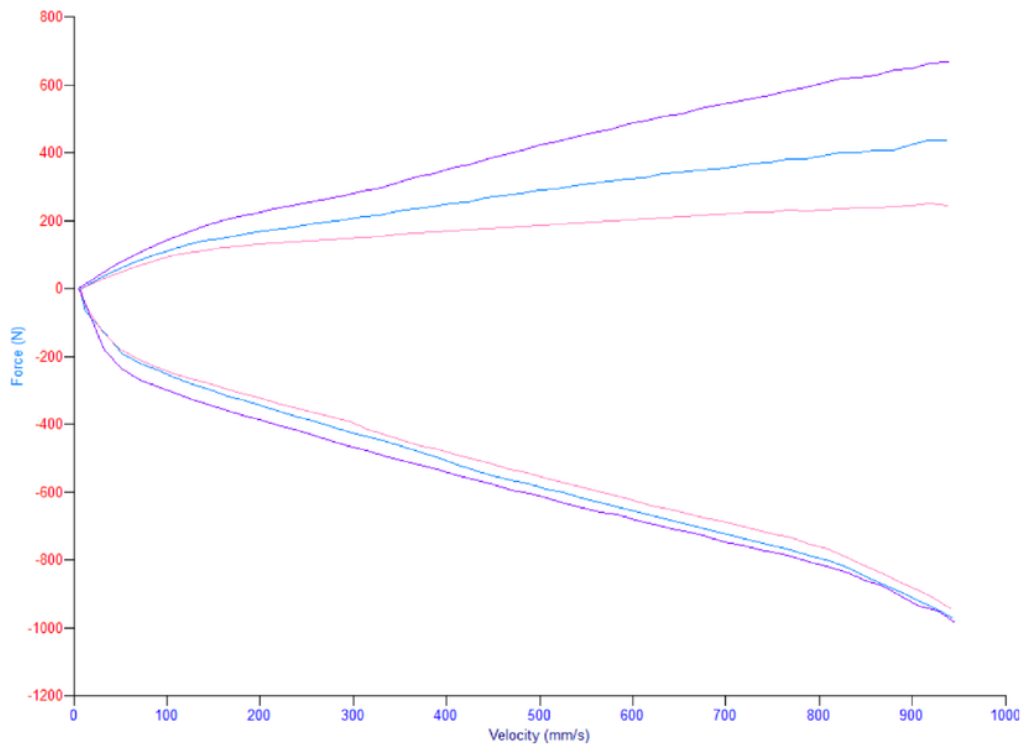


Graph 27: Full frequency force-displacement graph for 18° cone with 25 N/mm spring (red) and 100 N/mm spring (orange)

For the 11° cone with differing spring rates, the below table represents respective peak forces for compression and rebound shown graphically on graph 28.

Cone°	Spring (N/mm)	Peak Compression (N)	Peak Rebound (N)	Graph Colour
11	25	245.52	954.68	Pink
	100	435.71	990.62	Blue
	240	668.9	993.27	Purple

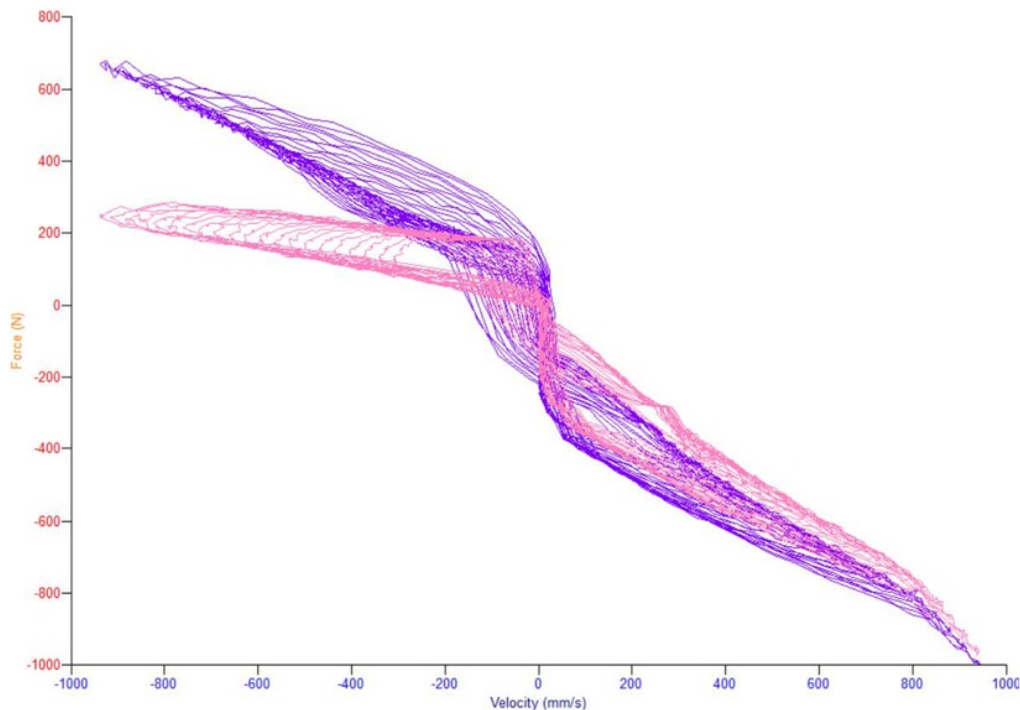
Table 7: Peak forces for 11° cone with differing spring rates and corresponding graph colour



Graph 28: Average force-velocity graph for 11° cone with differing spring rates

Similarly to graph 20 with the 18° cone, the increased spring rates amplify the gradient of compression damping with the 11° cone to take on more linear characteristics whereby peak compressive forces with the 240 N/mm spring are almost triple that of the 25 N/mm spring.

The anomalous increase in rebound damping with a softer spring which was seen with the 18° cone is not seen with the 11° cone and therefore is believed to be a result of crossover effects due to the unintentional adjustment in low-speed compression damping.

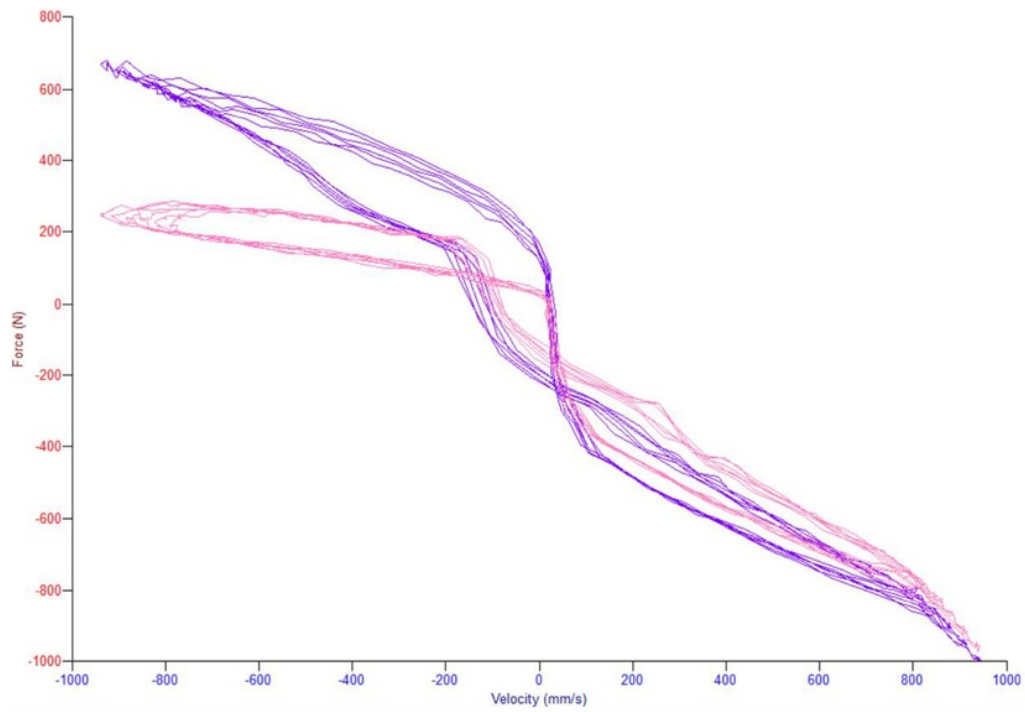


Graph 29: Full frequency force-velocity graph for 11° cone with 25 N/mm spring (pink) and 240 N/mm spring (purple)

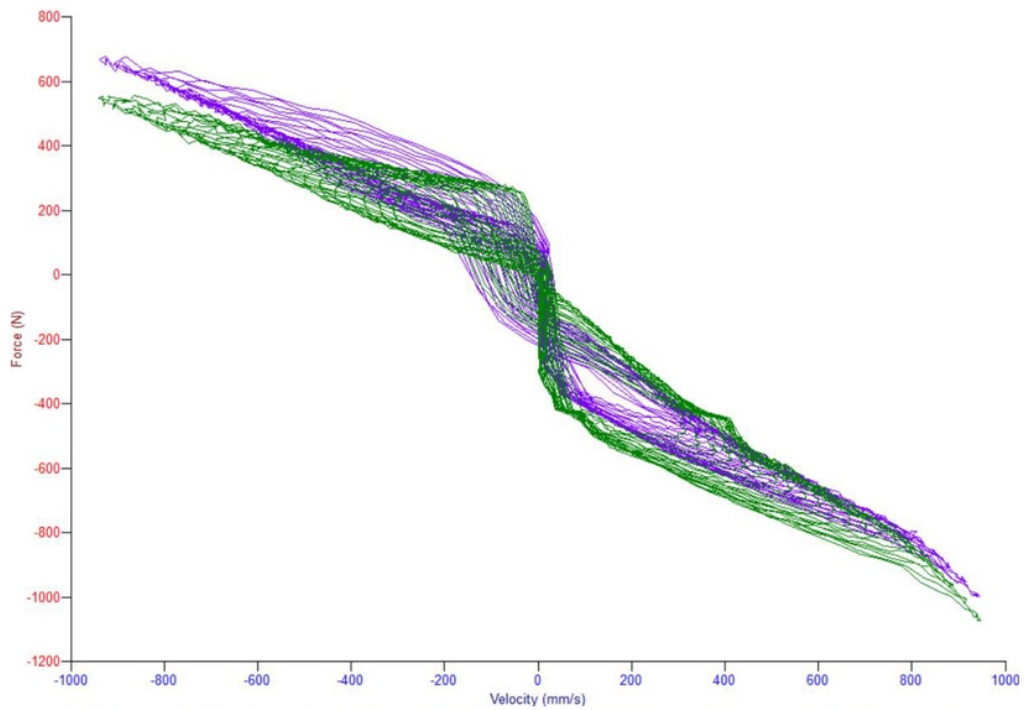
Examination of graph 29 shows drastic differences in the behaviour of the system when the 11° cone is coupled with the stiffest spring. Low speed characteristics are similar for both traces, implying that the compression adjuster was at the same setting when the system was tested. High frequency runs however exhibit unique properties such that compression damping is much higher upon acceleration than it is with deceleration, as seen emphasised on graph 30. This conflicts with results seen previously whereby deceleration forces tend to be higher, however due to the much stiffer spring combined with the steepest gradient of the 11° cone it was theorised that damping would be of a higher magnitude.

The 240 N/mm spring offers much more resistance to oil flow than any other spring tested, therefore upon initial acceleration, even at higher frequencies oil is choked through what little displacement the cone valve offers and additionally through the low speed compression circuit, resulting in the higher damping forces.

After previous analysis into the function of the low-speed compression adjuster it is believed that deceleration forces are lower due to the positioning of this adjuster. Graph 31 displays results of the 11° and 15° cones when coupled with the 240 N/mm spring.



