Inside TTX

The Öhlins TTX40 manual

HVM (nee Herdez Competition) *was one of the first Champ car teams to test the TTX.*

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Figure 1.1 *A complete TTX damper with spring*

1 Introduction

Congratulations on choosing the Öhlins TTX damper, the latest generation of twin tube dampers from Öhlins.

Get to know your TTX, and be sure to read this manual thoroughly before using the dampers. We recommend that you keep this manual handy.

The TTX damper is the culmination of three decades of Öhlins' successful participation in world championship events winning more than 100 World Championships. Many years of work together with some of the world's most successful racing teams together with advanced dynamic analysis methods developed at Öhlins Racing headquarter in Sweden has given Öhlins the unique knowledge needed to design the TTX damper.

The Öhlins TTX damper, originally developed for formula racing, is designed to handle the demanding damping characteristics needed for all types of tracks, from street courses to super speedways.

The TTX damper is fully adjustable with maximised damper response together with qualities you've never seen before when it comes to "settings".

Low and high speed compression and rebound damping are externally adjustable and fully independent. The adjustment range is huge with equal increments of force throughout the adjustment range. Even the shape of the damping curve can easily be changed. All adjusters affect the flow from the main piston, not the piston rod displacement volume.

The compression damping forces of the TTX damper are not, as in a conventional damper, caused by a pressure drop on the rebound side, but by increased pressure on the compression side. This reduces the risk of cavitation and makes any reservoir valve or high gas pressure unnecessary. So, no balancing of reservoir damping to main piston damping is needed to avoid cavitation and improve damping response. Maximum response and minimum risk of cavitation will always occur. With no reservoir valve, the internal pressure of the damper unit will be kept to a minimum. The low amount of hysteresis results

Figure 1.2 *A TTX damper unit.*

in excellent short stroke/high force performance. Also, a very low gas pressure can be used without any loss of damping performance.

Along with the damper comes a unique Valving Reference Program (available for download free of charge at www.ohlins.com). This computer model of the damper will allow you to find damping curves without a dynamometer. It will reduce building time tremendously and allow exact damper adjustments in pit lane. The TTX product will revolutionise the work for mechanics and engineers in the racing business.

This manual text is based on TTX dampers starting with Öhlins part number TTX NE0. These are through rod type dampers loaded with several new concepts. As always, all dampers are tested before they are delivered to the customer. In keeping with Öhlins long tradition of perfection, quality is outstanding and long life is to be expected.

Welcome to the World of Öhlins.

2 Design Criteria

After the Öhlins TT44 was introduced to the market in 1996, it very quickly became one of the most popular dampers in formula racing. For some period, more than 95% of the cars in The Champ Car World Series were using TT44 dampers.

There are several reasons why the TT44 became so popular. One reason is that it came with some new features not available on other dampers. One of them is the powerful low speed adjusters, totally independent and with the compression adjuster restricting the oil flow from the main piston, not only from the piston rod displacement. Another is the compression high speed adjuster, giving new possibilities to reshape the compression curve.

When designing the new TTX, the goal was to come up with a damper which would be just as big a step forward as the TT44 had been. Highest priority should be not only to design a damper with excellent performance, but also a damper easier to work on and use than any other product available.

During the development several patent applications were made.

Six of the most important design criteria for the TTX are listed below *(Figure 2.1)*.

1. No reservoir valve

The damper design should be with no reservoir valve. In dampers where reservoir valves have to be used to avoid cavitation, one more parameter has to be optimised – the right amount of reservoir damping.

The hysteresis will be minimised, as no reservoir valve has to be used. All damping comes from the pressure drop over the main piston. Damping forces from a reservoir valve always causes more delay in the damping force build up. See chapter Hysteresis for more information.

Using reservoir valves always increases the internal pressure. The friction from the piston rod seal/seals can be kept low because of the low internal pressures.

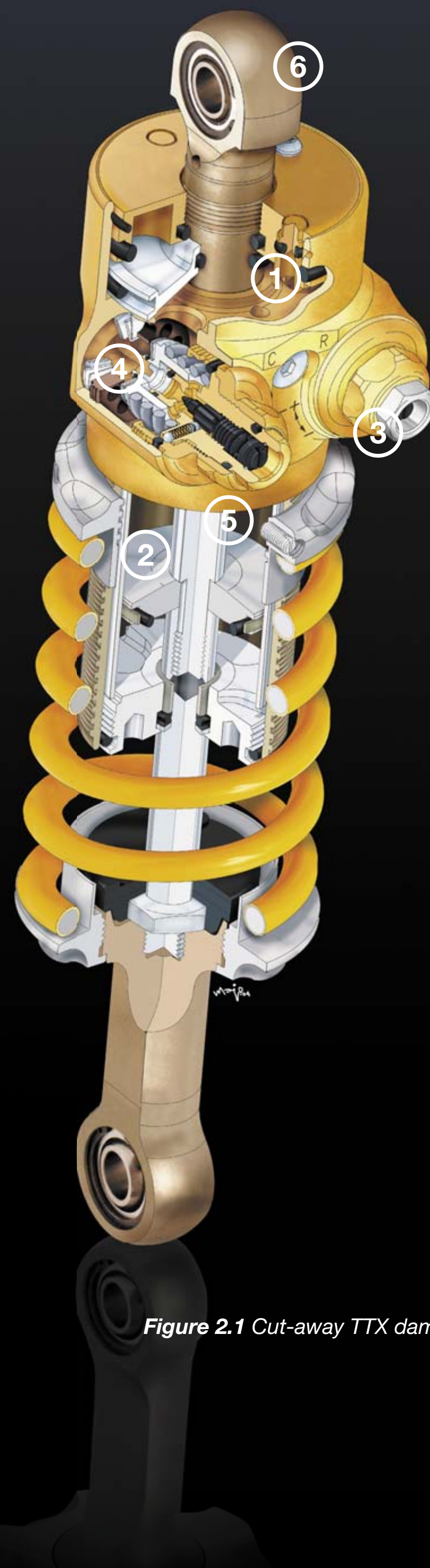


Figure 2.1 *Cut-away TTX damper.*

2. Main piston flow

Another criterion was to have all the adjusters regulate the flow from the main piston. This will give the maximum pressure area and because of this, the maximum oil volume to regulate.

The larger the pressure area is, the lower the internal pressure will be for a given damping force. The lower the internal pressure, the less flex there will be. The flex is caused by expansion/compression of the damper body and compression/expansion of the oil. The result is excellent short stroke/high force performance.

With a large volume of oil passing through the valves , it becomes easier to control the restriction of the oil. In other words, the matching of dampers will be improved.

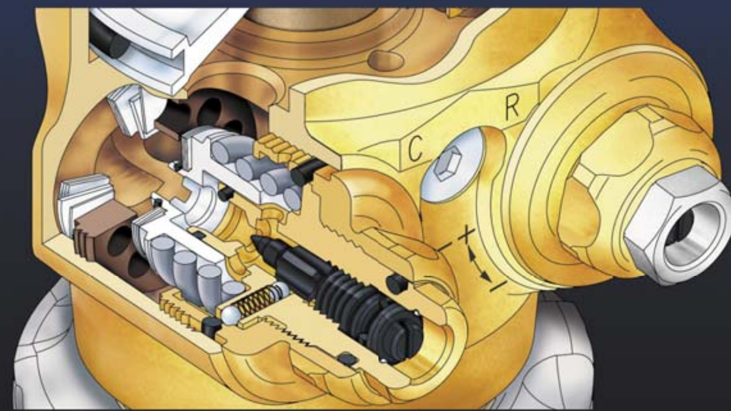


Figure 2.2 *High speed Compression and Rebound adjusters.*

3. Full adjustability

On the TT44/TT40, it was never possible to use a high speed rebound adjuster in combination with a high speed compression adjuster. On the TTX, we wanted to be able to combine those two while keeping them completely independent from each other, as with the low speed compression and rebound adjusters.

Poppet valves preloaded by coil springs were picked to become the high speed valves, as they can be made very compact in size and precise in opening pressure. This type of valve very often gives an abrupt opening characteristic, resulting in a sharp ”knee” in the damping curve. To make the ”knee” more rounded and to be able to change its shape, some shims are added to the face of the poppet valves. By changing these shims, the shape of the ”knee” can be affected.

4. Simple valve changing

Even if the adjustment range of the external adjusters is huge, sometimes there might still be a need to change the valving of the dampers. In other words, change one or several of the following parts: poppet valve/valve seat, coil springs and nose shims. As this very often is done at the track and has to be done quickly, this job has to be simplified as much as possible. Compared to reshimming a conventional damper, any of the changes in the TTX will be a lot quicker. The result exceeded our demands.

Also it should be possible to fill the damper without a vacuum-filling machine, as this otherwise would be a limiting factor.

5. Through rod damper

A through rod damper has some technical benefits. One is packaging, which is a main issue on formula cars. The reservoir volume can be very small, as there is no piston rod displacement. Here no external reservoir is needed. Also there is no gas force pushing the piston rod out of the damper body. (The word ”nose pressure” is sometimes used for this force.) Here the nose pressure is zero. This has several advantages. The nose pressure doesn’t vary due to temperature changes and you don’t have to fight the gas force when installing the damper on the car or in the dynamometer.

Designing a through rod damper gives the possibility to separate the rod bushings and keeps the distance between them constant. If coilover springs are used, the amount of friction will be tremendously reduced.

As the piston area for compression and rebound are identical, the damping forces will be the same if the same valving is used and the adjusters are set the same. To some degree, this simplifies the use of the damper.

For all the above reasons, race teams have been interested in through rod dampers. Also, when introducing the TTX, we wanted it to be something very different from the other products out on the market.

6. External clocking

Another strong side of the TT44/TT40 was the possibility to clock the reservoir bracket at any angle. This function we wanted to keep on the TTX damper, to ensure an optimum installation on any car. Just as on the TT44/TT40, the clocking of the adjusters on the TTX in relation to the top eye should be possible to change without opening the damper.

3 How the Damper Works

The description below is divided into compression and rebound damping cycle.

- Compression damping cycle.
- Rebound damping cycle.

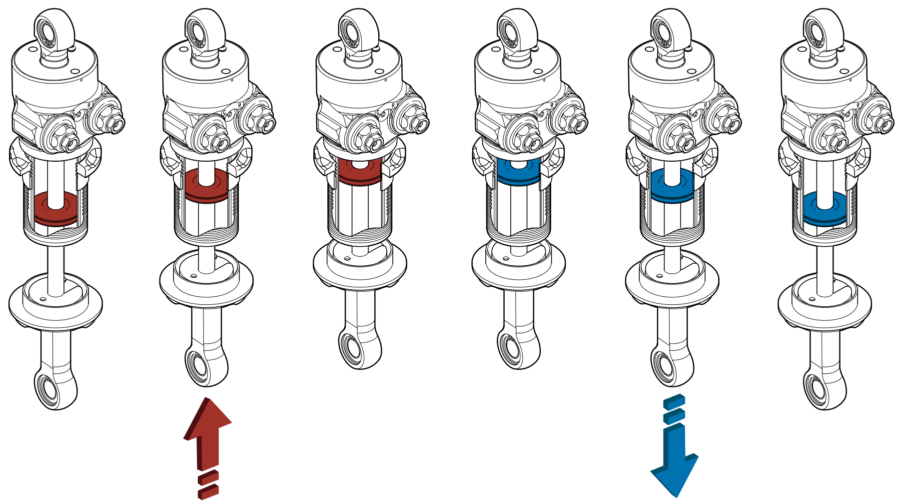


Figure 3.1 *Compression and Rebound damping cycle.*

General description

The compression damping cycle describes the situation when the rod and piston unit moves into the damper body shortening the length of the damper. While the rebound damping cycle describes the situation when the rod and piston unit moves out from the damper body extending the length of the damper.