Graph 31: High frequency cycles 2.5-3Hz for 11° cone with 25 N/mm spring (pink) and 240 N/mm spring (purple)



Graph 30: Full frequency force-velocity graphs for 11° cone (purple) and 15° cone (green) with 240 N/mm spring

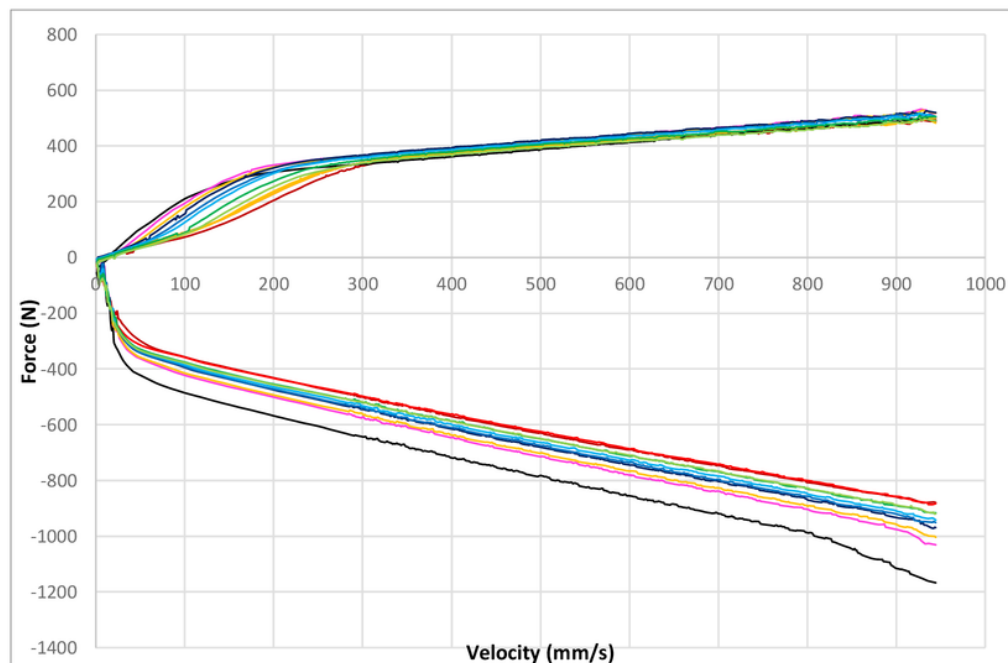
It can clearly be seen that although the 11° cone should offer more damping than the 15°, the compression damping deceleration forces at low velocities are higher with the 15° cone. Thus the hysteresis characteristics between acceleration and deceleration are a direct result of the low-speed compression adjuster position.

Due to inconsistencies in compression adjustment the results of the 15° cone with differing spring rate cannot be directly compared, though from the analysis of other cones conclusions can still be drawn on the influence that spring rate has on the system as a whole. It is apparent that changing spring rate has a more noticeable effect on damping than changing cone valve alone, whereby higher velocities exhibit more linear properties as spring rate increases.

It was earlier stated that the lack of installed preload of the spring against the cone may be attributed to the decrease in rebound damping as oil is allowed to seep through the gap between the cone and adapter. A 1mm shim was thereby placed beneath the spring to prevent this free-play in the system and to preload the spring against the cone valve.

5.5.1 Preloaded Spring

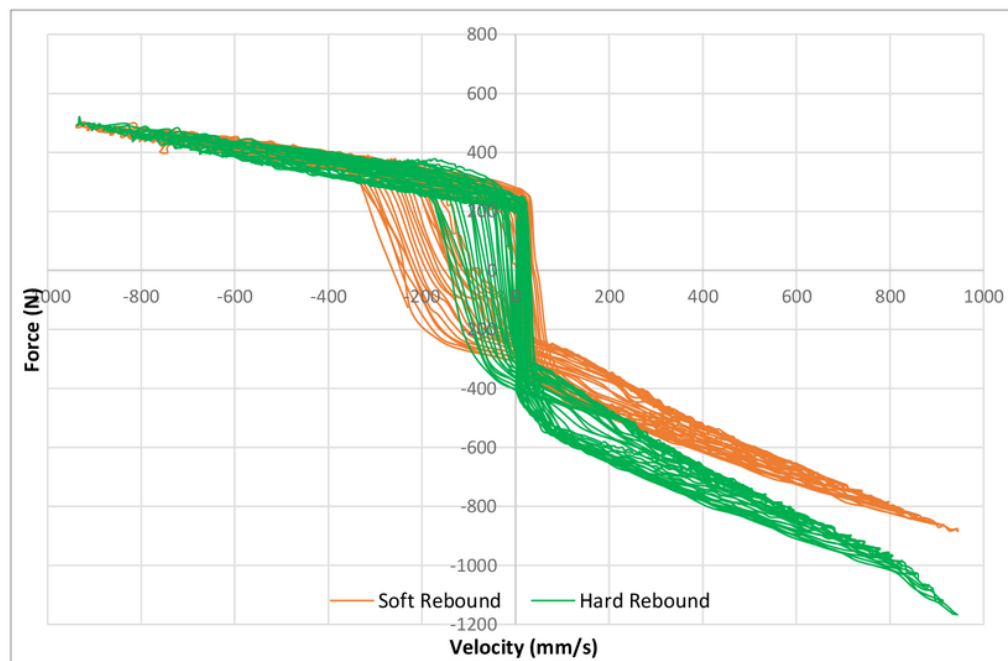
The preload shim was tested in combination with the 11° cone and 100 N/mm spring and results of changing the low-speed rebound adjuster position can be seen on the following graph. Without the preload shim, minimal significant change in compression damping was recorded in conjunction with adjustment of the rebound adjuster.



Graph 32: Average force-velocity graph showing effects of rebound adjustment with 11 cone, 100 spring and preload shim

As shown on graph 32, the addition of the preload shim and subsequent adjustment of the low-speed rebound adjuster makes an extraordinary difference to low speed compression damping. With rebound damping at a minimum position, low speed compression damping shows characteristics similar to a three stage profile, whereby two significant changes in gradient occur. As the rebound adjuster is wound in, the first two changes in gradient begin to blend together until at maximum rebound damping it holds properties of a two-stage profile again.

These changes in compression damping come as a result of the hysteresis behaviour on low-speed acceleration and deceleration. High-speed compression damping has very few losses and presents similar damping properties regardless of whether velocity is increasing or decreasing, however the differences lie in low speed reactions. The transition from compression to rebound becomes more disjointed when the rebound adjuster is at its minimum position.



Graph 33: Full frequency force-velocity graphs at maximum and minimum rebound settings for 11 cone, 100 N/mm spring

These hysteric losses on the deceleration of the compression stroke are believed to be exaggerated by the introduction of the preload shim and therefore the loss of a route for oil to pass through, forcing oil to instead travel through the rebound circuit. Previous damping losses in this area have been noted but are assumed to be caused by the increase in speed of the dynamometer, since there have been no notable changes in this characteristic due to other variables. However graph 33 shows a correlation between rebound setting and this hysteresis and with all other variables kept equal it suggests causation is due to the preload shim directing oil through the low-speed needle valve.

Thus the results of adding a preload shim outline the importance of the free-play between cone and spring to the transition from compression to rebound. Allowing a slight passage of oil prevents too much opposing pressure building up before the system has completed its compression stroke and therefore presents a much smoother transition.

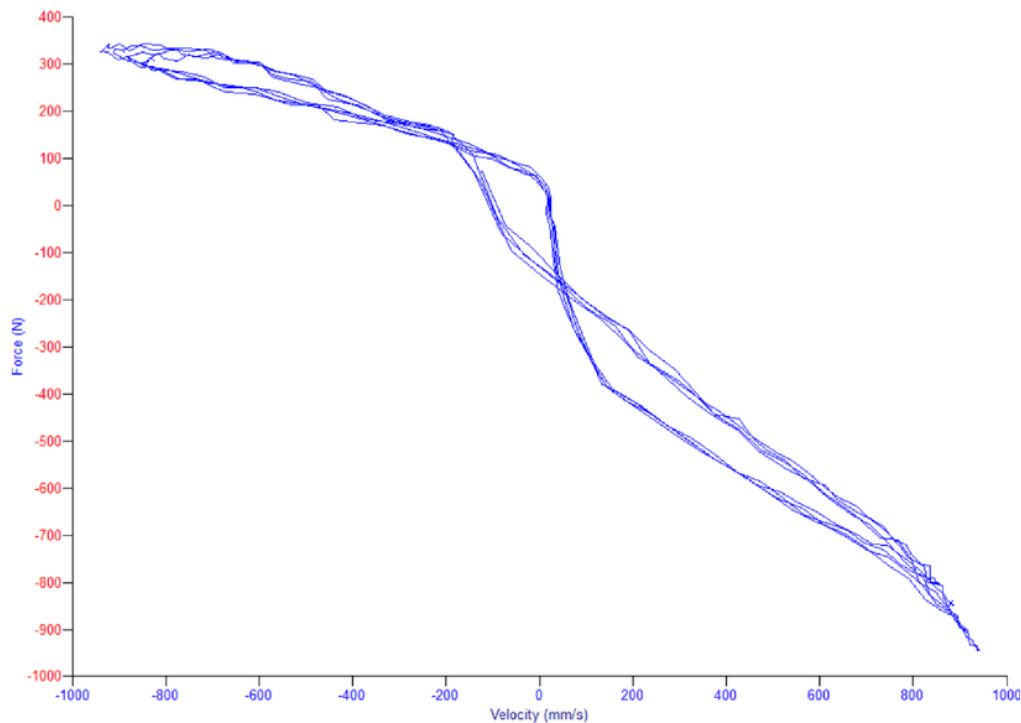
5.6 Variations in Low Frequency and High Frequency Damping

Throughout testing and analysis it has been noted on numerous occasions the hysteresis that exists in damping during low speed and high speed oscillations, whereby damping characteristics change between acceleration and deceleration of the system.

Most noticeable is the hysteresis at low-speed which occurs regardless of spring rate or cone angle. Low-speed oscillations are generally regarded as those caused by dynamic motion of acceleration or braking and its likely the rider will be able to sense how the damping changes through the stroke much easier than with high-speed oscillations. The WP cone valve damping fork leg is paired with an air sprung leg which works in unison to control the motorcycles motion. Unlike conventional sprung forks, air springs change spring rate as a function of displacement, whereby the effective spring rate increases exponentially as the air in the system is compressed. Damping however is a function of velocity rather than displacement, though it was seen in the force-displacement graphs that damping does present differences throughout the stroke.

These differences are exaggerated through adjustment of the low-speed compression adjuster towards its maximum position. It is believed that the low-speed hysteresis has been purposely designed to function in tandem with the changing spring rate of the air sprung leg. Low-speed oscillations often occur over large displacements, therefore at the top of the stroke the corresponding spring rate would be relatively small. As the forks are compressed the spring rate increases, and as has been analysed damping also increases as the suspension velocity transitions between acceleration and deceleration. Therefore at the bottom of the compression stroke where the spring rate is maximised, the damping force is also of greatest magnitude and offers the most support.

High frequency oscillations generally correspond with small displacements and therefore the changes in air spring rate would be minimal. Likewise the hysteresis in compression damping at higher frequencies is seen on graph 34 to be much less defined and damping remains somewhat more equal throughout acceleration and deceleration. Damping support at high velocities therefore has been designed to have equivalent properties due to the minimal changes in displacement and thus spring rate.

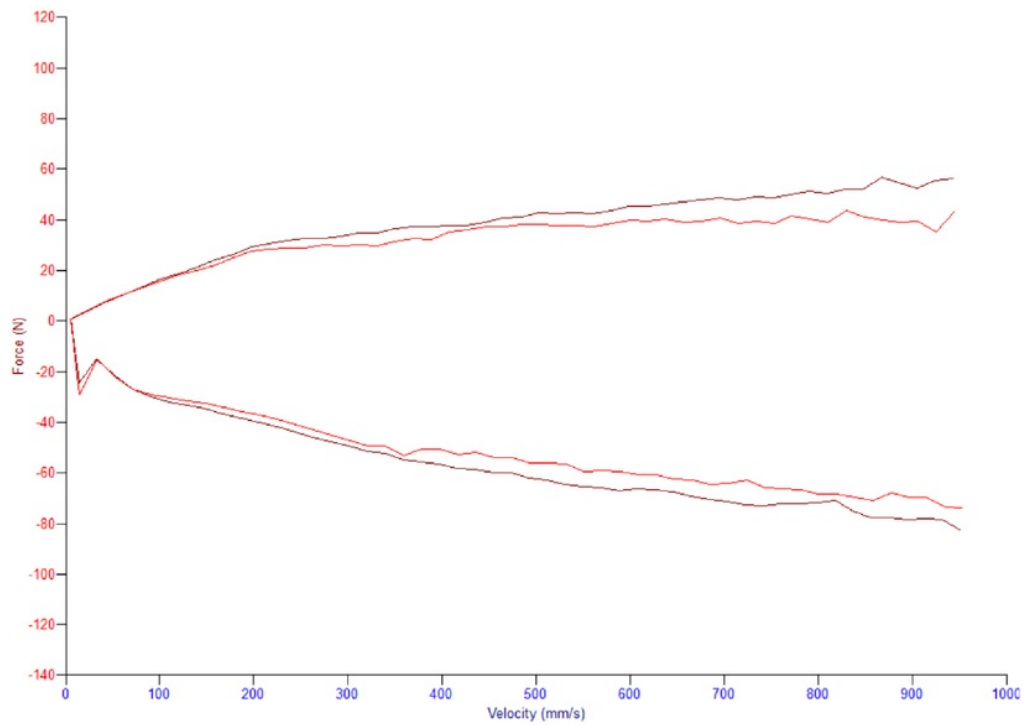


Graph 34: High frequency force-velocity cycles

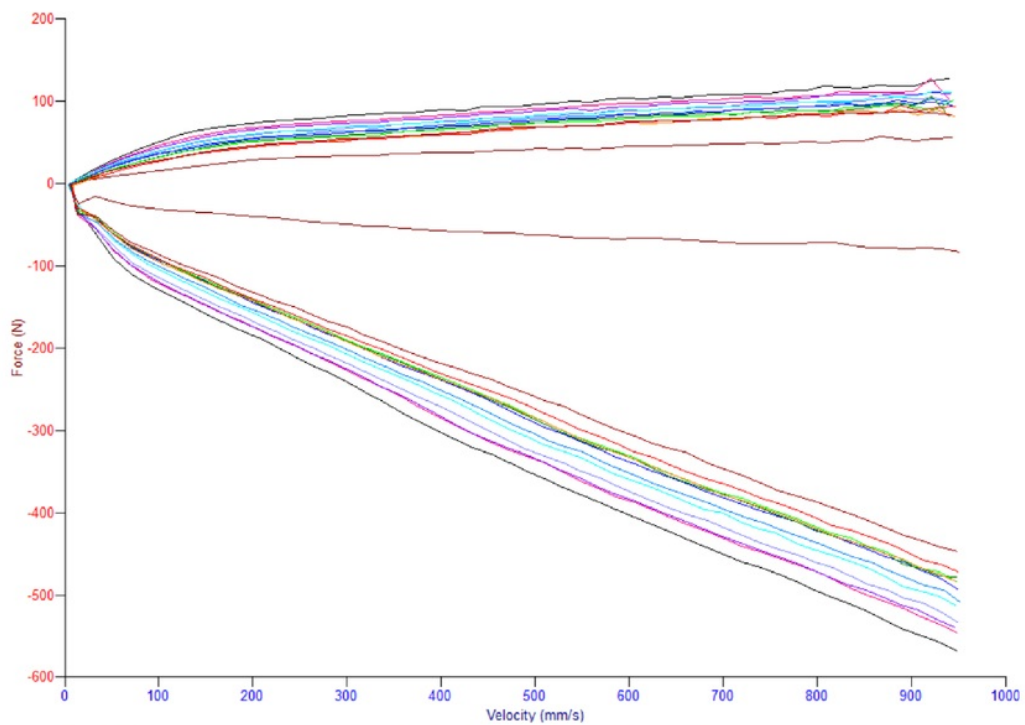
5.7 Comparison Against Standard Forks

To complete the practical analysis of the WP Cone Valve forks and fully understand the benefits it is necessary to compare the system to one with standard valving. The forks tested throughout this study fit to numerous KTM and Husqvarna off-road and enduro motorcycles, including that of the Husqvarna TE250i which the author had access to. The standard WP forks from the TE250i were thus tested on the damper dynamometer using the same methodology as the cone valve forks. However, the standard forks have damping in both fork legs whereby the left leg is responsible for compression damping and the right leg controls rebound damping. As such the forks are tested individually but considered as a whole system.

Graph 35 shows the results of the compression damping leg whereby an extreme lack of damping can be seen. The two traces represent the data recorded with the compression adjuster at maximum and minimum positions. At maximum position the peak compression damping force is only 56.3N and shows an extreme flaw in the design and adjustability of the system. The following graph 36 represents the damping forces of the rebound fork leg throughout its range of adjustment.



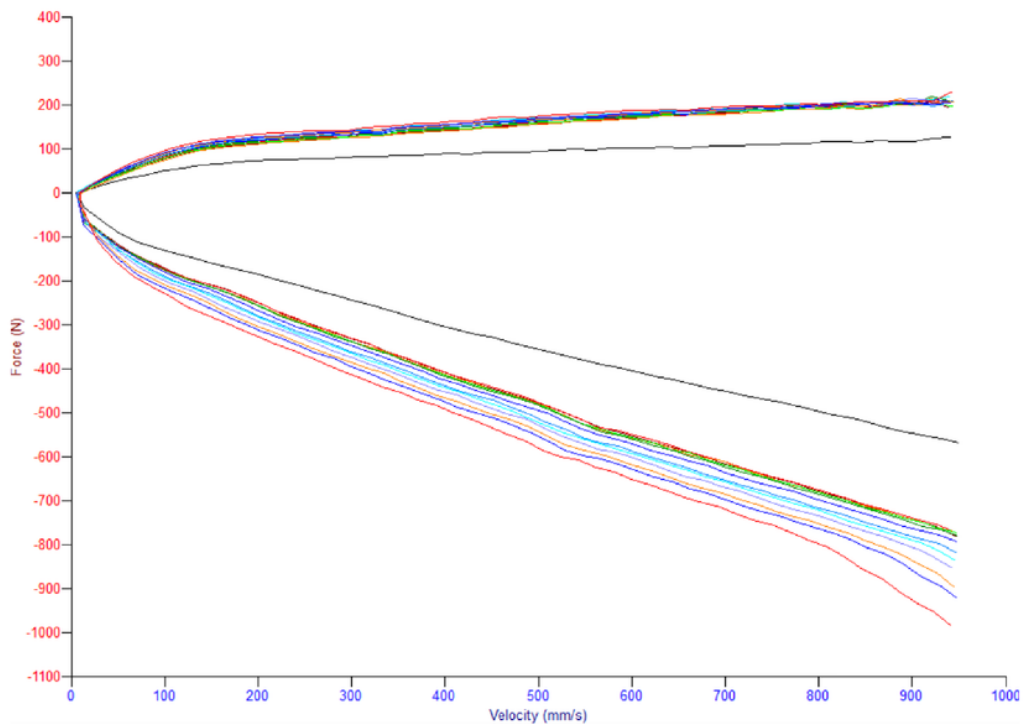
Graph 36: Average force-velocity graphs of standard WP compression fork leg



Graph 35: Average force-velocity graphs of standard WP rebound fork leg, with compression trace for comparison

Compression damping forces from testing the rebound leg are measurably more than that of the actual compression leg indicating some serious crossover effects from the rebound adjuster. This may not necessarily be a bad thing however judging by the flawed design of the compression leg offering next to no damping control.

Graph 37 compares the maximum compression and rebound damping forces from graph 36 to the range of damping offered by the softest cone valve combination of 18 cone and 25 spring whereby even this arrangement offers more support to both low-speed and high-speed damping. The standard WP forks may be adequate for road use, however as a performance system they do not measure up to the damping characteristics that the cone valve system offers.

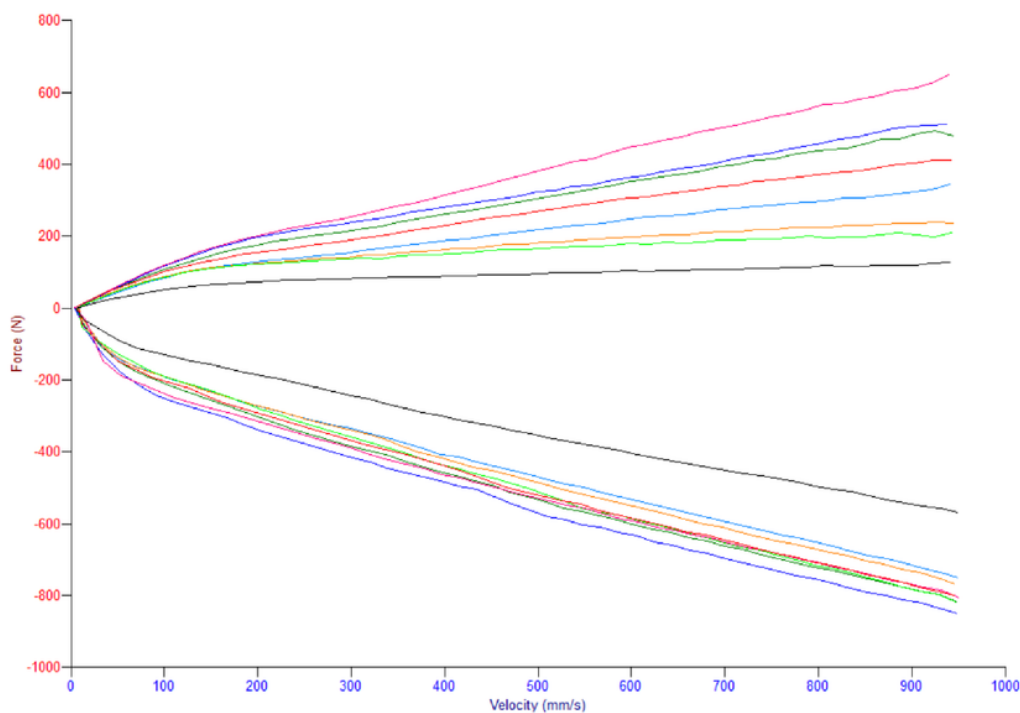


Graph 37: Average force-velocity graphs for rebound adjustment range of WP cone valve forks with 18 cone and 25 spring, compared against maximum forces of standard forks

6. Post-Processing Analysis

From the damper dynamometer data collected throughout the practical analysis stage a select number of cases were chosen for post-processing analysis. Due to the progressive nature of the high-speed rebound damping with the rebound adjuster at its maximum position, it was decided to select data traces at 10 clicks in from maximum for subsequent analysis. This setting is within the recommended range of adjustment suggested by WP and therefore likely to be a common set-up.

Data traces selected are a variation of cone valve and spring rate combinations, in addition to a trace from the standard WP forks, whereby the average force-velocity graphs will be processed into graphs representative of damping ratio. Though damping characteristics change between acceleration and deceleration, it is necessary to take a more holistic approach and consider damping forces as an average. The benefit of this is that any unintentional changes in low-speed compression damping during testing won't have a negative impact on the average results, though cross-over effects may have to be considered for rebound damping forces.



Graph 38: Selected data traces for analysis

Graph 38 shows the data traces selected for comparison and analysis in regard to damping ratios and transmissibility characteristics and the correlating valving combinations for each trace can be seen on the following table.

Cone	Spring Rate (N/mm)	Peak Compression(N)	Peak Rebound (N)	Graph Colour
11	25	237.54	780.16	
	100	411.06	817.66	
	240	640.42	812.29	
15	100	345.02	750.79	
	240	516.76	858.99	
18	25	210.23	829.46	
	240	480.14	820.38	
WP STD	n/a	125.58	575.79	

Table 8: Peak forces for all selected data traces and corresponding graph colour

6.1 Calculation of Damping Ratio

As described in section 2.1.2, damping ratio is known as the ratio between actual damping coefficient and the critical damping value given by:

$$\zeta = \frac{c}{c_{crit}} = \frac{c}{2\sqrt{km}} \quad [52]$$

Where:

$$c = \frac{\text{Force}}{\text{Velocity}} = \text{Ns/m} \quad [53]$$

6.1.1 Damping Constant

From the measured dynamometer data, damping coefficients can be extracted from the force tables given in the software as shown in figure 11. Compression and rebound forces are given for every 1.2mm/s; the minimum allowable velocity interval, thus enabling the calculation of damping coefficients for each respective data trace.