The terminology ”compression side” of the piston here refers to the oil volume in front of the piston when the external piston rod is moving into the damper body (compression cycle). The ”rebound side ” of the piston refers to the oil volume in front of the piston when the external piston rod is moving out of the damper body (rebound cycle).

When the rod and piston unit doesn’t move, the internal pressure in the whole damper unit is equal with the set gas pressure. When track conditions cause the vehicle suspension to move, the damper piston will attempt to move through the damper oil. In order for the piston to move, oil must flow from one side of the main piston to the other. The restriction of the valves causes a pressure difference between the two sides of the piston, resulting in damping forces. In the TTX, this pressure difference comes from increased pressure on the forward side of the piston and not reduced pressure on the backside, as in conventional dampers.

Unless a different valve configuration is used compression to rebound, the compression and rebound valves are identical. On both sides there are three type of valves used for adjusting the damping characteristics.

- Bleed valve
- Shim valve
- Poppet valve

The compression bleed valve is in parallel with the compression poppet valve and the rebound bleed valve is in parallel with the rebound poppet valve. The poppet valves are pushed against their seats by preloaded coil springs. The preload is externally adjustable. The amount of preload of the poppet valves determines the pressure differentials across the main piston necessary to make the poppet valves open. For more information about the bleed valves and the poppet valves, see chapter *External adjusters.*

The shim valves are placed on the nose of the poppet valves. These shim stacks affect the opening characteristic of the poppet valves. The shim configuration can be changed to achieve different opening characteristics of the poppet valve. See chapter *Internal adjustments* for more information.

Also, there are two check valves installed in the damper, making the compression and rebound valves fully independent.

Paul Tracy, *Forsythe Championship Racing*.

Jason Bright, *Ford Performance Racing.*

Flow circuit at compression cycle

How the oil flows from the compression side to the rebound side of the piston will be described here. This is caused by increased pressure on the compression side of the main piston, while the pressure on the rebound side is almost constant at the set gas pressure.

1. The oil will reach the compression valves by passing through the port of the separating plate *(Figure 3.3-A)* extending into the cylinder head and leading the oil into a chamber below the compression valves *(Figure 3.3-B)*. Because of the small restriction of this port, the pressure in this camber will be very much the same as the compression side of the cylinder tube. The piston velocity and how the valves are set determine the pressure in the camber. The pressure will help to close the check valve in this camber.

2. Depending on the pressure, different things will occur. As the velocity increases, the pressure will rise.

a) In the initial part of a compression stroke, when the velocity of the piston is low, the oil will pass through the adjustable low speed compression valve. In this bleed valve, the restriction takes place in the passage *(Figure 3.3-C)* between the needle seat (integrated to the needle housing) and the needle. As long as the piston is moving and the bleed valve is not fully closed, some oil will always flow through the bleed valve. If the bleed valve is fully closed, this passage will be blocked.

b) As the velocity increases, the shim stack on the nose of the poppet valve will start to open and oil can pass between the shim stack and the poppet valve seat *(Figure 3.2-D and 3.3-D*. The stack configuration will decide the opening pressure. An increased stiffness of the stack will raise the opening pressure and thus raise the damping force. The shape of the nose on the poppet valve gives the shims freedom to bend and lift from the seat, no matter how much preload from spring there is on the poppet valve. This will allow the shim stack to always open gradually and therefore a small amount of oil will pass through the shim stack even with a very low pressure drop over the piston.

c) As the piston velocity increases further, the internal pressure rises. At a certain velocity the movement of the piston creates a pressure difference across the main piston that is equal to the predetermined pressure required to open the poppet valve. The oil is now free to flow between the poppet valve and the seat *(Figure 3.2-E and Figure 3.3-E)*. Due to the oil flow, the nose shims will follow the poppet valve up from the seat.

NOTE:
In practice, the piston often does not reach a velocity high enough to cause a sufficient pressure drop and open the poppet valve.

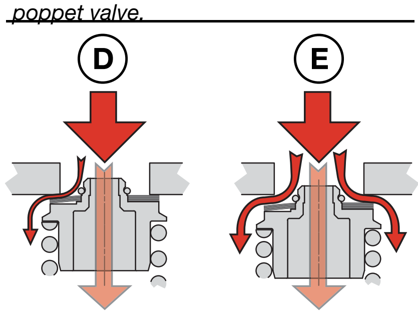


Figure 3.2 *Poppet valve*

By using a very stiff shim stack in combination with little preload on the poppet valve, the oil flow through the shim stack will be very limited before the poppet valve opens. This will make the opening of the poppet valve more abrupt and the shim stack will open at a higher velocity. This will change the characteristics of the damping curve.

NOTE:
The opening characteristic of the poppet valve is always abrupt, unlike the gradual opening characteristic of the shim stack.

3. The oil has now reached the low-pressure zone at the gas reservoir *(Figure 3.3-F)*. This volume is in direct contact with the separating piston, separating the oil from the nitrogen gas. Here the pressure is always equal to the set gas pressure.
   As the TTX is a through rod damper, there will be no fluid displacement by the piston rod. However, a gas volume is still needed to reduce changes of the static internal pressure due to volume changes caused by temperature variations.
   The rising temperature of the damper will increase the volume of the oil. Also the damper body will expand as the temperature increases, but not all to the same extent.

4. Now the oil will flow through the compression check valve *(Figure 3.3-G)* positioned at the rebound valves. However, as long as the low speed rebound bleed valve isn't fully closed, some oil will flow the through this valve backwards (*Figure 3.3-H)*.

NOTE:
The compression check valve is placed together with the rebound valves.

5. From here the oil flows between the two tubes *(Figure 3.3-I)*. The oil re-enters the main tube on the rebound side through ports placed between the end cap and the inner tube (Figure 3.3-J). The compression flow circuit is completed.

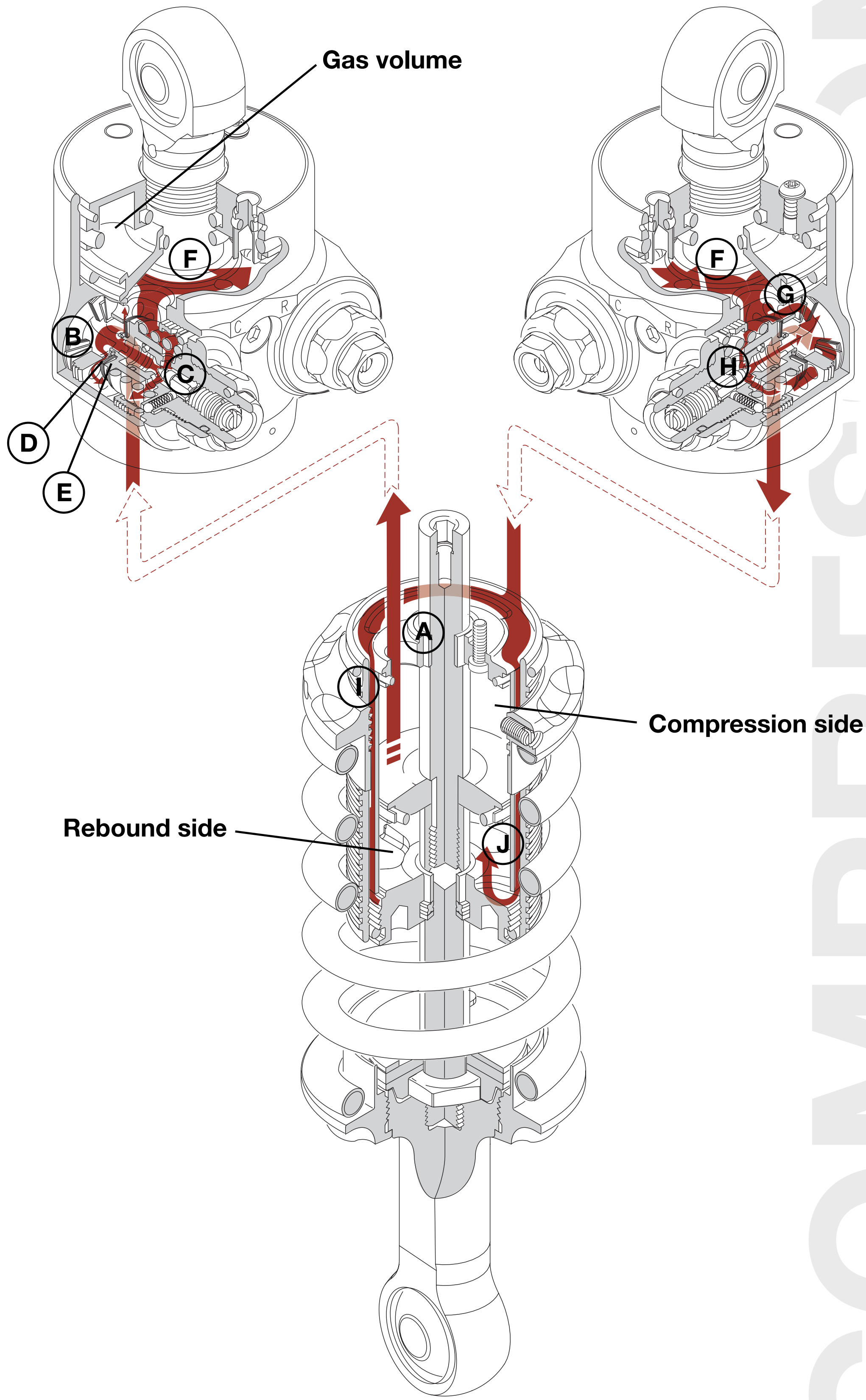


Figure 3.3 *Flow cicuit during compression cycle.*

Flow circuit at rebound cycle

Below is a description of how the oil flows from the rebound side to the compression side of the piston. The rebound cycle is very similar to the compression cycle, but the flow will be in the opposite direction and the oil will move through other valves. During the rebound stroke, the pressure of the rebound side of the main piston is increased, while the pressure of the compression side is kept almost constant.

1. First the oil has to get to the rebound valves. The ports between the end cap and the inner tube *(Figure 3.5-A)* will lead the oil to the volume between the tubes *(Figure 3.5-B)* from where the oil will reach the chamber below the rebound valves *(Figure 3.5-C)*. The pressure here will be roughly the same as in the rebound side of the cylinder tube due to small restrictions of the oil flow. The pressure will help to close the check valve in this camber.

2. See ”section 2” above in chapter *Flow circuit at compression cycle* for more detailed information as the rebound valves are identical to the compression valves.

a) Unless the low speed rebound valve is fully closed, the oil will first pass through this valve *(Figure 3.5-D)*.

b) The second valve to open is normally the nose shim stack *(Figure 3.4-E and Figure 3.5-E)*.

c) If the pressure level reaches the opening pressure of the poppet valve, the poppet valve will open *(Figure 3.4-F and Figure 3.5-F)*.

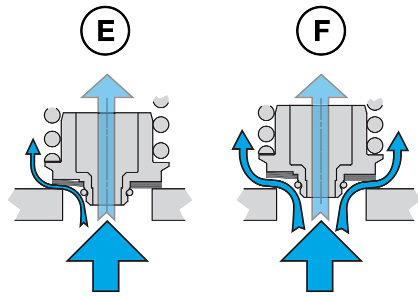


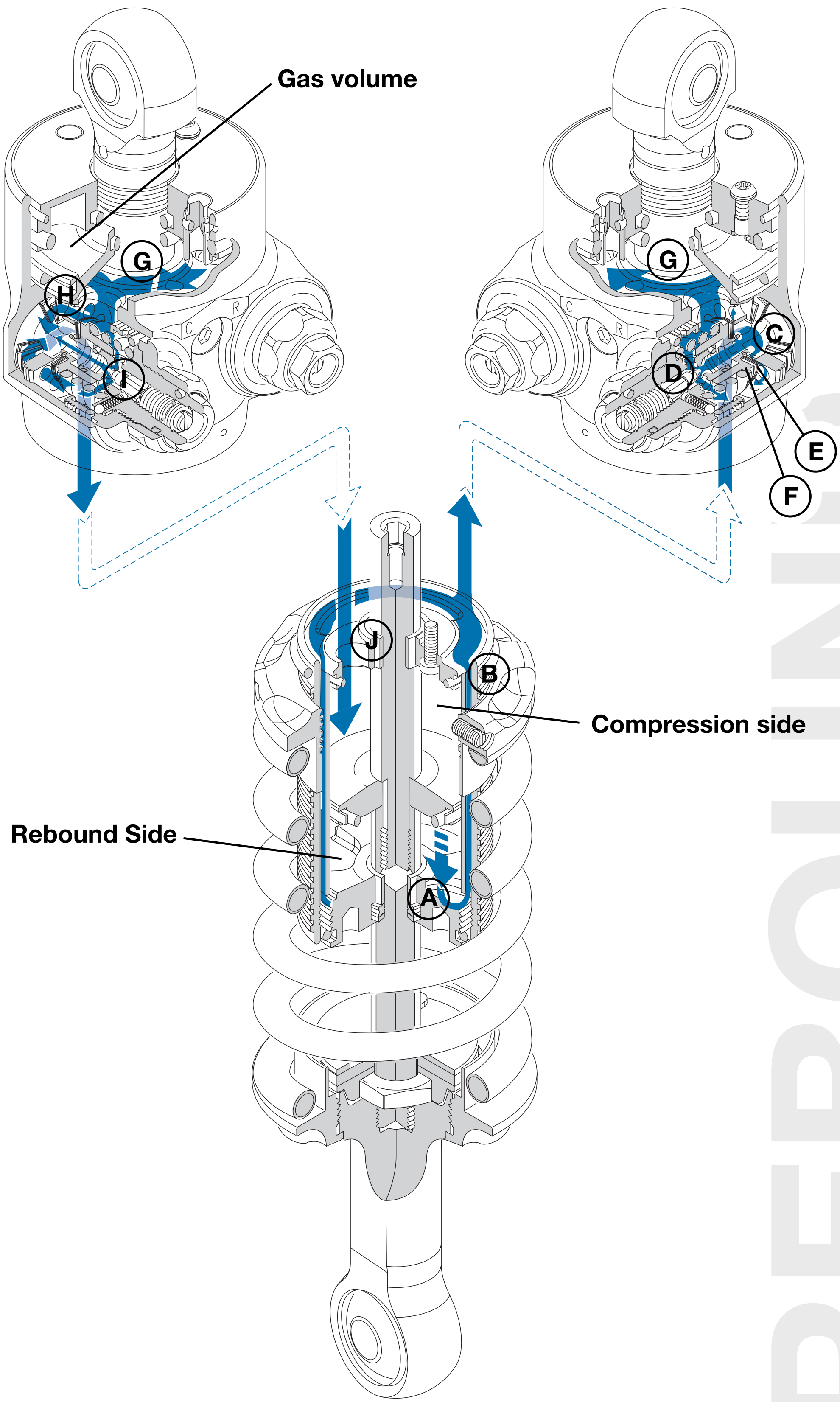
Figure 3.4 *Poppet valve*

3. Now the oil has reached the low-pressure zone at the gas reservoir *(Figure 3.5-G)*, where the pressure is equal to the gas pressure.

4. The oil will now flow through the rebound check valve *(Figure 3.5-H)* positioned at the compression valves. Some oil can, in the same way as described above in *Flow circuit at compression cycle*, flow backwards through the low speed compression valve *(Figure 3.5-I)* unless it is set to the fully closed position.

NOTE:
The compression check valve is located together with the rebound valves.

5. Finally the oil re-enters the main tube on the compression side through a port in the separating plate *(Figure 3.5-J)*. The rebound circuit is completed.



REBOUND

Figure 3.5 *Flow cicuit during rebound cycle.*

Figure 4.0 *TTX cylinder head*

4 Hardware Description

The TTX damper uses some concepts not used by any other damper manufacturer. In this chapter there will be information about some of this unique hardware. For information about the valves, see External adjusters and internal adjustments. For information regarding assembly or disassembly, see chapter *Work section*.

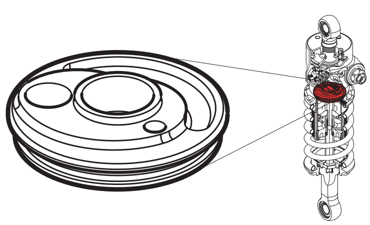


Figure 4.1 *Separating plate.*

Separating plate

Separating plates have been used by Öhlins since 1992 with development teams. The first product with separating plate available for the public was the TT44 introduced on the market in 1997.

The separating plate separates compression side from rebound side and guides the oil flow to and from the valves.

Thanks to the separating plate, the machining of the cylinder head will be simplified and the damper length minimised. Sometimes the desired design would not even be possible to do without a separating plate.

Unlike the separating plate of the TT44/TT40 there are no check valves installed in the separating plate of the TTX.

No seal has to be used between the plate and the cylinder head. Flat and smooth surfaces and large clamping force will ensure negligible leakage.

The mounting of the separating plate of the TTX is simpler than ever. The cylinder head will centre the plate and a screw will guarantee the correct mounting angle. In the TTX damper a bushing is installed in the separating plate, making the installation of support ring and x-ring in the cylinder head very simple.

The o-ring between the plate and the inner tube is mainly to ensure that the inner tube doesn't come loose when removing the end cap.

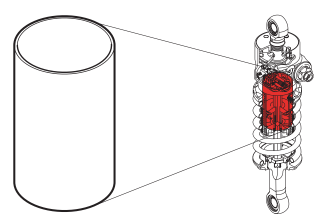


Figure 4.2 *Inner tube.*

Inner tube

Just as on the TT44, a steel inner tube is used to ensure maximum stiffness, but to make the assembly of the damper easier, the inner tube is now made symmetrical. Therefore the tube can be installed with either end first.

As the end cap bottoms against the inner tube, the inner tube will push the separating plate against the cylinder head. In so doing the inner tube does not only serve as a wall between fluid flows, but also ensures clamping the separating plate to the cylinder head.

NOTE:
The steel inner tube has to be protected from corrosion when not installed.

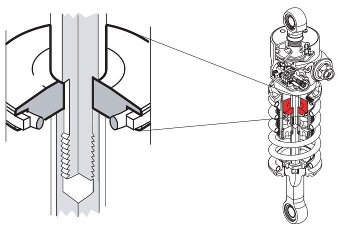


Figure 4.3 *Piston installation.*

Piston installation

The TTX damper uses a solid symmetrical main piston without shims. Instead of being the main path for fluid to flow to the opposite side of the piston, it acts more as a plunger, pushing fluid through the valves in the cylinder head. The piston band in this case, is not a load bearing seal (as described below in the *"Piston rod guide/seal"* section.) and contributes with very little friction to the damper.

The internal and external piston rods are screwed together clamping the piston, so here the "displacement rod" (internal) also serves as a lock nut – two functions in one. The long, thin "screw" works excellent as a screw element as it is very elastic.

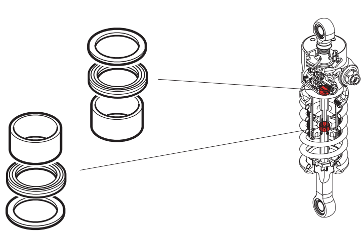


Figure 4.4 *Piston guide/seal*

Piston rod guide/seal

The two piston rod bushings are static bushings placed at the largest possible distance from each other to minimise friction and give maximum stiffness – one is, as in most other dampers, placed in the end cap, the other is placed in the separating plate. No piston bushing is needed, or even desired, because the entire side load is taken by the rod bushing and not the piston. Instead, a piston seal that can handle some misalignment between piston and piston rod is used.

Two identical piston rod seals of x-ring type are used, one for the external piston rod and one for the internal (displacement) piston rod. Due to the low gas pressure that is used, the level of friction can be kept low.

When using the damper as a coil over unit most of the friction in the damper comes from side load, caused by the torque from the springs acting on the bushings. By separating the bushings in the TTX the effect of this side load can be reduced significantly.

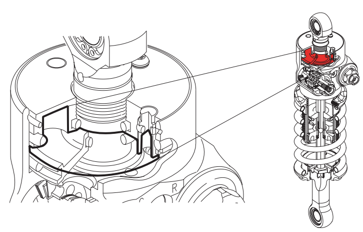


Figure 4.5 *Gas reservoir.*

Gas reservoir

By using the volume outside the area needed for the internal rod on top of the valves for the gas volume, a very compact installation can be made. Because the separating piston does not move along with the rod, friction from the separating piston is not important to the damper function. Therefore, the use of both internal and external seals is not a problem here.

Due to the fact that the TTX is a through rod damper, the eye to eye length can become the limiting factor when fitting the damper to a car. Therefore Öhlins has put a lot of effort in making the design extremely compact. In all types of pressurised dampers, there has to be some volume of fluid directly below the separating piston, making it possible to move some distance before it bottoms. This gives the separating piston margin from bottoming due to oil flex in the damper when pressurising the reservoir, oil leakage or temperature drop. As the gas reservoir of the TTX is placed on roughly the same diameter as the valves, it has been possible to design the damper so the oil volume between the separating piston and the cylinder head can be used to transport oil between the two valves

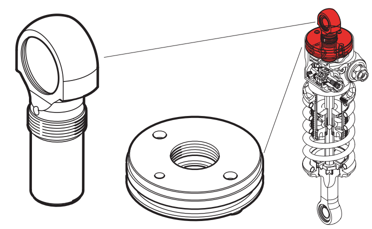


Figure 4.6 *Reservoir end cap and top eye.*

Reservoir end cap and top eye

The reservoir end cap has the function of preventing the pressurised nitrogen gas from leaking out and as a lock for the top eye which threads into the centre of the cap. The top eye can be positioned in any angle. See section *Reclocking top eye* in *Work section*.

The top eye has an open bore in the centre. This is the cavity that the displacement rod (internal) enters as it leaves the main body through the separating plate. The cross holes in the top eye, just below the spherical bearing, keep this cavity at atmospheric pressure.

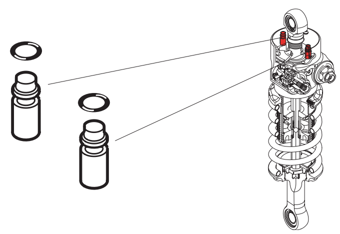


Figure 4.7 Pressure indicators.

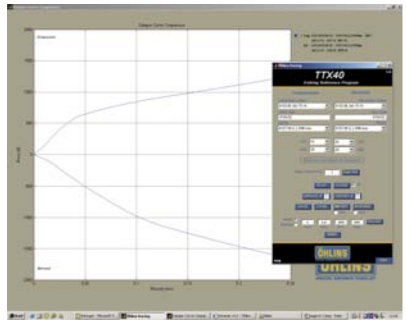


Figure 4.8 VRP user interface.

Pressure indicators

On the reservoir end cap there are two buttons, we call these “pressure indicators”. They have several uses. One is just as the name says to indicate if the damper is pressurised or not. In a through rod damper, as mentioned earlier, there is no force from the internal pressure pushing the piston rod out of the damper body. Therefore, it can be hard to tell if a through rod damper is pressurised or not.