	Bump force	Rebound force	Peak Bump	Peak	Bump force	Rebound	Peak Bump	Peak	Bump force	Rebound	Peak Bump	Peak	Bump force
0.00 mm	-25.05 N	-25.05 N	-291.70 N	-291.70 N	-12.86 N	-12.86 N	-432.80 N	-432.80 N	-22.95 N	-22.95 N	-289.80 N	-289.80 N	-17.31 N
1.20 mm	-39.03 N	-18.90 N	-291.81 N	-290.01 N	-25.37 N	-11.51 N	-432.72 N	-430.33 N	-28.85 N	-15.77 N	-289.99 N	-288.23 N	-47.98 N
2.40 mm	-36.26 N	-17.79 N	-291.92 N	-288.32 N	-56.85 N	-10.17 N	-432.64 N	-427.86 N	-42.64 N	-14.54 N	-290.17 N	-286.67 N	-53.38 N
3.60 mm	-39.63 N	-16.48 N	-292.02 N	-286.62 N	-56.52 N	-8.85 N	-432.56 N	-425.39 N	-41.66 N	-13.38 N	-290.36 N	-285.10 N	-71.22 N
4.80 mm	-38.59 N	-15.19 N	-292.13 N	-284.93 N	-48.80 N	-7.59 N	-432.48 N	-422.92 N	-36.92 N	-12.24 N	-290.55 N	-283.53 N	-72.35 N
6.00 mm	-37.91 N	-13.80 N	-292.24 N	-283.24 N	-56.38 N	-6.36 N	-432.40 N	-420.46 N	-41.48 N	-11.10 N	-290.73 N	-281.97 N	-65.86 N
7.20 mm	-39.29 N	-12.42 N	-292.35 N	-281.55 N	-53.34 N	-5.03 N	-432.32 N	-417.99 N	-42.11 N	-9.91 N	-290.92 N	-280.40 N	-67.68 N
8.40 mm	-47.72 N	-11.08 N	-292.45 N	-279.85 N	-59.82 N	-3.75 N	-432.24 N	-415.52 N	-43.59 N	-8.62 N	-291.10 N	-278.83 N	-68.99 N
9.60 mm	-51.62 N	-9.59 N	-292.56 N	-278.16 N	-68.58 N	-2.44 N	-432.16 N	-413.05 N	-48.85 N	-7.21 N	-291.29 N	-277.27 N	-68.48 N
10.80 mm	-63.37 N	-8.03 N	-292.67 N	-276.47 N	-74.47 N	-1.06 N	-432.08 N	-410.58 N	-51.71 N	-5.77 N	-291.48 N	-275.70 N	-67.59 N
12.00 mm	-68.45 N	-6.16 N	-292.78 N	-274.78 N	-77.10 N	0.32 N	-432.00 N	-408.11 N	-56.69 N	-4.32 N	-291.66 N	-274.13 N	-70.20 N
13.20 mm	-71.55 N	-4.29 N	-292.89 N	-273.08 N	-89.72 N	1.80 N	-431.93 N	-405.64 N	-66.53 N	-2.88 N	-291.85 N	-272.57 N	-74.87 N
14.40 mm	-78.12 N	-2.82 N	-292.99 N	-271.39 N	-93.02 N	3.32 N	-431.85 N	-403.17 N	-75.53 N	-1.50 N	-292.04 N	-271.00 N	-76.45 N
15.60 mm	-81.81 N	-1.48 N	-293.10 N	-269.70 N	-102.58 N	4.85 N	-431.77 N	-400.71 N	-78.80 N	-0.06 N	-292.22 N	-269.43 N	-79.26 N
16.80 mm	-88.56 N	-0.18 N	-293.21 N	-268.01 N	-106.72 N	6.26 N	-431.69 N	-398.24 N	-81.11 N	1.40 N	-292.41 N	-267.87 N	-82.43 N
18.00 mm	-91.34 N	1.13 N	-293.32 N	-266.31 N	-109.49 N	7.61 N	-431.61 N	-395.77 N	-80.40 N	2.96 N	-292.60 N	-266.30 N	-86.90 N
19.20 mm	-98.95 N	2.43 N	-293.42 N	-264.62 N	-115.48 N	8.92 N	-431.53 N	-393.30 N	-83.07 N	4.65 N	-292.78 N	-264.73 N	-89.70 N
20.40 mm	-102.22 N	3.72 N	-293.53 N	-262.93 N	-118.29 N	10.26 N	-431.45 N	-390.83 N	-94.18 N	6.30 N	-292.97 N	-263.17 N	-91.31 N
21.60 mm	-104.75 N	4.93 N	-293.64 N	-261.24 N	-116.36 N	11.60 N	-431.37 N	-388.36 N	-93.06 N	7.91 N	-293.15 N	-261.60 N	-93.94 N
22.80 mm	-107.29 N	6.14 N	-293.75 N	-259.54 N	-124.56 N	12.92 N	-431.29 N	-385.89 N	-100.17 N	9.53 N	-293.34 N	-260.03 N	-96.23 N
24.00 mm	-109.56 N	7.51 N	-293.85 N	-257.85 N	-127.82 N	14.17 N	-431.21 N	-383.42 N	-102.95 N	11.23 N	-293.53 N	-258.47 N	-98.45 N

Figure 11: Force-velocity tables

6.1.2 Calculation of Air Spring Rates

For further calculation of damping ratio it is necessary to calculate the critical damping value for this system. Critical damping relies on both the mass and spring rate of the system as a whole, however it was previously stated that the air sprung fork leg causes spring rate to change progressively with travel, thus it cannot be defined as having a singular value. Therefore using Boyle's law and the principles of polytropic processes whereby:

$$P_1 V_1 = P_2 V_2 \quad [54]$$

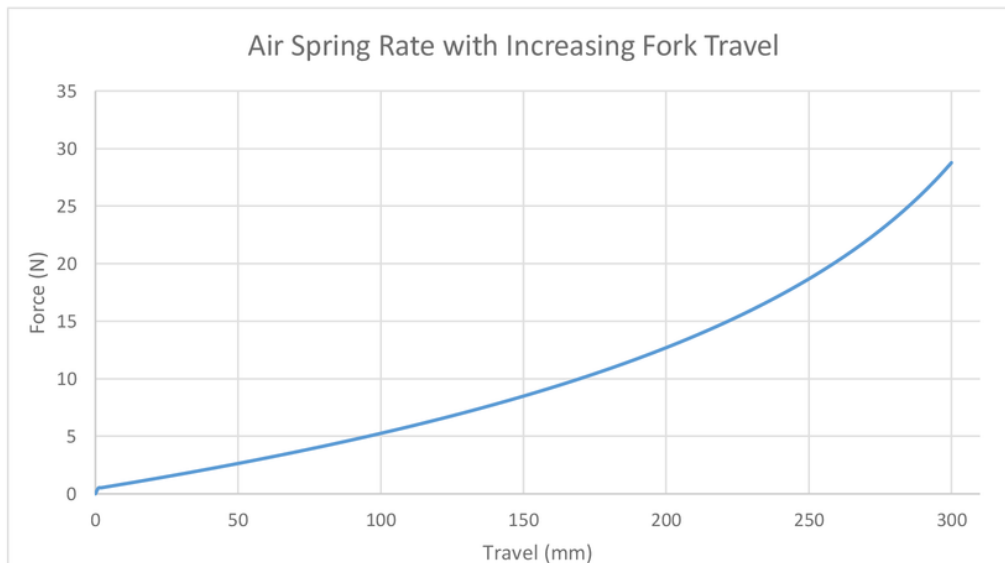
$$PV^n = \text{Constant} \quad [55]$$

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n \quad [56]$$

$$\therefore P_2 = P_1 \left(\frac{V_1}{V_2}\right)^n \quad [57]$$

Pressures following that of the initial conditions can therefore be calculated with the change in volume of the gas chamber. The cartridge is measured to have a 34mm internal diameter and length of 360mm with 300mm available travel. Assuming cross-sectional area remains constant throughout the stroke, effective spring force can then be calculated for each millimetre of fork travel. A polytropic index of 1.1 was used to calculate pressures after experimental data conducted by (McGladdery, et al., 2018). It was thereafter decided to take the average spring rate across the travel for the initial calculation of damping ratios, assuming an initial air pressure of 11psi recommended by WP (WP, 2017).

$$\text{Average spring rate} = 10.24\text{N/mm}$$



Graph 39: Air spring rate with increasing fork travel

6.1.3 Damping Ratio

The mass of a Husqvarna TE250i is given to be 105.2kg (Husqvarna, 2019). Assuming an average rider weight of 72kg (Condon, 2018) this gives a total machine and rider mass of 177.2kg. Negating the effects of rider movement on the position of the centre of gravity, the mass distribution is assumed to be 50:50 with no bias towards front or rear, and therefore the mass supported by the front suspension is:

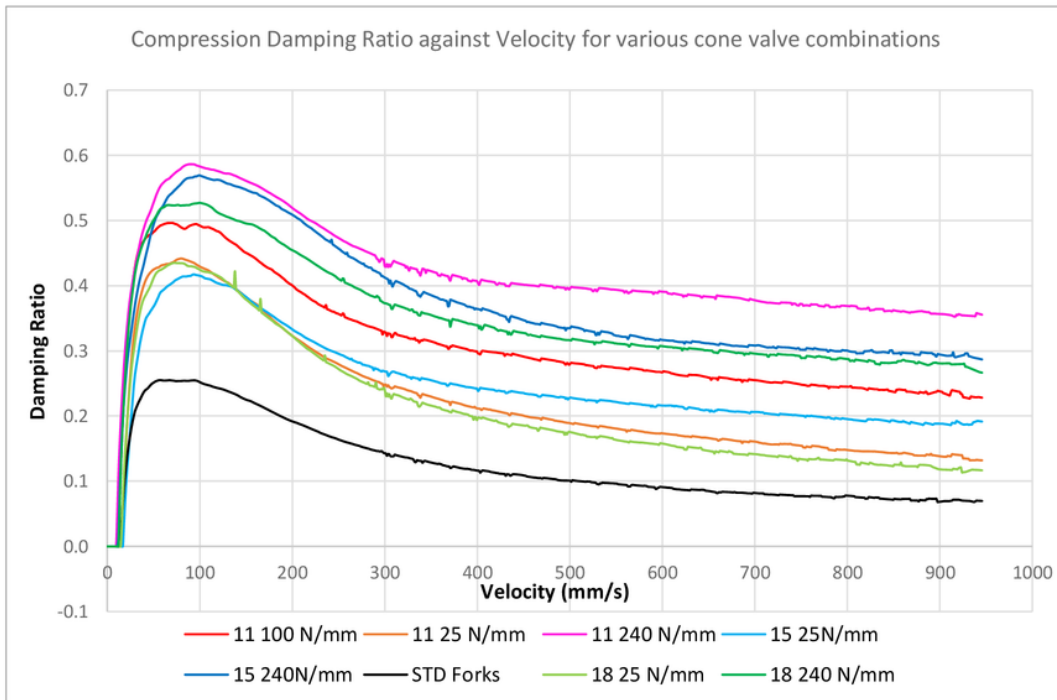
$$\frac{177.2}{2} = 88.6 \text{ kg} \quad [58]$$

Thus:

$$\zeta = \frac{c}{2\sqrt{10.24 \times 10^3 \times 88.6}} = \frac{c}{1905.096} \quad [59]$$

Compression Damping Ratios

Damping ratios can therefore be calculated for the selected data traces, Excel screenshots of which can be seen in the appendix. Graph 40 below compares the results of post-processing the compression damping data into damping ratio profiles, plotted against velocity. Where previously it was seen that low-speed compression damping characteristics seemed relatively similar, the values of damping ratio at low velocities are somewhat different.