By pushing in one or both pressure indicators, you are able to tell if the damper is pressurised or not by observing if they return to their static position. If the reservoir isn’t pressurised, the indicators will remain depressed. At 5 bar of internal pressure, more than 16 N (3.7 lbs) of force will be needed per indicator button to push them in.

The holes in the reservoir cap, where the pressure indicators can be seen are used to tighten/loosen the top eye. See section *Reclocking top eye* in *Work section* for information about reclocking the top eye.

Finally the pres sure indicators are used to position the separating piston when adding oil to the damper. Two special Öhlins tools are needed to do this. The tools use the two bosses on the top eye for positioning the separating piston. See section *Using Öhlins filling machine* in *Work section* for how to positioning the separating piston.

Valving Reference program

The use of Öhlins’ unique Valving Reference Program (VRP) for the TTX was obvious after the success of the previous VRPs for the TT44 and the DR4. This time the program is free of charge, to show the power of the TTX damper to as many as possible. The program is available on the Internet at www.ohlins.com

The TTX VRP is very easy to use. By positioning the cursor above the different ”buttons” in the program, information about how to operate the program will appear.

The damping forces used in this program have been measured on an Instron hydraulic dynamometer at constant velocity. For more information, see chapter *Damping force measurement*.

***Figure 5.0** A crank dynamometer from Roehrig.*

5 Damping Force Measurement

Damping forces are a frequent subject of discussions at race weekends. It is important to understand that depending on how the damping forces are measured, the force values can turn out very differently.

Within a race team, where the values normally always come from the same source this is normally not a problem. A team mainly needs a damper dynamometer to ensure their dampers produce the damping forces that are expected. This means that no “heavy duty” dynamometer is needed. For formula racing applications and many other types of asphalt racing a dynamometer that can reach 5 inch/second (0.127 m/s) is enough. The type of dynamometers mainly used are of the “crank type”. The price tag, size and simplicity are the strengths of these machines. A crank dynamometer can be used for “continuous measurement” or “peak velocity measurement”. A hydraulic dynamometer can be used for any type of measurement.

The requirements for a damper manufacturer, doing research and development, are very different and other types of machines are needed.

There is not one perfect way of measuring damping forces, as different situations ask for different needs.

See chapter *Damping curve terminology* for information about how to read damping graphs.

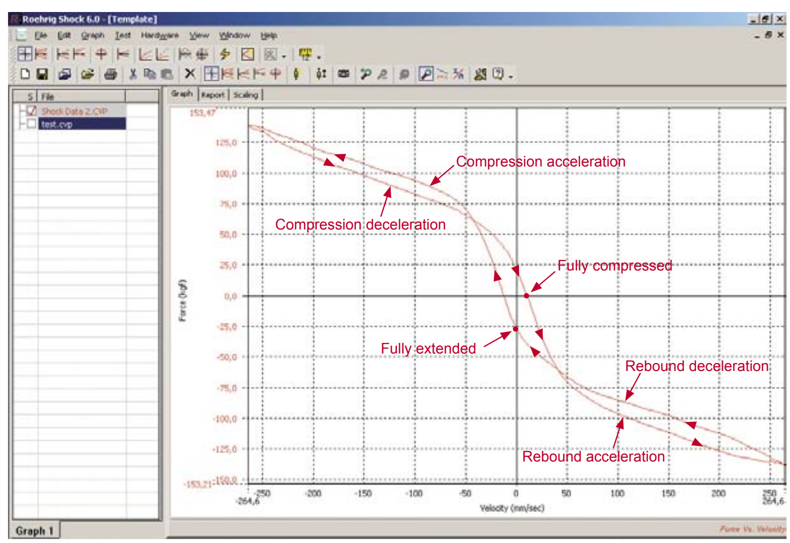


***Figure 5.1** Damping force measured continuously in a Roehrig dynamometer. The bleeds are fully closed.*

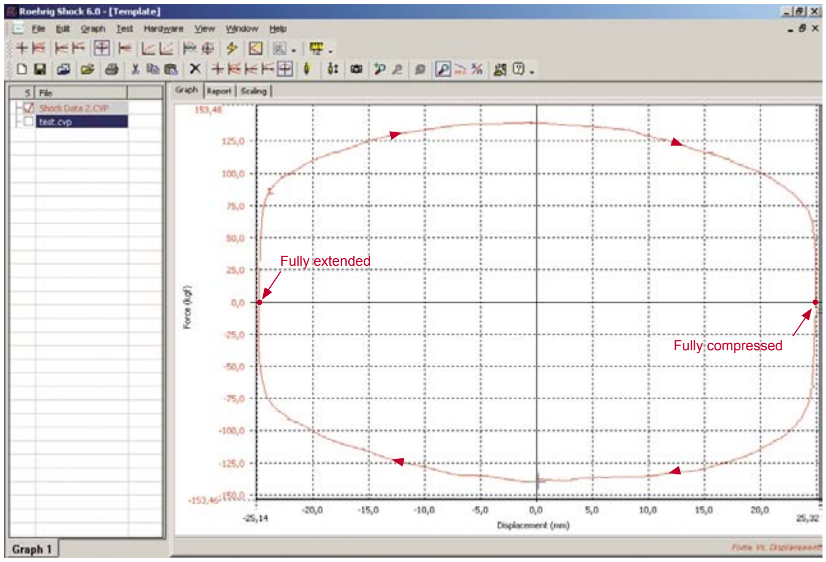


Figure 5.2 *Damping force measured continuously in a Roehrig dynamometer. Here the result is presented in a force-displacement graph. The data comes from the same run as in the previous figure.*

For a race team a continuous measurement of the damping force is very good: the measurement is quick, makes it very easy to tune the damping curve to a desired shape, hysteresis and cavitation is easy to detect and "dynamic problems" can be found. An example of a dynamic problem that can be detected this way is sticking check valves.

With continuous measurement of the damping force, the damper movement mimics a "sine wave", often in the region of 1.6 Hz. As both the acceleration and the deceleration part can be seen, there will be two force values for any given velocity except for the maximum velocity. Sometimes the terminology "dynamic testing" is used for this type of damper testing, including other types of movements than just sine wave movements, but with the common factor that the measurements are done during variations in velocity. As can be seen in the figure, the acceleration force values are lower than the deceleration force values when passing zero velocity. This separation at low velocities is called "hysteresis". See chapter *Hysteresis* for more information. When "matching" dampers, the teams often pick half of the cycle to keep the overlay of curves from different dampers as clean as possible. It is common to use compression opening (measurement during acceleration) and rebound closing (measurement during deceleration).

Many times the result of continuous damping force measurement can be presented in a force-displacement graph. These types of curves are sometimes named "egg curves".

If continuous measurement is used, the forces given are dependent on the stroke and frequency being used. By changing the crank length and the frequency of the dynamometer machine so to maintain the same peak velocity, let say 5 inch/second, the damping force curves will change. From the formula below, you find that, for example, a reduction of the amplitude to half will give the same peak speed if the frequency is doubled.

$v_{max} = A \bullet 2\pi \bullet f$ [m/s]
v_{max} = peak velocity   [m/s]
A = amplitude   [m]
f = frequency   [Hz]

The less travel required to reach a specific velocity, the more pronounced the hysteresis will be in the graph. As hysteresis is found when the movement change direction, in other words at zero velocity, tests where different stokes have been used to produce the same peak velocity will differ at low velocities due to hysteresis. See chapter *Hystersis* for more information.

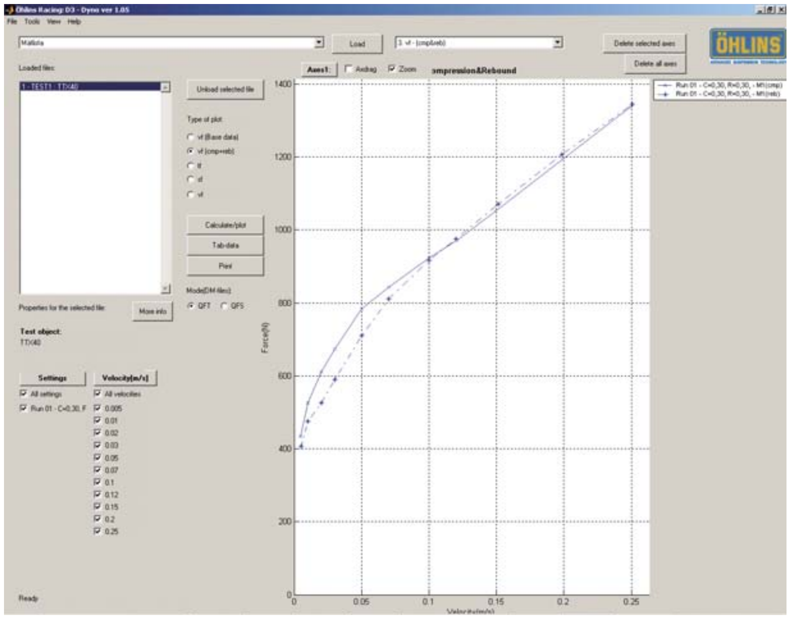


Figure 5.3 *Damping force measured at constant velocity. The lined curve is compression, the dotted curve is rebound. The setting is the same as in the two previous figures.*

Sometimes the software uses a filter to reduce the level of noise in the graphs.

NOTE:
Comparing the forces figures 5.2 with the force of 5.3, there is a huge difference in force at low speed. When the damping force is measured continuously the hysteresis give the impression that the bleeds are quite open and a small bleed change can be hard to notice in the graph

Some race teams do damping force measurement at constant velocity. Sometimes the terminology “static testing” is used for this type of testing. For matching dampers this method is excellent. For a damper manufacturer, working with different race teams using different methods of measuring damping forces, measuring at a constant velocity is preferred. Otherwise as explained above, when using continuous measurement, there is always a risk that the values discussed comes from different methods of measuring.

When the damping force is given at a constant velocity, there is only one value of the damping force at each specific velocity. A crank dynamometer can’t be used for constant velocity measurement, but by measuring the damping force at different peak velocities, the result is normally very close. When a crank dynamometer is used, the machine always produces sine wave movement. By changing the frequency of the dynamometer, different peak velocities are reached. The number of runs varies depending on the needs. Of course small steps extend the test time. In a hydraulic dynamometer, the piston rod is accelerating to the desired velocity in a short distance, and the flow of oil through the valves in the damper is kept “static” during a large part of the stroke. This makes it possible to take hundreds of measurement before the dynamometer decelerates the piston rod.

NOTE:
In this manual, if there is no other information, all graphs illustrated come from Öhlins TTX VRP. The data used in the TTX VRP are produced by an Instron hydraulic dynamometer at Öhlins Laboratory. The forces are measured at constant velocity.

It is very important to know when dynamometer testing dampers if the forces measured are compensated for gas force or not.

Most racing dampers (pressurised, and not through rod type) add a gas force to the damping force. The gas force should be seen as an extra spring force from a spring with very low rate. This force is position dependant (close to constant) and not velocity dependant and should therefore be removed when damping force is plotted.

An idea of the amount of gas force a damper produces at a specific piston position (normally small variations at piston positions) can be found by compressing the damper by hand and keeping it at a static position. The gas force will now try to push the piston rod out of the damper body. This force is calculated as

$$\boxed{F_{rod} = p_{gas} \bullet A_{rod} \quad [\,N\,]}$$

p_{gas} = gas pressure above atmos phere pressure (the value read on the pressure gauge when pressurizing the damper) [N/m^2, 1 bar=10^5 N/m^2 = 15 psi]

A_{rod} = piston rod area

For example, at 10 bar (150 psi) gas pressure in the TT44/40 (∅16 mm piston rod), the gas force is approximately 200 N (45 lbs).