Graph 40: Compression Damping Ratio against Velocity for various cone valve combinations

(Shaikh, et al., 2013) idealised that damping ratios should be higher at low velocities to reduce the amplification zone of transmissibility characteristics, whereas at high suspension velocities damping ratio should be reduced. This correlates with the results obtained in graph 40, whereby all selected traces conform with this idealisation. It therefore could be said that the trace with the largest difference in maximum low speed damping ratio and minimum high speed damping ratio would perform the most efficiently.

Cone	Spring Rate (N/mm)	Peak ζ	Minimum ζ	Average ζ	Graph Colour
11	25	0.442	0.132	0.310	Orange
	100	0.496	0.228	0.268	Red
	240	0.586	0.356	0.231	Pink
15	100	0.417	0.192	0.226	Blue
	240	0.569	0.287	0.282	Dark Blue
18	25	0.435	0.117	0.318	Green
	240	0.527	0.267	0.260	Dark Green
WP STD	n/a	0.255	0.070	0.185	Black

Table 9: Damping ratio data for each selected trace

Table 9 outlines the differences in maximum and minimum damping ratios for each selected valve combination, whereby the 3 values in bold are the three traces showing the greatest differences. The 25 N/mm paired with the 11° and 18° cones show the largest variations in damping ratio however have very similar overall characteristics. Thus it was decided to further analyse only that of the 18° cone, in addition to the 15° cone with 240 N/mm spring, which also exhibited a large difference in damping ratio but as a whole showed substantially different characteristics. The transmissibility of these two final traces will be compared against that of the standard WP forks, which showed the least change in maximum and minimum damping ratios.

It is important to reiterate that the damping ratios calculated in graph 40 are with an average air spring rate of 10.24N/mm, whereas holistically damping ratio depends on fork travel since critical damping changes with spring rate. Thus, figure 12 shows the effective compression damping ratios throughout the 300mm fork travel. As can be seen damping ratios begin very high due to the very low spring rates, and decrease in magnitude the further into the stroke the system is.

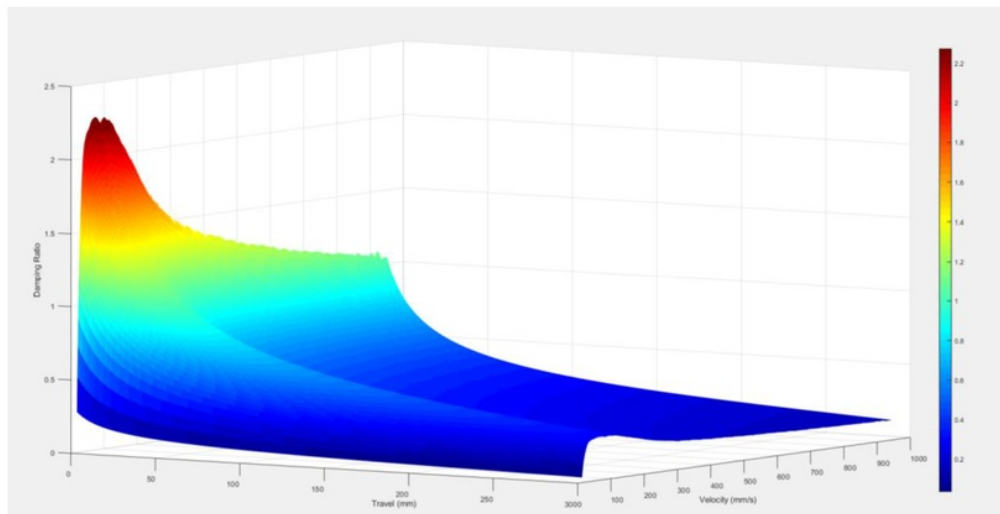
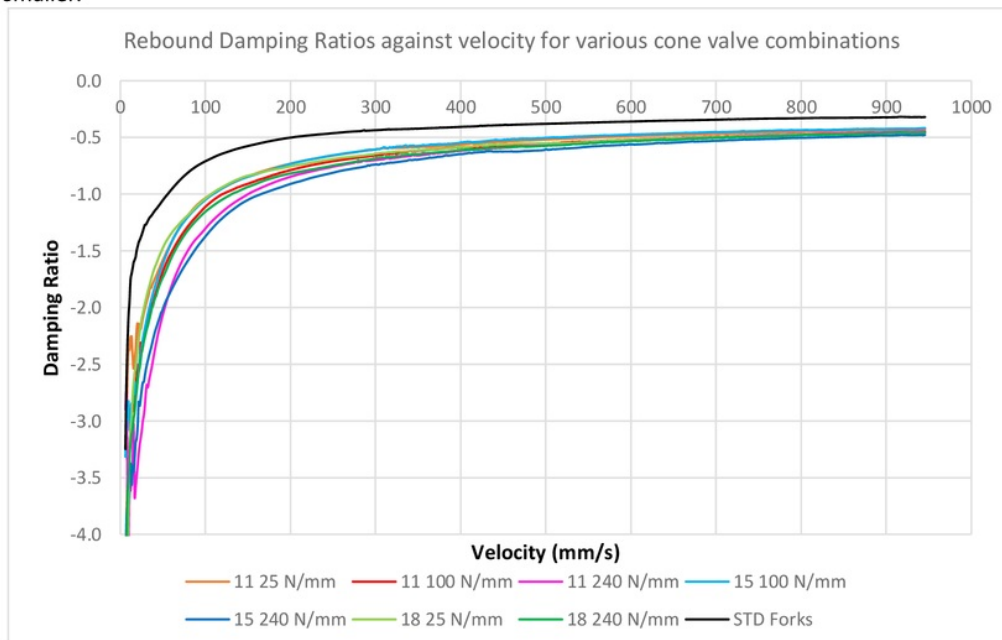


Figure 12: 3D graph showing how damping ratio increases with fork travel

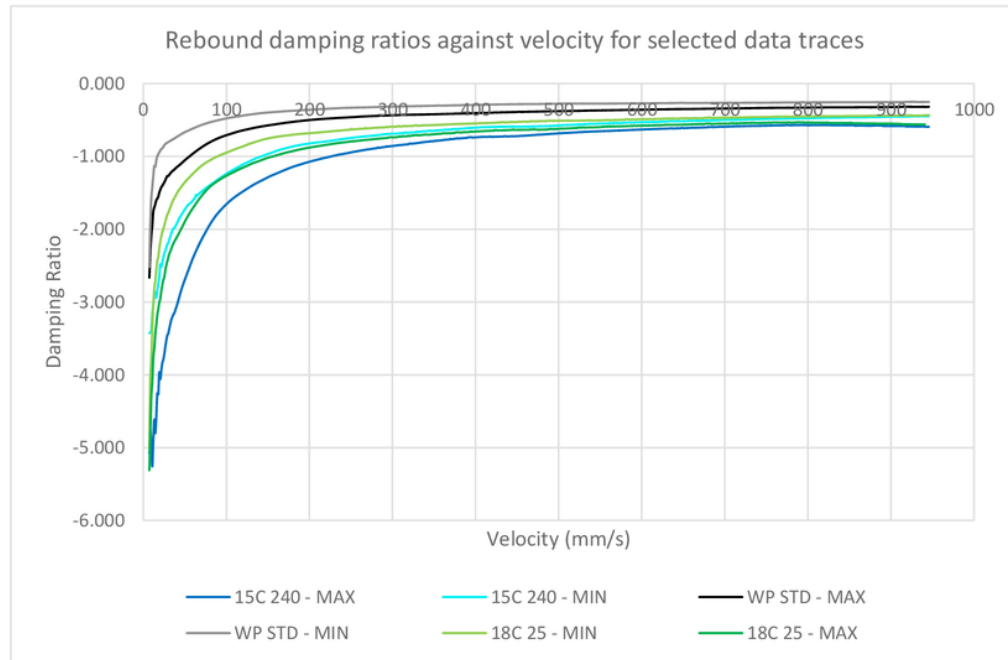
Rebound Damping Ratios

Damping ratio profiles for rebound damping are drastically different from that of compression, however between traces don't necessarily represent much change due to the fact that altering cone and spring rate is to primarily to control compression damping. As can be seen on graph 41, at low velocities damping ratios are extremely high and as velocity increases damping ratio gets progressively smaller.



Graph 41: Rebound damping ratios against velocity for various cone valve combinations

It was therefore decided to instead compare the damping ratios of the three selected traces at maximum and minimum rebound adjustment, seen on graph 42. Damping ratios are output as negative values due to the negative damping forces measured by the dynamometer. Therefore the lowest negative indicates the highest damping ratio.



Graph 42: Rebound damping ratios against velocity for selected data traces

The trends in rebound damping ratios are much more discernible when analysing the rebound traces at maximum and minimum rebound adjustment for the selected valve combinations. At low speeds, the damping ratios are extremely high as the system begins the rebound stroke and get progressively lower as velocity increases.

Greatest rebound damping ratios are seen with the 15° cone and 240 N/mm spring with the low-speed adjuster at maximum position, whereas the lowest ratios are that of the standard WP forks. Even with the rebound adjuster wound all the way in damping ratios do not surpass that of the 18° cone and 25 N/mm spring with minimum damping adjustment.

6.2 Transmissibility

As discussed in section 2.2.1 the transmissibility of a system describes the ratio of input to output frequencies, such that should the forcing frequency coincide with the natural frequency of the system the amplification of force or displacement is of greatest magnitude. Transmissibility is generally plotted against varying input frequencies and a singular damping ratio. Therefore the fact that damping ratios change with velocity poses a complication in representing a global interpretation of damping performance. Taking the average damping ratio may give an insight into the differences between the valve combinations, however for a true understanding 3D graphs must be created showing how damping truly reacts to suspension behaviour.

6.2.1 Compression: Displacement Transmissibility

$$T_d = \frac{X}{Y} = \frac{1 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2}{\sqrt{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta \frac{\omega}{\omega_n}\right)^2}}$$

To simplify the above equation one can define $\frac{\omega}{\omega_n}$ as the frequency ratio given the symbol r . Due to the fluctuating nature of the natural ride frequency of the system as a result of changing spring rates, defining this as a ratio allows the calculation of transmissibility simply in reference to the natural frequency, whereby should this change the transmissibility plot still applies. Thus:

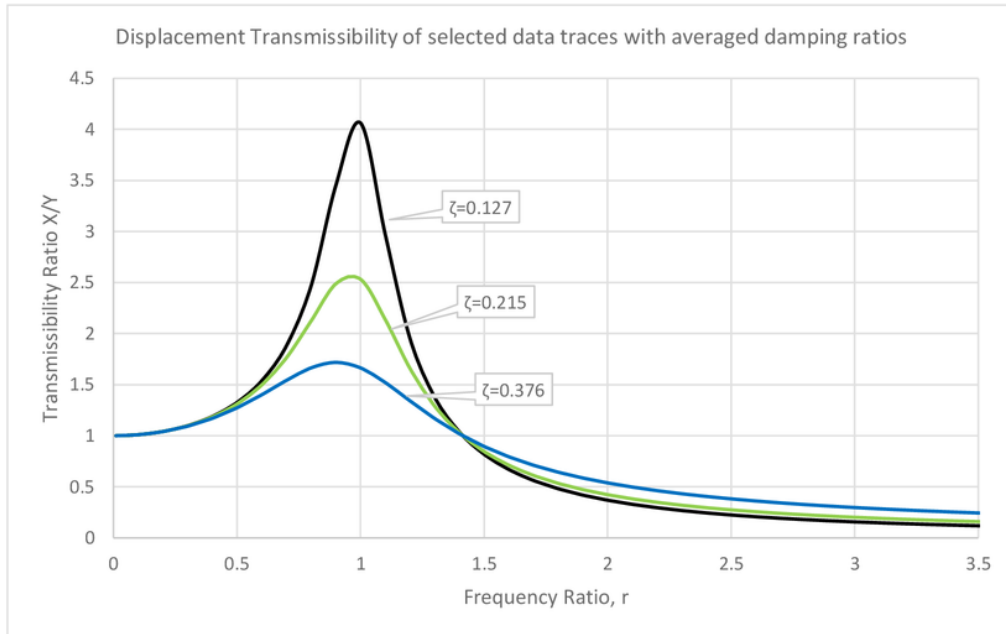
$$T_d = \frac{1 + (2\zeta r)^2}{\sqrt{(1 - r^2)^2 + (2\zeta r)^2}}$$

The average damping ratios for the selected valve combinations are given below.

Valve Combination		Average ζ	Graph Colour
15° Cone	240 N/mm spring	0.376	Blue
18° Cone	25 N/mm Spring	0.215	Green
Standard WP Forks		0.127	Black

Table 10: Average Damping Ratios

When $r < \sqrt{2}$ is known as the amplification zone (Institute of Engineering: SUT, n.d.), whereby input frequencies are amplified and are of greatest magnitude when the input frequency approaches the natural frequency of the system, as can be seen on graph 43. Lower damping ratios in this area cause the transmissibility of road input frequencies to drastically increase such that oscillation displacement is much higher than that of static equilibrium; thus the oscillations transmitted to the sprung mass are increased. High damping ratios at frequency ratios less than $\sqrt{2}$ attenuate this amplification, hence why it is seen that at low velocities damping ratios are highest.



Graph 43: Transmissibility Ratio against Frequency Ratio

Following the isolation point where $r = \sqrt{2}$, as frequency ratio increases displacement transmissibility drops below that of the static displacement. Thus high frequency oscillations correlate with minimised displacement transfer. Graph 43 shows the respective transmissibility plots for the averaged damping ratios of the selected traces. With higher average damping ratio, the 15° cone with 240 N/mm spring exhibits beneficial transmissibility characteristics at low frequency ratios and therefore increased ride comfort. However at higher frequencies the responsiveness of the system is worse and lower damping ratios are preferred to minimise the transference of oscillations, making the standard WP forks preferable at high suspension frequencies.

Nevertheless the above graph remains to be an unrealistic representation of damping performance as a whole due to the changing damping ratios with velocity. As such transmissibility was then calculated in excel using the actual measured damping ratios with respect to velocity shown in graph 40. The values were then plotted in MATLAB as a 3D graph to show how damping behaviour changes.

Seen on figure 13 is the 3D graph for compression damping transmissibility for the 15° cone and 240 N/mm combination. Displacement transmissibility ratio is presented on the Z-axis, velocity in mm/s on the X-axis and frequency ratio on the Y-axis. As damping ratio profile changes with velocity, the differences in transmissibility characteristics can be seen changing with frequency. Peak transmissibility occurring at initial velocity is primarily due to the climbing damping ratios early in the stroke.

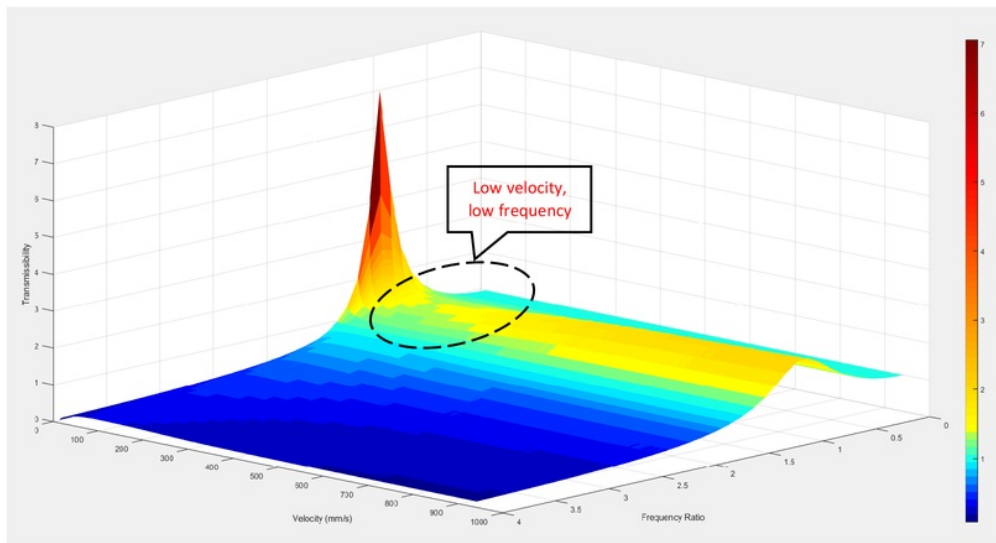


Figure 13: 3D graph of displacement transmissibility against velocity and frequency ratio for 15° cone and 240N/mm spring

Since damping is a property of velocity alone, yet frequency is a function of both velocity and displacement, figure 13 is a representation of transmissibility across the entire scope of suspension behaviour. High velocity and high frequency oscillations show the least transmissibility in displacement due to the low damping ratio. However, should such velocities occur over a large travel the resultant transmittance of displacement would increase.

The 3D transmissibility plot of the 18° cone and 25 N/mm spring can be seen below on figure 14.

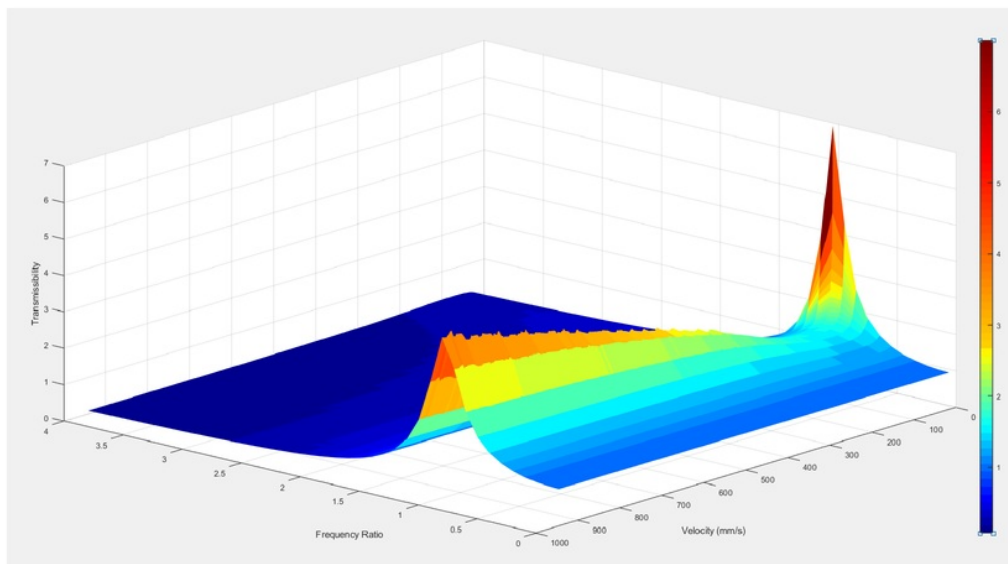


Figure 14: 3D graph of displacement transmissibility against velocity and frequency ratio for 18° cone and 25N/mm spring

The 18° cone and 25 N/mm spring previously exhibited lower damping characteristics in regard to overall damping force as well as damping ratios. As can be seen in figure 14 this results in higher transmissibility at lower frequencies. High frequency characteristics are very much similar for both combinations and it is unlikely that the average rider could differentiate between the two systems in such conditions. The increase in low frequency is therefore the most noticeable and important difference. This change correlates with a softer feeling front end when negotiating low-frequency road inputs which the rider may feel in the movement upon initial compression.

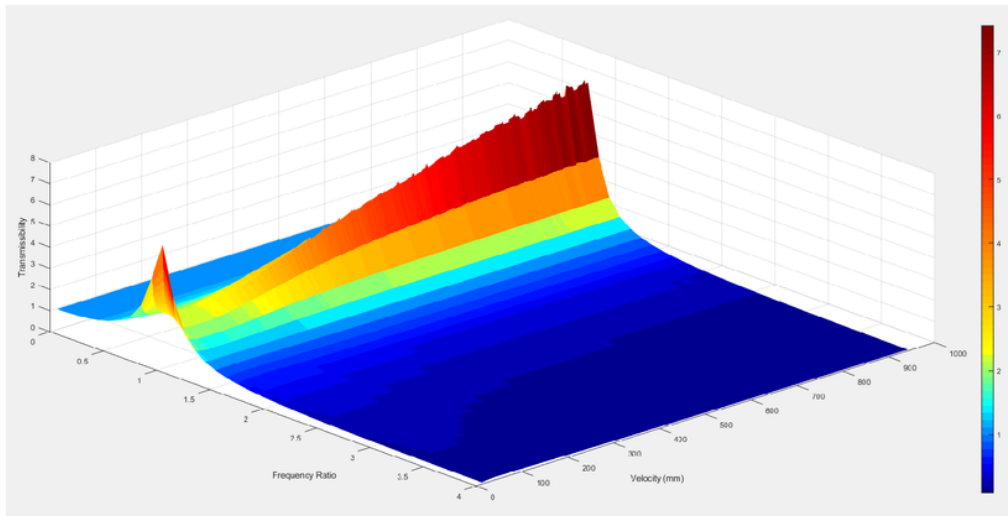
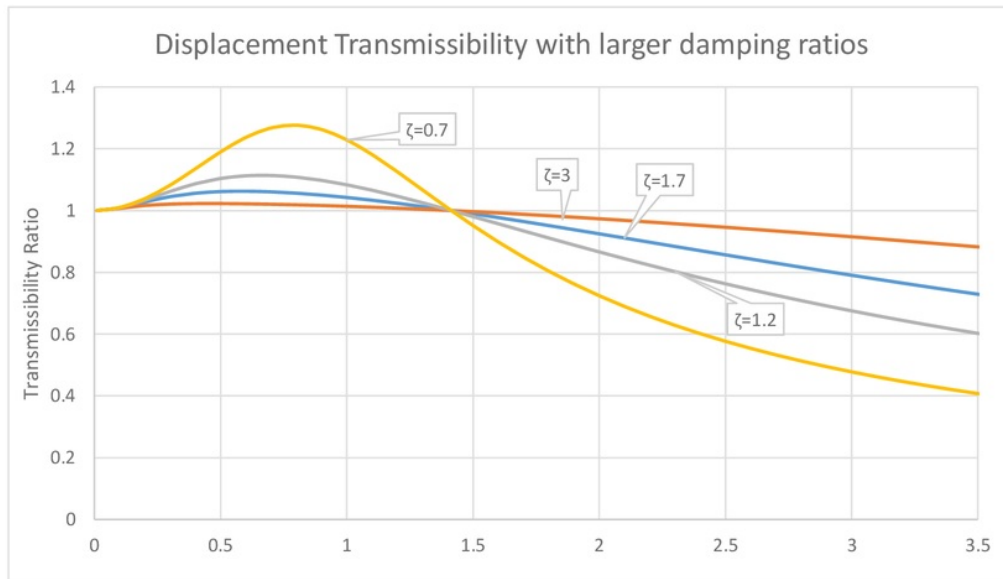


Figure 15: 3D graph of displacement transmissibility against velocity and frequency ratio for standard WP forks

Transmissibility of the standard WP forks shows extreme levels of displacement transfer at low frequencies, such that the damping offers very little control of compression movements. Such large amplifications in input frequencies imply that front end stability would be compromised and such damping would be insufficient for performance purposes.

6.2.2 Rebound: Displacement Transmissibility

Since rebound damping ratios differed so greatly between maximum and minimum values, it was deemed unrealistic to take an average of the trace for initial comparison. However the effects of such high damping ratios on the transmissibility ratio are visualised on graph 44. As damping ratio increases above that which is critically damped the amplification zone of transmissibility is attenuated, though at high frequency ratios increased damping ratio has a negative impact.



Graph 44: Displacement Transmissibility against frequency ratio with larger damping ratios

The regressive nature of the rebound damping ratio profiles imply that that the 3D plots will be transitionally smoother between low and high velocities and transmissibility will largely be less than what is seen with compression.

Figure 16 displays the 3D transmissibility plot for the minimum low-speed rebound position paired with 15° cone and 240 N/mm spring. Collectively the rebound damping transmissibility is on a much narrower scale than compression due to the comparatively higher damping ratios and thus minimal transmissibility.