When you look at a dynamometer curve that has been gas force compensated, you are looking at the actual damping forces produced by the damper.

For non gas compensated dynamometer curves, the actual damping forces are calculated from the formulas below.

Compression damping = measured compression force – F_{rod}

Rebound damping = measured rebound force + F_{rod}

The TTX damper has no resulting force from the gas pressure ($F_{rod} = 0$), so it always gives the same forces no matter if the testing is gas compensated or not.

NOTE:

If matching the forces from a TTX damper with the forces from a conventional damper tested without gas compensation, the forces of the conventional damper has to be gas force compensated with the formulas above to get the same amount of damping.

NOTE:

Keep in mind that even if the method for measuring the damping forces is identical there can be some small variations in the result from different damper dynamometers, due to individual variations between the machines.

NOTE:

All damping curves will change with temperature, so always keep track of the temperature. See chapter Temperature stability for more information.

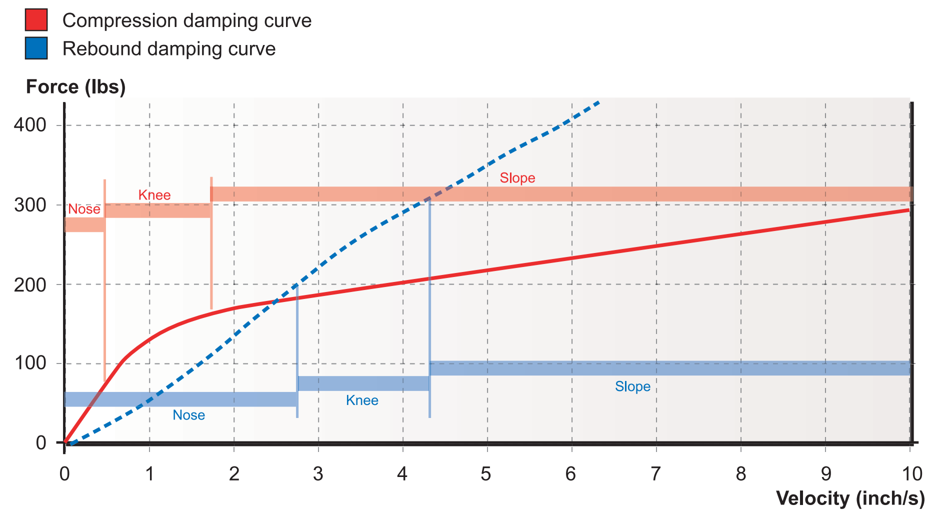


Figure 6.0 *Terminology*

6 Damping Curve Terminology

In order to understand the next part of this manual we must all speak the same language. In the damper industry there are terms used to help describe different parts of a dynamometer graph. The three key words are *nose (low speed), knee* and *slope (high speed)*. Careful study of this section will yield a complete understanding of these terms and allow you to read damper curves. Later chapters will show you how to manipulate the damper to produce alternate curves.

The horizontal axis in the figure gives damper velocity and the vertical axis shows damping force.

NOTE:

The velocity of the damper refers to the velocity of the piston rod movements, not to the speed of the car. Most of the piston rod movements on a race car reach only low velocities and the percentage number of strokes going though all zones described below is low.

Take a look at the figure above and notice the first portion of the damping curves – starting at 0 inch/second and ending at about 0.4 inch/second on the compression curve and about 2.7 inch/second on the rebound curve. This part of the curve is called the *nose* and is also referred to as *low speed*. The valve affecting this part of the curve is the low speed adjuster. It is always of a fixed orifice type (the size of the orifice is not variable by pressure, but by clicker adjustments) and is often called *bleed*. The design and size of the bleed determines the characteristics and shape of the nose. For further information, see section *Low speed adjusters* in *External adjusters*.

The finish of the nose zone coincides with the beginning of the *knee* zone. Its location in the curve can be found by identifying where the upwards curve first begins to level off into a radius. Locating where the knee radius stops and blends into the straight line identifies the end of the knee zone and the beginning of the *slope* zone.

The shape of the knee is determined by the opening characteristic, gradual or abrupt. The more abrupt the opening phase is, the sharper the knee will be. The converse is also true.

Normally the shape of the curve in the knee zone comes from the transition of the nose shim stack from closed to open position. Many times the poppet valve is set to blow of just after the nose shim stack has opened. Then the opening of the poppet will constitute the end of the knee zone. This is the case here on the compression side at approximately 1.2 inch/second.

The TTX damper has the possibility to produce two knees that can be very much separated (not in this example). Here the first knee is determined by the initial opening of the shim stack (at about 0.4 inch/second) and the second knee by the opening of the poppet valve (at about 1.2 inch/second). The abrupt opening of the poppet valve gives a shorter knee than the more gradual opening of the nose shim stack.

Due to the more open bleed on the rebound. The kneesare not so pronounced.

NOTE:

The knee from the poppet valve is extremely sharp compared to the knee from the shim stack. Both the low and high speed adjustment in combination with the poppet valve hardware including the shim stack in the nose determines the position and shape of the knee/knees. For further information, see section Nose shims effect on damping curve and section High speed adjusters effect on damping curve.

NOTE:

Wide open bleeds in combination with a stiff shim stack and a lot of preload on the poppet valve can allow the low speed zone to extend into relatively high velocities.

The size of the poppet valve and the rate of the coil spring normally determine the slope, also referred to as high speed. In most cases the slope will continue to rise in a straight line to damper speeds well beyond those found in most racing dynamometer charts. Eventually the slope will increase at an exponential rate. This happens when the size of the channels transporting the oil begin to restrict the oil flow (channels are also fixed orifices). The slope angle relative to the horizontal plane defines the magnitude of the slope and can be quantified as Pound/(Inch/Sec.) or N/(m/s).

Some special cases should also be mentioned here. A combination of a stiff nose shim stack and low preload of the poppet valve could take the nose shims more or less out of action. This would give only one sharp knee. More likely is that single knee curves are achieved by running high preload on the poppet, so it never comes off the seat. If so, the high speed is determined by the shim stack only.

The nose, knee and slope are key words to understanding the following concepts.

Figure 7.0 *External adjusters*

7 External Adjusters

General description

Most dampers that are external adjustable have some type of low speed adjuster. Low speed adjusters are almost always externally adjustable orifices that become fixed after adjustment. Fixed in the sense that the orifice area is not dependent on the pressure drop over the orifice. In the damper industry, these adjustable orifices are often referred to as bleeds or low speed adjusters. Unlike shim stacks, bleed orifices do not change size in response to changes in pressure. Because oil will always travels the path of least resistance, it will first flow through the open bleeds until there is enough pressure to open any other valves. Oil flows through the bleeds any time the piston rod is moving, and continues to flow in parallel with the flow through the piston shim stack after the stack has opened.

The most common type of external high speed adjuster is an adjuster that moves the knee up or down without changing the slope, or just marginally changing it. To achieve this, the amount of force pushing the valve, shim or poppet valve, against its seat is varied. That is done by changing the preload of the spring element, shim stack, coil spring, cup spring etc.

The oil flow that is controlled by the external adjusters varies between different type of dampers. The larger the flow is, the better the conditions will be for a powerful adjuster. There are two reasons for that.

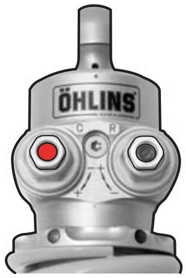
- A larger flow is easier to control. The tolerances for the dimensions on the valve parts have to be tightened if the flow is reduced.

- A larger pressure area, the pressure area is proportional to the oil flow, will keep the internal pressure of the damper at a lower level. This increases the damper response and the damper will build up damping force quicker.

External adjusters summarised

The TTX damper from Öhlins comes as a 4-way externally adjustable damper.

As some racing classes have rules about the maximum number of external adjusters there might be some optional parts available in the future reducing the number of external adjusters.

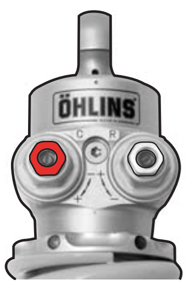


Low Speed Compression Damping Adjuster (LSC)

Type of adjuster:	Bleed adjuster.
Effects:	The flow from the main piston during a compression stroke only.
Pressure area*:	1143 mm^2 (∅40 - ∅12)
Identification:	The black screw with an internal 3 mm hex** inside the gold 12***mm hex at the cylinder head.
Number of positions:	40
Click position 0:	Fully clockwise at maximum force (orifice closed).

Low Speed Rebound Damping Adjuster (LSR)

Type of adjuster:	Bleed adjuster.
Effects:	The flow from the main piston during a rebound stroke only.
Pressure area*:	1143 mm^2 (∅40 - ∅12)
Identification:	The black screw with an internal 3 mm hex** inside the silver 12***mm hex at the cylinder head.
Number of positions:	40
Click position 0:	Fully clockwise at maximum force (orifice closed).



High Speed Compression Adjuster (HSC)

Type of adjuster:	Poppet valve.
Effects:	The flow from the main piston during a compression stroke only.
Pressure area*:	1143 mm^2 (∅40 - ∅12)
Identification:	The golden 12**mm hex at the cylinder head.
Number of positions:	Approx 50
Click position 0:	Fully clockwise at maximum force (spring max preloaded).

High Speed Compression Adjuster (HSC)

Type of adjuster:	Poppet valve.
Effects:	The flow from the main piston during a rebound stroke only.
Pressure area*:	1143 mm^2 (∅40 - ∅12)
Identification:	The silver 12**mm hex at the cylinder head.
Number of positions:	Approx 50
Click position 0:	Fully clockwise at maximum force (spring max preloaded).

* As it is a through rod damper, the compression and rebound pressure areas are the same. The pressure area multiplied with the piston velocity give you the flow (volume per time unit) of oil that passes through the valves.

** During year 2005, an internal 3 mm hex replaced a screwdriver slot.

*** The 12 mm hex has previous been 13 mm.

As can be seen in the figure, the cylinder head is market “C” and “R” together with “+” and “-”. This will help the first time users to separate Compression from Rebound and tells what direction the adjusters should be turned to increase (+) or reduce (-) damping.

***NOTE*:**
All external adjusters are ”fully hard” when turned clockwise until they stop. The clicker positions, including the high speed adjusters, is always counted from ”fully hard”.

The reason is ”full hard” is always an absolute position. ”Fully soft” will vary more depending on the tolerances, so the matching wouldn’t be as good if the clicker positions were counted from full soft.

The first click and/or detent is counted as ”zero” position.

To match damping curves of a pair of dampers in the dynamometer, sometimes the clicker numbers will end up a couple of clicks from each other. Often they match within ± 1-2 clicks, but sometimes you can see ± 3 clicks.

Just remember maximum clockwise is “full hard” for all adjusters and the adjusters are counted from full hard.

Low speed adjusters

The TTX follows the tradition from the TT44/40 with two fully independent low speed adjusters adjusting the main piston oil flow. For a specific damping force, the internal pressures can be kept low, giving a minimum of delay in the damping force build up.

The two low speed adjusters (bleeds) LSC and LSR are identical and are designed so that in the normal operating range each click of the adjusters will change the damping in close to equal steps. They are tapered needles working in fixed orifices, but due to a double coned nose of the needle, the damping force doesn’t increase progressively per click as the needle is closed. The adjusters are powerful over the whole range making it easier to find the optimum settings.

Both adjusters have a range of approximately 40 clicks. Normally they match within ± 2 clicks.

The bleeds are adjusted with a 3 mm Allen key. During year 2005, an internal 3 mm hex in the needle replaced a screwdriver slot. Do not use too much torque when closing the bleeds completely.

Generally it is better to start with the adjusters a little more open and gradually close them off. See chapter *Damping guidelines* for more information.

Low speed adjusters effect on damping curve

Assuming the LSC and the LSR have the same clicker setting, the piston velocity is the same and that there is no oil flowing through the high speed valves (including the nose shims of the poppet valves), exactly the same amount of oil will flow through both valves. The LSR has exactly the same affect on the nose of the rebound damping curve as the LSC has on the nose of the compression damping curve.

The different curves in the figure are achieved by adjusting only the LSC/LSR. The numbers above the curves represent low speed adjuster clicker settings. The dotted curves pointing upwards indicate the theoretical shape of the curves if there was only a low speed valve and no high speed valve to open. The poppet used here to produce the curves is special made with a flat nose instead of the standard triangular shaped nose to ensure that no oil goes through the nose shim stack. See chapter *Adjustment and valving charts* or the VRP (Valving Reference Program) for real values when the low speed valve is used in combination with different nose shim stacks.

If the adjuster is turned counter clockwise the clicker numbers get higher. As the bleed is opened more and more, the damping is reduced and the speed at which the knee starts increases and the

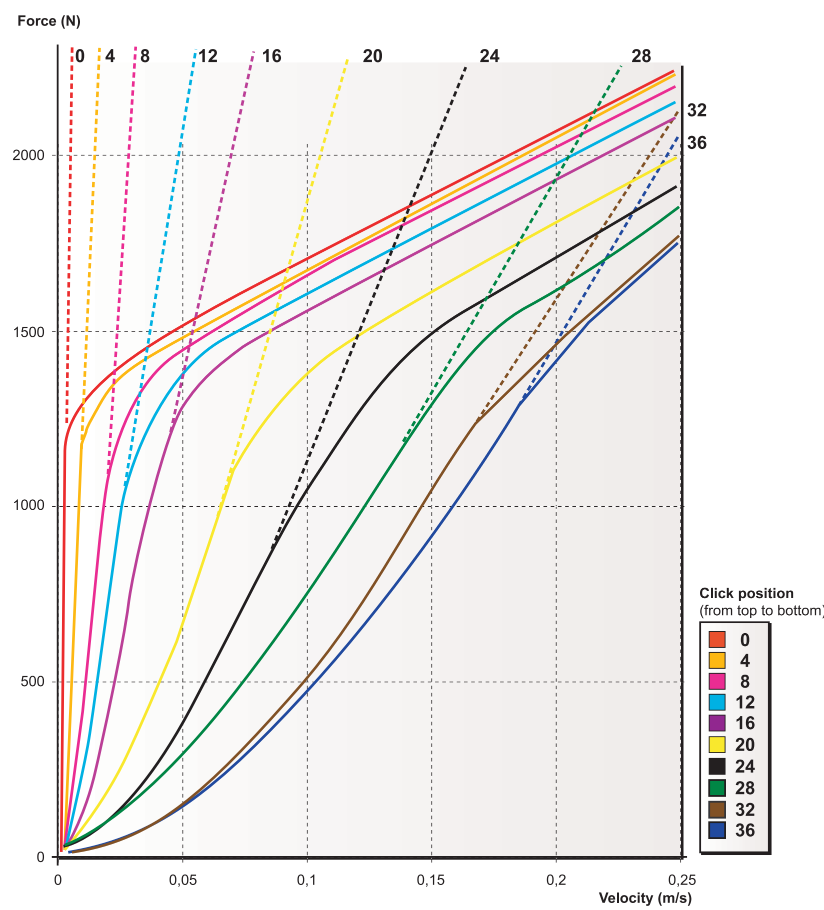


***Figure 7.1** Influence of the low speed adjusters. The graph represents both the LSC and the LSR.*

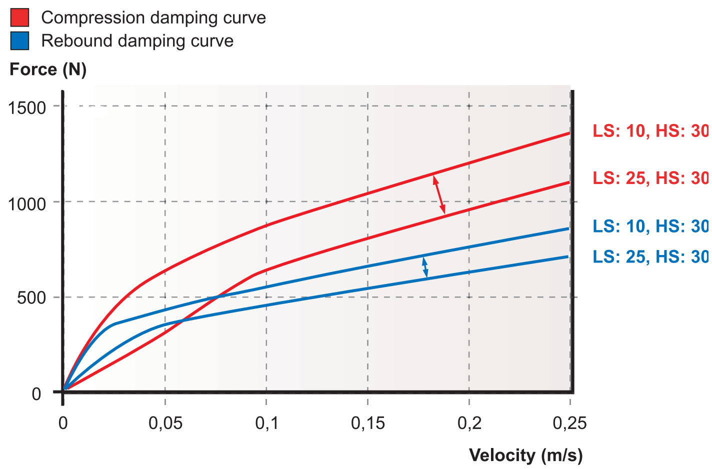


Figure 7.2 *The low speed adjusters affect on the high speed at different high speed slopes. See chapter "Adjustment and valving charts" or the VRP) for real values.*

nose is stretched longer and longer. Also, the force of the knee and the slope angle will remain the same. This is because only the high speed valve determines the knee height and the slope.

This means that even the high speed damping will drop as the low speed adjuster is opened. The more slope there is, the more the high speed will drop. Conversely, if you wish to keep the knee force constant, the poppet preload has to be increased every time the low speed adjuster is opened.

If the low speed adjuster instead is turned clockwise, the opposite from above will be true.

High speed adjusters

The TTX has two hight speed adjusters; one for compression and one for rebound. A damper like the TTX is often called fully adjustable. This provides race engineers and technicians with more external control over the damping. Anything that can make an adjustment quicker and thereby gain time during practice and qualifying is important.

Just as the low speed adjusters, the high speed adjusters HSC and HSR are identical (only the colour of the needle housing is different), fully independent and adjust the main piston flow. They are delivered with the same valving.

Both high speed adjusters consist of a poppet valve preloaded by a coil spring. The coil spring pushes the poppet against its seat. The preload of the spring determines the pressure differential needed to make the poppet lift from its seat. The adjusters change the amount of preload. A preload change will change the height of the knee. Each click of the adjusters changes the damping in almost equal steps.

A poppet valve works like a shim stack. Both are pressure regulators that control oil flow by opening at a pre-determined pressure and thereby providing a path for the oil to flow.

In the nose of the poppet valves, shims make it possible to adjust the sharpness of the knee. An alternative poppet valve is available with a corresponding valve seat. Also there are three other springs available. See Internal adjustments for more information about the alternative hardware in the valves.

Each high speed valve consists of the following parts that can be changed

- 04104-02 Valve seat ⌀12
- 04103-06 Poppet valve
- 04107-04 Spring c=40 N/mm, t = 2.20 mm
- 01415-14 x 3 Nose shim, 0.15x14

In the first generation of TTX dampers the poppet valves and springs were different. These dampers are marked TTX NE020 and TTX NE040. In these dampers, the parts used are

- 04103-04 Poppet valve
- 04134-04 Spring, c = 40 N/mm, t = 2.40 mm

These parts generate the same forces. The new 04107-04 Spring is 1 mm shorter (15.5 mm instead of 16.5 mm), but on the other hand the 04103-06 Poppet valve is 1 mm higher (3.5 mm from the surface of the triangle to where the spring stands instead of 2.5 mm). The same thing is true for the new alternative poppet valve 04103-07 compared to the old 04103-05. The reason fo r the change is that the new combination keeps the high speed less progressive at high velocities (about 1.5 m/s).

For information about the alternative valve hardware, see Internal adjustments.

Both high speed compression and rebound adjusters have a range of approximately 50 clicks. Normally they match within ± 2 clicks. The thickness of the nose shim stack affects the number of clicks. An increased thickness of the stack reduces the number of clicks. For example, if the shim stack thickness is increased by 0.25 mm, there will be about 4 clicks less and vice versa.

The preload is adjusted with a 12 mm wrench. A 13 mm wrench has previous been used.

Note: It is no problem to run the high speed adjusters at clicker position 0, as it is the needle housing that bottoms against a circlip and not the spring that is coil binding.

High speed adjusters effect on damping curve

If the high speed compression and rebound valves have identical hardware and the adjusters are set the same, the damping forces will turn out the same.

The height of the knee is changed and the steps are about the same over the whole adjustment range. The low speed isn't affected at all and the slope remains unchanged. To change the slope, the hardware has to be changed.

As mentioned before, there is one alternative poppet valve (part no. 04103-07) and three other springs (04107-01 (10 N/mm), -02 (20 N/mm) and –08 (80 N/mm)) available. The figures below illustrate how these parts will change the adjustment range. For information about the alternative hardware, see Internal adjustments.

The alternative poppet valve 04103-07 increases the adjustment range and raises the slope. This poppet valve can of course also be combined with the alternative springs.

Figure. The alternative valve springs affect the adjustment range compared to the standard spring 04107-04 (40 N/mm) if the other valve parts are kept standard and the low speed adjuster is set at clicker position 20.

What can be noticed is that the adjustment range changes a lot depending on the spring used, but the influence on the slope is fairly small. The window of adjustment starts at about the same level for each spring, but reaches different maximum limits. The difference in the maximum force is proportional to the difference in spring rate. The same is true for the change in force per click adjustment.

Shaping the high speed

The height of the knee depends on the stiffness and preload of the spring together with the pressure area of the poppet. If the area is small it takes a higher pressure to overcome the force that is acting on the poppet. The spring rate and the pressure area also control the slope of the graph by restricting the maximum size of the fluid path past the poppet.. This is why the alternative poppet (part no. 04103-07), that is smaller, lifts the knee higher and increases the slope.

Note: The spring rate is not a very powerful tool when it comes to changing the slope. For example if the spring rate is increased by a factor 10, the slope will only rise about 60%, but the adjustment range is increased about ten times.

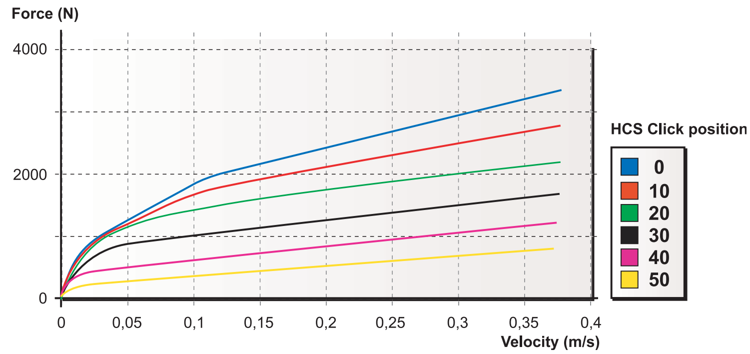


***Figure 7.3** The influence of the high speed adjusters. Here the standard valving is used and the low speed adjuster is set at clicker position 0. The graph represents both the HSC and the HSR.*

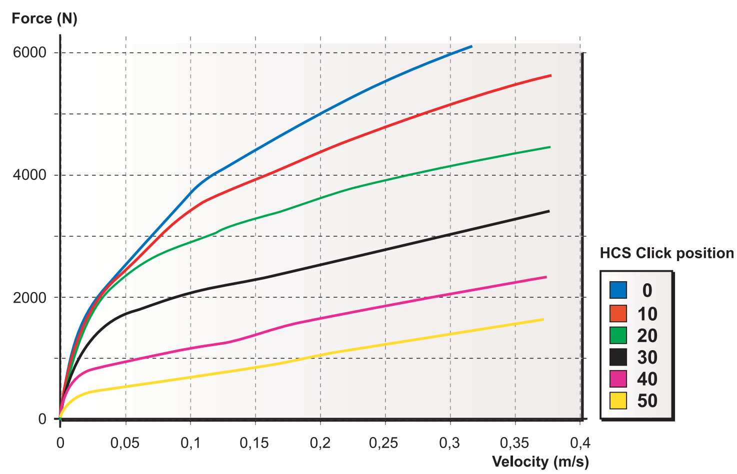


***Figure 7.3** The alternative poppet valve 04103-07 affects on the adjustment range compared to the standard poppet valve 04103-06 if the other valve parts are kept standard and the low speed adjuster is set at clicker position 0.*