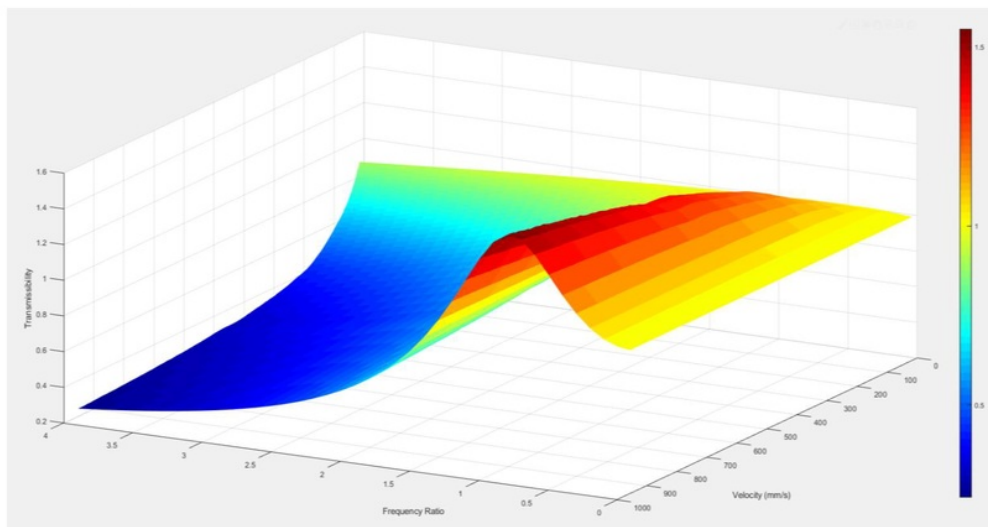


Figure 16: 3D graph of displacement transmissibility against velocity and frequency ratio for 15° cone and 240N/mm spring with minimum rebound

At low velocities and low frequencies the transmissibility is nominal and steadily increases with velocity. This increase isn't nearly as extensive as seen with compression, whereby figure 16 shows a maximum displacement transmissibility ratio of only 1.5. It is also noted that the transition of transmissibility to high velocity and frequency is much more gradual with rebound characteristics. 3D plots for compression showed very little change in transmissibility after a certain frequency ratio, whereas the rebound displays significant changes throughout suspension behaviour, though on a comparative scale of transmissibility rebound changes the least.

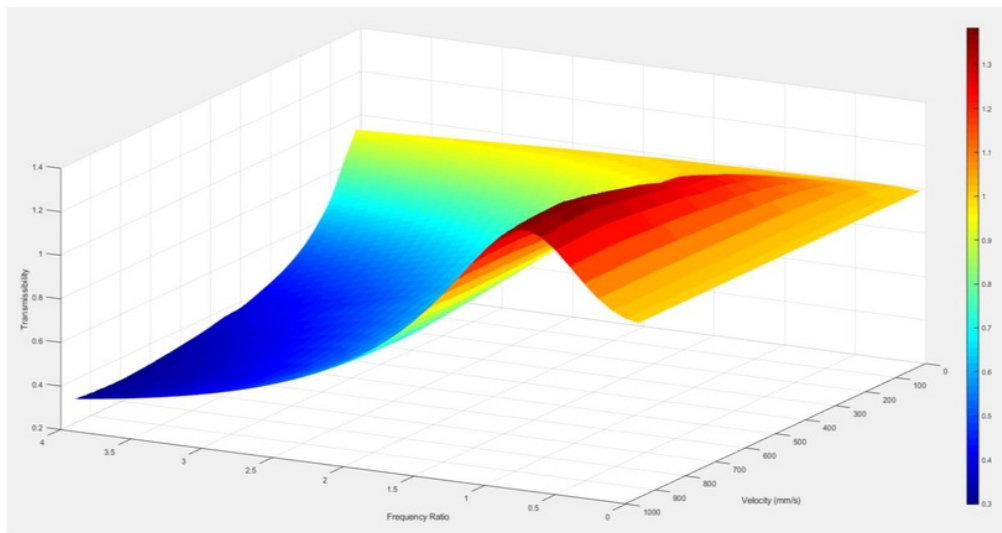


Figure 17: 3D graph of displacement transmissibility against velocity and frequency ratio for 15° cone and 240N/mm spring with maximum rebound

As seen on figure 17 which displays the 15° cone with maximum low-speed rebound damping, there are only minor differences between this plot and the previous at minimum low-speed position. Other than a reduction in maximum transmissibility to around 1.3, the differences are extremely subtle.

The 18° cone with 25 N/mm spring however shows more noticeable changes seen on figure 18. At minimum rebound setting the 3D plot shows greater transmissibility at low frequencies, peaking at 1.6. Low-velocity properties are somewhat similar, but a slightly higher gradient means as velocity increases with frequency remaining the same transmissibility increases more rapidly with the softer valving. Beneficially, transmissibility at high velocity and frequency is lower and thus oscillations in these conditions won't be transferred to the rest of the system as much as previous.

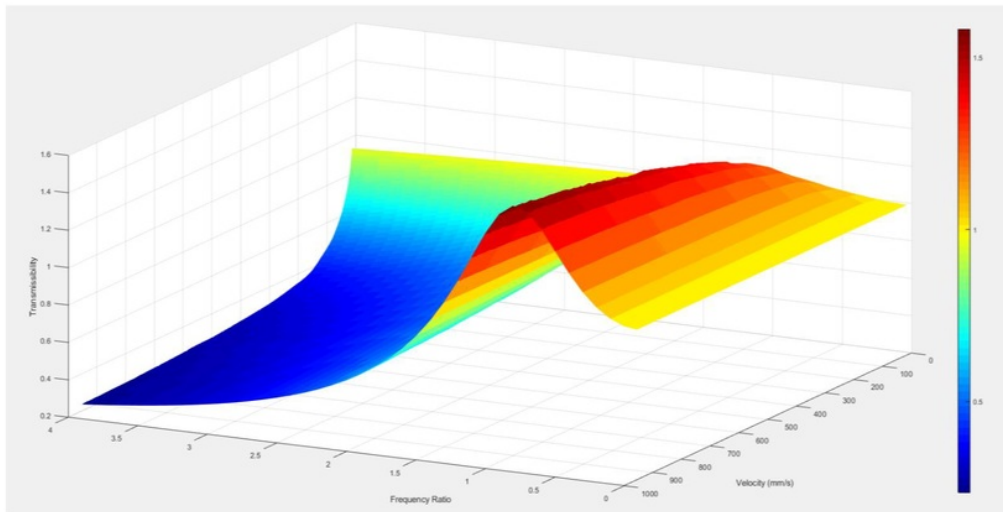


Figure 18: 3D graph of displacement transmissibility against velocity and frequency ratio for 18° cone and 25 N/mm spring with minimum rebound

The standard WP forks from the Husqvarna TE250i represented transmissibility characteristics shown in figure 19. Again the increase in transmissibility ratio at low frequencies is much higher. Though the low damping ratios at high frequency are beneficial, the higher transmissibility at low velocities negatively impact on the ride comfort. It could be said that rebound damping is much more important than compression, as without sufficient control through damping the wheel loses contact with the road beneath and grip is lost.

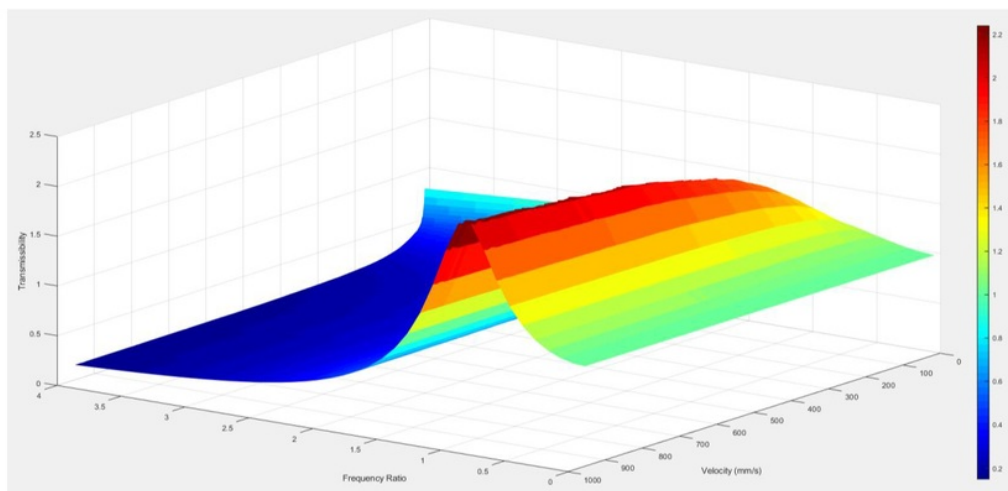


Figure 19: 3D graph of displacement transmissibility against velocity and frequency ratio for standard WP forks with minimum rebound

Evidently from the displacement transmissibility analysis, the minimal differences in high frequency damping for both compression and rebound emphasise the importance of low-speed characteristics. It would therefore suggest that the ideal valve combination is that which has a lesser magnitude of low-frequency low-velocity transmissibility: the 15° cone and 240 N/mm spring. However this is primarily in terms of displacement transmissibility.

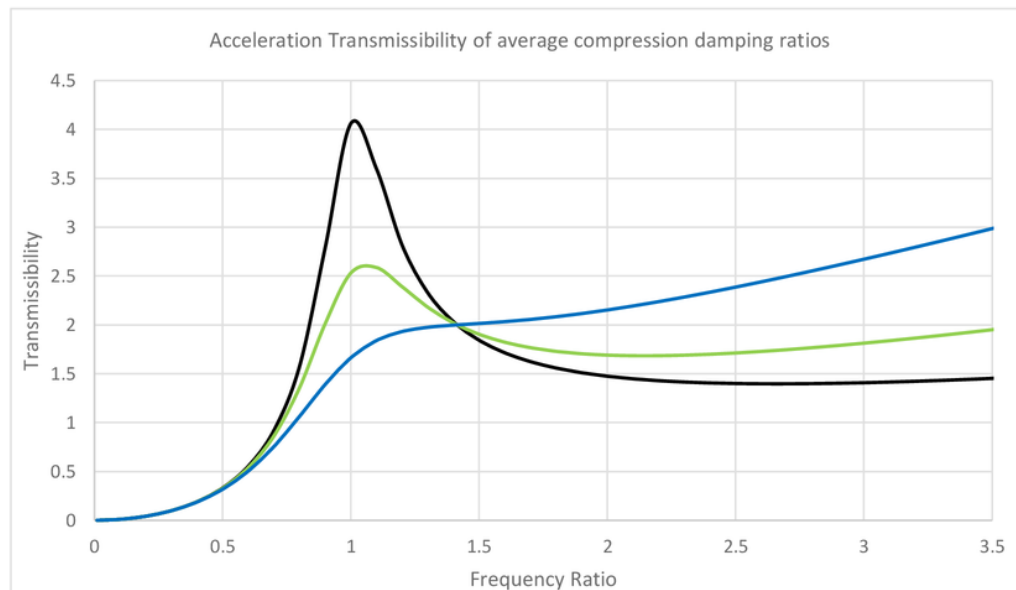
6.2.3 Acceleration Transmissibility

Considering ride comfort, the forces experienced by the rider are proportional to the vertical accelerations of the sprung mass such that it is ideal for accelerations to be minimised to provide the sensation of comfort. Acceleration transmissibility is given by the equation:

$$T_a = r^2 \sqrt{\frac{1 + (2\zeta r)^2}{(1 - r^2)^2 + (2\zeta r)^2}} \quad [60]$$

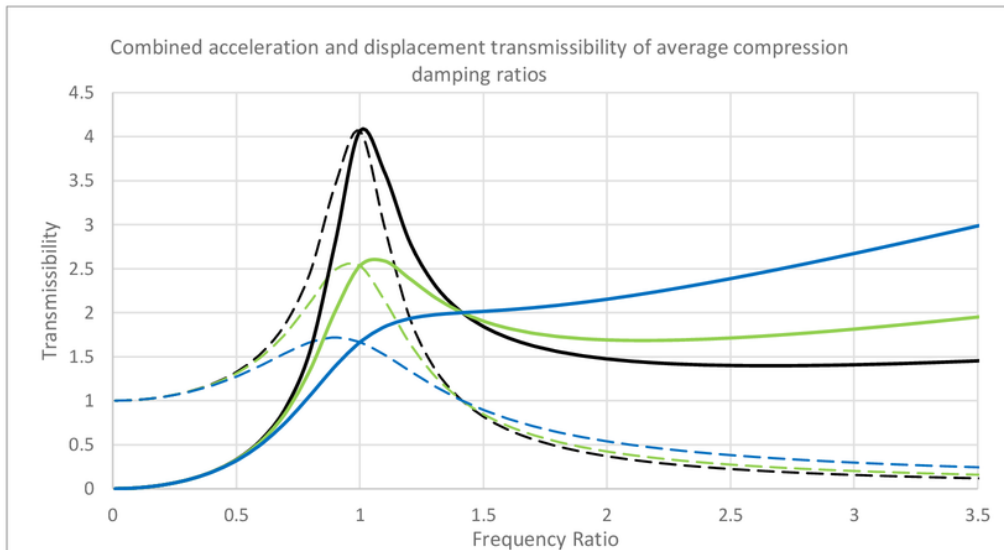
Thus, using the average compression damping ratios of table 10 the characteristics of transmissibility in terms of acceleration can be seen.