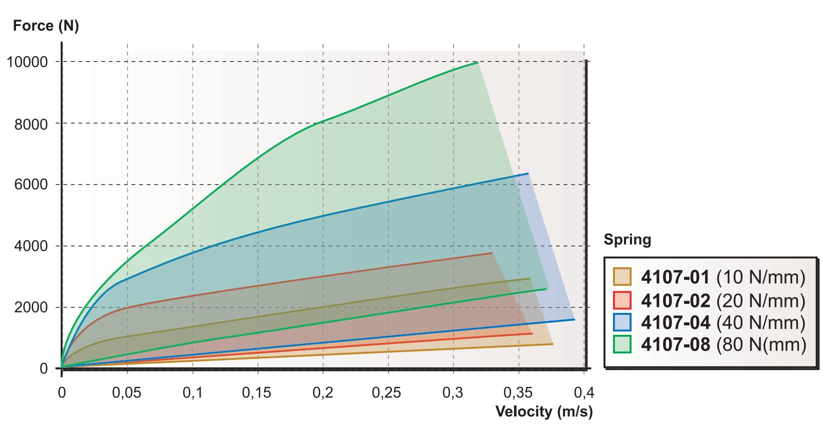


***Figure 7.4** The alternative valve springs affect the adjustment range compared to the standard spring 04107-04 (40 N/mm) if the other valve parts are kept standard and the low speed adjuster is set at clicker position 0.*

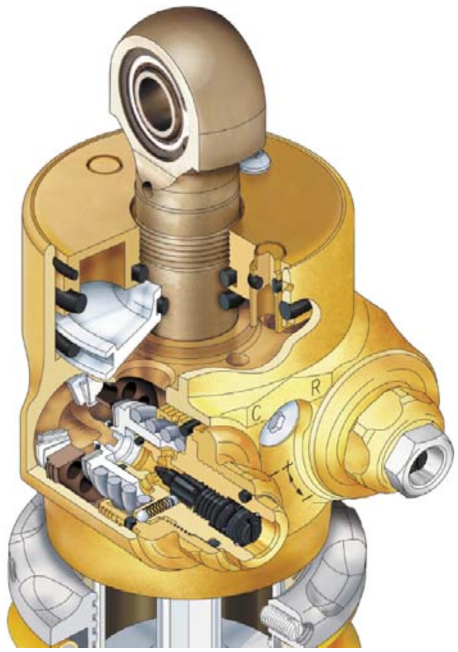


Figure 8.0 *Internal Adjustments*

8 Internal Adjustments

Now we have just learned about the external adjusters. We now need to look inside to see what tools are available to influence the damping.

Unlike the TT44/40 damper, there are no valves on the main piston. The high speed valves instead consists of two identical poppet valves. This gives more freedom to use the same poppet for either compression curves and rebound curves.

Poppet valves

There are two different poppet valves available. They are

- 04103-06 Poppet valve (standard)
- 04103-07 Poppet valve

The optional poppet valve has a smaller nose and goes with smaller valve seat (∅10 mm hole). The two available valve seats are

- 04104-02 Valve seat (∅12, goes with poppet 04103-06)
- 04104-01 Valve seat (∅10, goes with poppet 04103-07)

NOTE:
Do not use other combinations of poppet valve and valve seat.

In the first batch of TTX dampers the poppet valves and springs were different. See section High speed adjusters for more information.

The standard poppet valve (04103-06) has a triangular plateau on the nose, while the smaller poppet (04103-07) has a rectangular shaped nose. The reason for these shapes is to allow the nose shims to lift away from the seat. With the high speed coil spring pushing the poppet from behind, it clamps the stack against the seat. The standard poppet clamps the shims at the corners of the triangle. This means that the nose shims can lift in the three areas outside the triangle. The optional poppet does the same thing except the shims are allowed only to bend in two areas opposite from each other along the rectangular plateau. This shim action is comparable to the difference between the shim movement on a conventional 3 port piston versus a 2 port piston.

If the poppet together with the hole of the valve seat is smaller, the forces from the nose shim stack increase and due to the physical reduction in the shim area allowed to bend (leverage), changes in shim thickness will do very little to lower the stack stiffness. To get the leverage back, and thus the adjustability of the shims the nose on the smaller poppet has gone from a triangular shape to rectangular.

Each poppet valve is guided by its needle housing. On each needle housing there is an o-ring that both seals and damps the poppet. Without the o-ring, the poppet would oscillate on the undamped spring.

See the previous chapter External adjusters for information about how the adjustment range will be affected by a poppet valve change.

Valve springs

There are four different valve springs available. They are

- 04107-01 Spring c=10 N/mm, t = 1.70 mm
- 04107-02 Spring c=20 N/mm, t = 1.80 mm
- 04107-04 Spring c=40 N/mm, t = 2.20 mm (standard)
- 04107-08 Spring c=80 N/mm, t = 2.75 mm

In the first batch of TTX dampers the poppet valves and springs were different. See section High speed adjusters for more information.

See the previous chapter External adjusters for information about how the adjustment range will be affected by a poppet valve change.

Nose shims

A poppet valve generally gives a very sharp knee due to the abrupt opening/closing characteristic. However, it's possible to design a poppet valve that opens smoothly. To do this with the poppet alone you would need many different designs each with there own knee characteristics. Instead of doing this, the TTX uses shim stacks to manipulate the opening characteristic of the poppet. Before the poppet has lifted from the seat, the shims are clamped between the poppet and the valve seat. As the shim stack gradually starts to open before the poppet does, the oil is released gradually creating a smooth transition from closed to open. Using nose shims gives more freedom of adjustment and keeps the cost down for the customers as they don't need a lot of different poppet valves.

When the poppet has lifted from the seat, the pressure and oil flow will keep the shims towards the poppet. So the damper can actually be used without the circlip in the nose of the poppet, but this is not recommended.

The shim stack continuously interacts with the poppet valve. Even after the poppet has lifted from the seat the shims in the stack continue to move. This makes it hard to predict the shape of the damping curve without testing the particular poppet, spring and shim stack combination.

By using these shims, it is also possible to run steep slopes at high speed by just using the shim stack as the high speed valve. This is achieved by using a fairly stiff shim stack in combination with a high preloaded poppet valve so the poppet doesn't open. The highest opening pressure is achieved with the small poppet (part no. 04103-07) in combination with the stiffest spring (part no. 04107-08) used at clicker position 0. However, this eliminates the external high speed adjustment.

The internal diameter of all shims is 6 mm. For each poppet, only one outer diameter is used.

The available shims for the standard poppet valve 04103-06 has outer diameter 14 mm and are

- 01410-14 Shim t = 0.10 mm
- 01415-14 Shim t = 0.15 mm
- 01420-14 Shim t = 0.20 mm
- 01425-14 Shim t = 0.25 mm
- 01430-14 Shim t = 0.30 mm

The available shims for the alternative poppet valve 04103-07 has outer diameter 12 mm and are

- 01410-12 Shim t = 0.10 mm
- 01415-12 Shim t = 0.15 mm
- 01420-12 Shim t = 0.20 mm
- 01425-12 Shim t = 0.25 mm

The number of combinations is huge.

The combinations used in the Valving Reference Program (VRP) for the standard poppet 04103-06 are:

- 2x 0.15-14
- 2x 0.15-14 + 1x 0.10-14
- 3x 0.15-14
- 2x 0.20-14
- 4x 0.15-14
- 2x 0.20-14 + 1x 0.15-14

The combinations used in the Valving Reference Program (VRP) for the alternative poppet 04103-07 are:

- 2x 0.15-12
- 3x 0.15-12
- 4x 0.15-12

These stacks are listed softest to stiffest.

Note: *Keep in mind that a thickness change of the shim stack changes the preload of the poppet. A height change of 0.0625 mm corresponds to 1 click.*

Nose shims effect on damping curve

If the low speed adjuster is set to a closed position, the nose shim stack knee can be studied.

As the bleed is fully closed here, all flow goes through the shim stack until the poppet lifts from the seat. The slope between the two knees is depending on the rate of the shim stack. The slope raises as the stiffness of the shim stack increases.

Note: The nose shim stacks can be compared with non preloaded main piston shim stacks, as the load from the poppet on the shims doesn't preload the shims, but just clamps them.

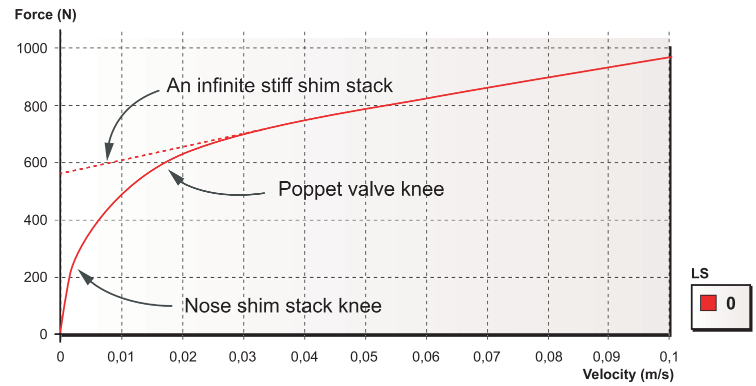
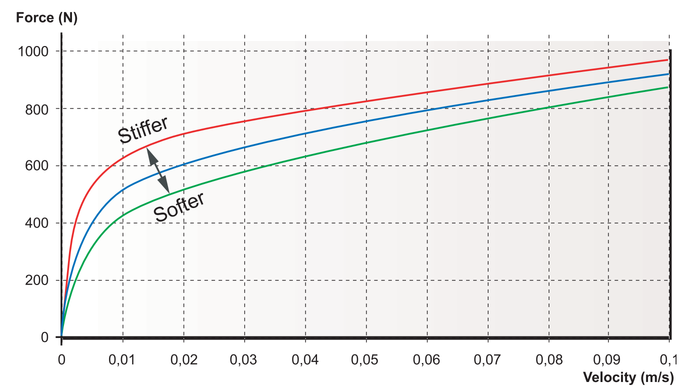


Figure 8.2 *The shim stack knee and the poppet valve knee.*



9 Limits in Force and Velocity

The TTX is designed for pretty low damper velocities. It is hard to give a definitive limit in velocity, but peak velocities from 1 to 1.5 m/s (40 to 60 inch/s)are no problem.

When it comes to the limit in damping force, it is important to know that the peak forces are not normally the problem, fatigue is. Be aware that increased temperature reduces the strength of the materials. Peek loads at 90ºC (250ºF) reaching 8000 N (1800 lbs.) can be handled by the TTX with no problem.

10 Matching of Damping Force

Even with tight tolerances, the types of high speed valves used in the TTX are hard to get to match compared to main piston shim stack valves. This is the drawback of this type of valves. The adjusters can on the other hand be made extremely powerful and if a dynamometer is available, the graphs can be spot on if the adjusters are changed a few clicks.

Dampers set at the same clicker positions should match within ±10%, but many times they match within ±5%.

All TTX dampers are dynamometer tested at Öhlins Racing before they are delivered.