Valve Combination		Average ζ	Graph Colour
15° Cone	240 N/mm spring	0.376	Blue
18° Cone	25 N/mm Spring	0.215	Green
Standard WP Forks		0.127	Black



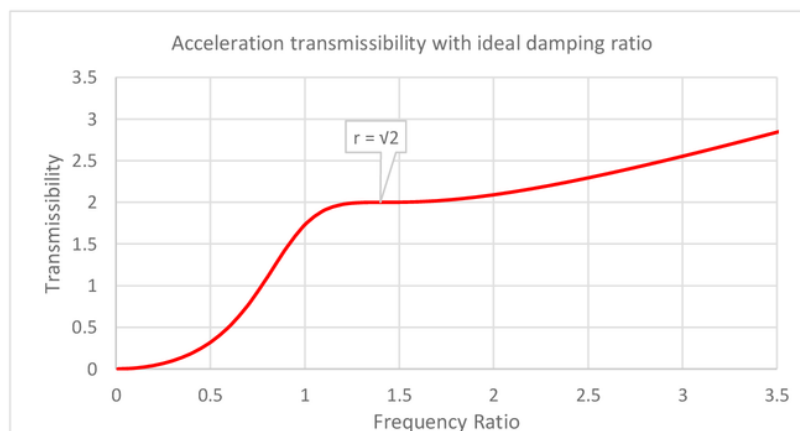
Graph 45: Acceleration transmissibility of average compression damping ratios

Seen on graph 45, regardless of damping ratio, at a frequency ratio equal to $\sqrt{2}$ the acceleration transmissibility takes a value of 2. Frequencies below this exhibit increased amplifications in conjunction with the standard WP forks and 18° cone 25 N/mm spring combination. Higher transfer of acceleration negatively affects perceived ride comfort and therefore the standard WP system paired with the higher displacement transmissibility would provide the least comfortable ride.



Graph 47: Combined acceleration and displacement (dashed) transmissibility of average compression damping ratios

(Cossalter, 2006) proposed that the ideal damping ratio is that which at a frequency ratio of $\sqrt{2}$ has no gradient, and occurs at a value of $\zeta = 0.354$ and shown on graph . At this value acceleration transmissibility is minimised in the area of maximised displacement transmissibility. As a result ride comfort at low frequency oscillations is optimal. It is therefore idealised that the cone valve combination which has a low speed average damping ratio closest to that of the optimal value put forward by Cossalter would achieve the best theoretical ride comfort.



Graph 46: Acceleration transmissibility with ideal damping ratio

Referring back to the damping ratio profiles of graph 40, the low-velocity damping ratios for each combination were averaged between 0 – 250mm/s to approximate a mean ratio at low speed.

Cone °	Spring Rate N/mm	Average ζ
11	25	0.360
	100	0.432
	240	0.523
15	100	0.346
	240	0.501
18	25	0.356
	240	0.468
WP STD	n/a	0.216

Table 11: Average low-speed compression damping ratios

The 18° cone and 25 spring combination was found to have low speed damping ratios on average almost identical to the theoretical value proposed by Cossalter with a zeta value of 0.356. This combined with the lower damping ratios at high speed mean it is potentially the optimal theoretical valving combination for the cone valve forks. Analysis of the 3D acceleration transmissibility plot given by figure 20 shows that at low velocities and low frequency ratios the transmissibility characteristics of acceleration are relatively flat, indicative of the ideal damping ratio.

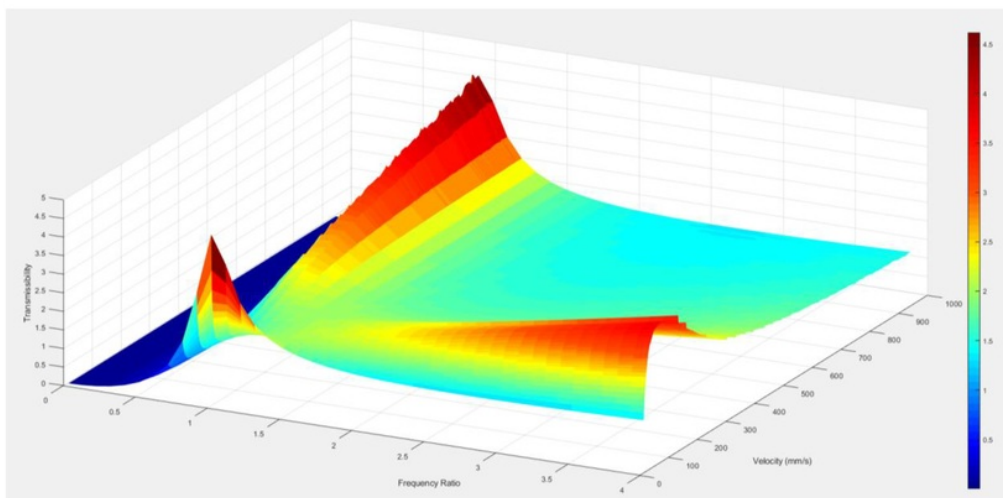


Figure 20: 3D graph of compression acceleration transmissibility against velocity and frequency ratio for 18° cone and 25N/mm spring

In this area the system will provide little excessive acceleration to the sprung mass whilst remaining responsive. Much of the rest of the 3D graph shows desirable characteristics whereby the majority of the high velocity and high frequency area exhibits similar transmissibility values and therefore a smooth response to excitations. The main area of concern would be that of high velocity and low frequency oscillations which would occur over a larger displacement. This area shows quite large amplifications of acceleration and therefore an undesirable feeling for the rider.

Figure 21 shows the acceleration transmissibility plot for the harder valve combination of 15 cone and 240 N/mm spring. Contrary to the previous graph it is noticed that the high velocity and low frequency area shows much improved characteristics of acceleration transmissibility, where amplifications are minimised due to the higher damping ratios. However this comes as a detriment to the low speed low frequency area whereby although accelerations are not emphasised the slow reaction of the system may result in a sluggish feeling to the rider. Also the remaining area of the graph at mid to high velocities and frequencies shows higher values of transmissibility than the previous valving resulting in potentially harsh ride characteristics.

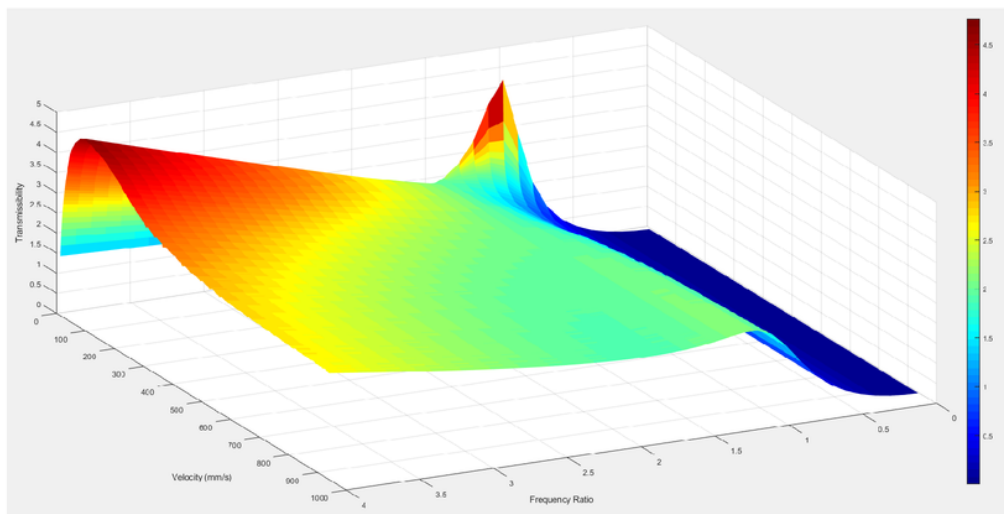


Figure 21: 3D graph of compression acceleration transmissibility against velocity and frequency ratio for 15° cone and 240N/mm spring

It is therefore suggested that the 18° cone and 25 N/mm spring offers an ideal potential baseline setting providing theoretical comfortable ride characteristics across the majority of frequencies and velocities.

7. Conclusion

The summation of this report outlines the complexities surrounding suspension damping design and tuning such that the inherent behaviour of a viscous damper differs with velocity, whereas the transmissibility characteristics of the system are dependent on oscillation frequency. Resultantly this enabled a holistic view of damping properties and subsequently how frequencies are transferred from road undulations to the sprung mass and the effect this has on the perceived comfort felt by the rider.

During practical analysis where effects of changing cone angle and spring rate were explored, it was found that the damping system as a whole is far more complex than first thought. The intricate subtleties in the design can initially be missed, such as that of the low-speed compression adjuster. With the bleed-hole cone adapter it was seen that any changes in position of the low-speed compression needle valve had a negligible affect on both average damping force-velocity graphs and full frequency graphs. When the adapter was swapped out for one without a bleed hole the same lack of change in the average graph was seen, however what wasn't considered was the effects on damping during acceleration and deceleration phases which are seen on the full frequency graphs. Increasing the compression damping needle position greatedened the hysteresis between the acceleration at the start of the stroke and deceleration towards maximum compression, hence why no change in average damping was seen. As a result the comparability of certain tests was compromised, though the effects of cone angle and spring rate could be discerned using the results of other tests.

It was shown in damper dynamometer data that steepening the angle of the cone valve causes a rise in gradient of the compression damping force as suspension velocity increases. Rebound damping remains generally unaffected by the change in cone angle, though some differences are noted with the change in spring rate. Due to the free-play between the cone adapter and valve, a small flow of oil is allowed through the system on the rebound stroke. A softer spring allows for this flow to seep through and compress the spring more than that with a stiffer spring, thus rebound damping is decreased.

However the primary function of changing spring rate is to alter compression damping characteristics. As with the cone angle, increasing spring stiffness alters the gradient of the force-velocity curve such that it takes on more linear characteristics. Though increased low-speed damping ratios are preferable for minimised displacement transmissibility, the higher damping ratios at high speeds have a negative effect whereby accelerations are amplified through the sprung mass.

Displacement and acceleration transmissibility analysis allowed a comprehensive study of how damping behaviour effects the perceived ride comfort, and using the ideal damping ratio a cone valve combination was selected that would theoretically provide the best feeling for the rider. It remains

theoretical as suspension set-up preferences are a subjective matter and are dependent on the rider's personal style. However the proposed 18 cone and 25 N/mm combination can be used as a baseline setting that can be improved upon according to rider feedback.

It is of no doubt that the WP Cone Valve forks are preferable in comparison to the standard WP forks. As has been explored the impact that such lower damping ratios has on the displacement and acceleration transmissibility means that these motions are amplified at low-frequency inputs and has a negative impact on the overall ride quality. Whereas the changeability of the various cone valve combinations allows the system to be personally tuned to meet a range of damping desires.

Thus the summation of this report offers a holistic insight into the mathematical principles of suspension reactions as well as a practical understanding of how the WP Cone Valve forks have been designed to control damping.

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