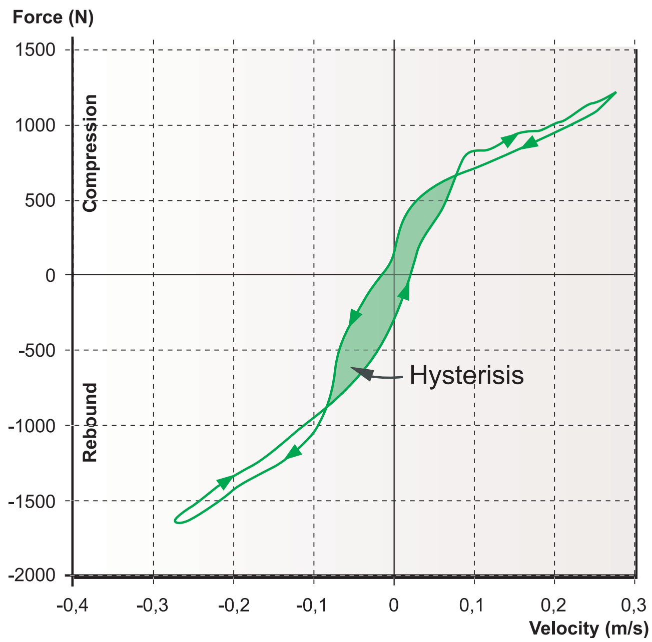


Figure 11.0 *Damping force during a complete sine-wave cycle illustrating hysteresis. Due to the low hysteresis of the TTX, some oil has been replaced with air to get a better illustration of hysteresis.*

11 Hysteresis

Start by studying chapter *Damping force measurement*.

Among teams, the experience of hysteresis mainly comes from continuous measurement of damping force during sine wave runs in a dynamometer. On those damper graphs, there is always a difference in ascending (acceleration) and descending (deceleration) parts of the curve around zero velocity. We normally refer to this area as *hysteresis*. At constant velocity or peak force measurements, the hysteresis can't be detected. Technically, the term hysteresis is related to energy losses, but here we are actually storing energy as the damper acts like a spring.

Hysteresis is actually flex in the damper system. The flex delays pressure rise and pressure drop. Hysteresis affects the performance of the damper. Generally, a minimum of flex is desired. Especially where there are very short damper movements. For example in single seaters, with very short strokes, you are dependent on quick damping force build up and if there is any delay, the damper might changes direction before there is any damping at all. Because of this, hysteresis needs to be kept to a minimum.

How much the delay affects the damping curve is very much related to the stroke and frequency in the test. When keeping the maximum velocity constant and varying the frequency and stroke, it is very obvious that with a short stroke and a high frequency the hysteresis deforms the damping curve more than long stroke and low frequency.

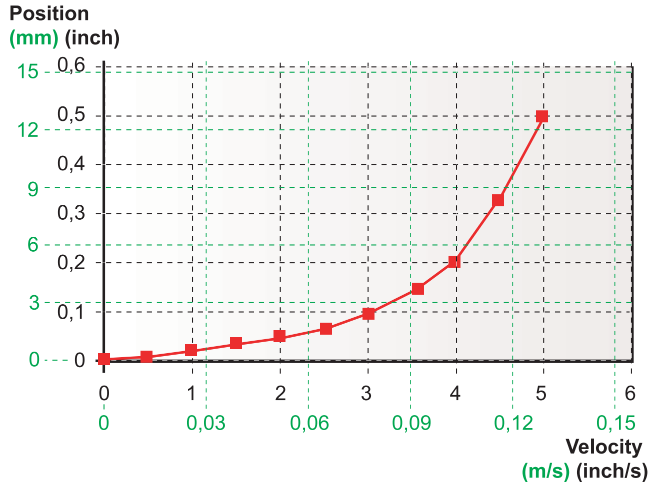


Figure 11.1 *Stroke-velocity relation in a typical dynamometer run. Peak velocity 5 ips (0.127 m/s) and frequency 2 Hz.*

In the graph (figure 11.1), the relation between displacement and velocity is shown for a peak velocity of 5 ips (0.127 m/s) and a frequency of 2 Hz.

NOTE:

The displacement at lower velocities is very small. Because of this, still with very little flex in the damper, the hysteresis in a dynamometer graph can easily be seen.

In a graph illustrating damping forces, the hysteresis changes the damping curve in a similar way as a more open bleed would do – both delay the damping force build up when using a sine input. The force drop from a more open bleed depends on the actual velocity, while the force drop due to increased hysteresis depends on the stoke. The setting and the hysteresis affect each other. The hysteresis increases at increased damping.

If only the compression acceleration and the rebound deceleration are illustrated, you don't see the amount of hysteresis. Therefore, it is easy to believe the bleed is more open when it actually is due to the hysteresis. There have been situations when teams have used the dampers with a low speed adjuster set to almost fully closed position. As the damping curve nose at continuously measurement still doesn't go vertical, the teams have asked for a different adjustment range of the low speed adjusters to put the actual clicker setting more in the middle of the range. Here, they have been misled by the hysteresis. Due to the low amount of hysteresis of the TTX compared to other dampers, this should be less of a problem.

Damper flex can be classified into three different groups: flex of damper parts, flex due to damper oil compressibility, and flex of gas present in the oil (dissolved or bubbles). The flex of the damper parts is fully elastic and linear. The compressibility of the oil itself is not linear. Some hysteresis comes from the compression and expansion of gases in the oil. Gas bubbles in the oil will be compressed in a very progressive way and will act almost like a slack in the system. Dissolved gas in the oil is a result of different factors. Air enters the damper during the filling procedure and some air is already dissolved in the oil when it is delivered. By using a vacuum filling machine, the amount of air in the oil can be minimised. With a vacuum machine there is no risk of trapping air in the damper. The oil used for filling the damper has been under low pressure for some time before it enters the damper. This removes air dissolved in the oil. But even without air, there will be gas bubbles at low pressures. Oil contains different additives that boil at different pressures/temperatures. At low pressure, those additives change to a gaseous form which creates bubbles. See chapter *Cavitation* for more information.

The amount of hysteresis for a certain damping force can be very different depending on the size of the piston (pressure area) and the volume of oil that is pressurised. The larger the piston is, the easier it will be to reduce the hysteresis. This is explained by the formula $F = p \bullet A$. F is the force, p is the pressure and A is the pressurised area. For a specified damping force (F) a smaller area (A) will lead to a higher pressure (p). The higher pressure will make the damper flex more. This will cause more hysteresis. On conventional dampers (the gas reservoir is connected to the compression side), not on the TTX, the piston rod acts as a small diameter piston sending oil to the valve in the reservoir. Because the effective pressure area is very small and the total oil volume is large, there will be a lot of hysteresis from this portion of the damping force compared to the damping force produced by the main piston.

On all dampers where the internal pressure of the damper pushes the piston rod out of the damper body, a "hysteresis effect" might be seen on the dynamometer graph. This is not the case with the TTX. How much of this "hysteresis effect" that will be seen depends on how well the dynamometer compensates for the gas force. The reason for this is that on a conventional damper the static piston rod force varies depending on the position of the piston rod. The more compressed the damper is, the higher the internal pressure will become due to the gas volume being compressed. For example, if the gas force compensation only reduces the compression damping and increases the rebound damping with the static piston rod force measured when the damper is at maximum length in the dynamometer, the compensation will not be enough when the damper is more compressed. This difference can be mistaken for hysteresis. Good gas force compensation will avoid this problem.

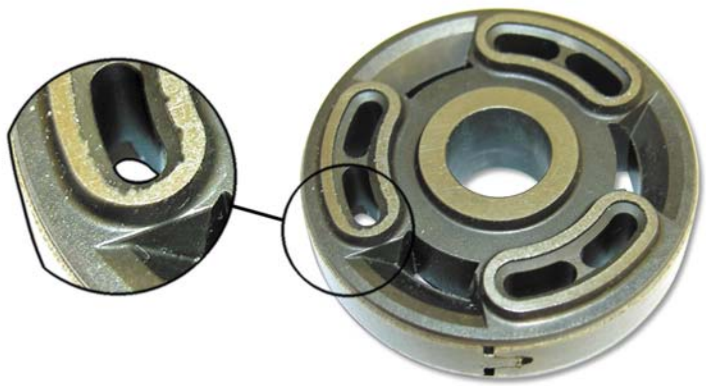


Figure 12.0 *Example of cavitation pitting on a shimmed piston.*

12 Cavitation

Cavitation is a word used a lot in the pump and damper industry for describing the phenomenon when gas bubbles are produced in fluids at pressure drops. The gas comes from both the fluid that has changed its state from fluid to gas and from air that had been dissolved in the fluid. Cavitation also includes the collapse of the gas bubbles when the pressure increases and the gas returns to liquid form.

In conventional dampers, cavitation very often occurs on the rebound side of the piston. If no reservoir valve is used the damping force mainly comes from a pressure drop on the rebound side. Here the set gas pressure has to be high enough to allow a pressure drop on the rebound side without letting it get too low. By using a reservoir valve the pressure doesn't have to drop as much on the rebound side as there also will be some increased pressure on the compression side due to the reservoir valve. However, if the piston rod is extended near full rebound, the volume on the rebound side will be small and the pressure will drop very quickly, so some cavitation will still occur.

Cavitation in dampers should always be prevented. Cavitation can cause serious malfunction, reduced performance and damage the damper.

Absolute pressure, temperature and type of fluid are factors that affect the risk of cavitation. Normally, an absolute pressure drop below 0.7-0.8 bar (10-12 psi) cause cavitation. For your information: The absolute pressure in the atmosphere is close to 1 bar (15 psi) at sea level altitude. High piston accelerations will increase the risk of cavitation as pressure changes do not immediately effect the entire volume of oil due to delays in the pressure distribution in the fluid. These different pressures in the same volume of oil are sometimes called "dynamic pressures".

As cavitation is a state change from fluid to gas, compare boiling of water, an increased temperature increases the risk for cavitation. However in damper applications the influence from temperature variations is normally relatively small compared to the influence from pressure variations.

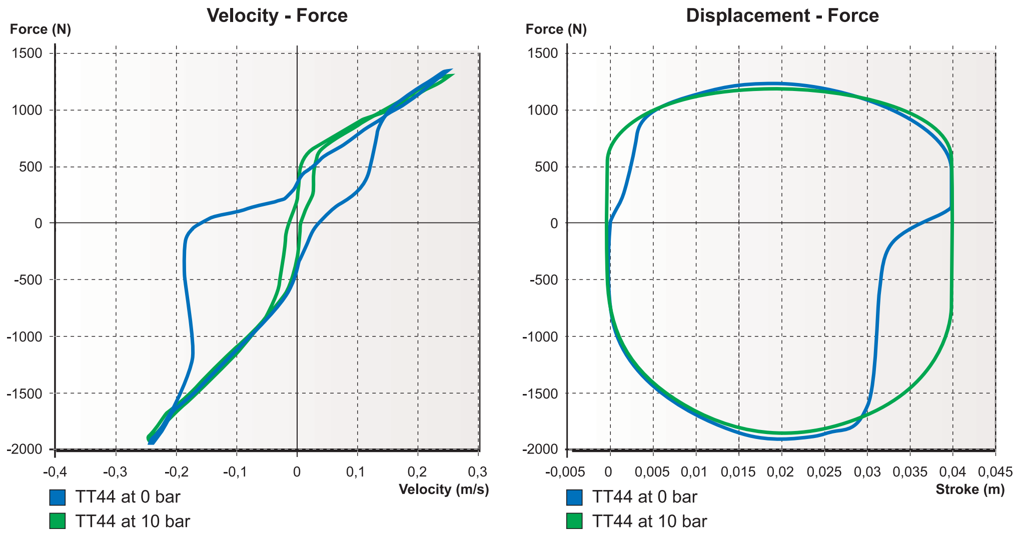


Figure 12.1 *Example of cavitation in a TT44 tested at 0 bar of reservoir gas pressure. The curves are produced at continuously force measurement at a sine wave cycle with peak velocity 0.25 m/s.*

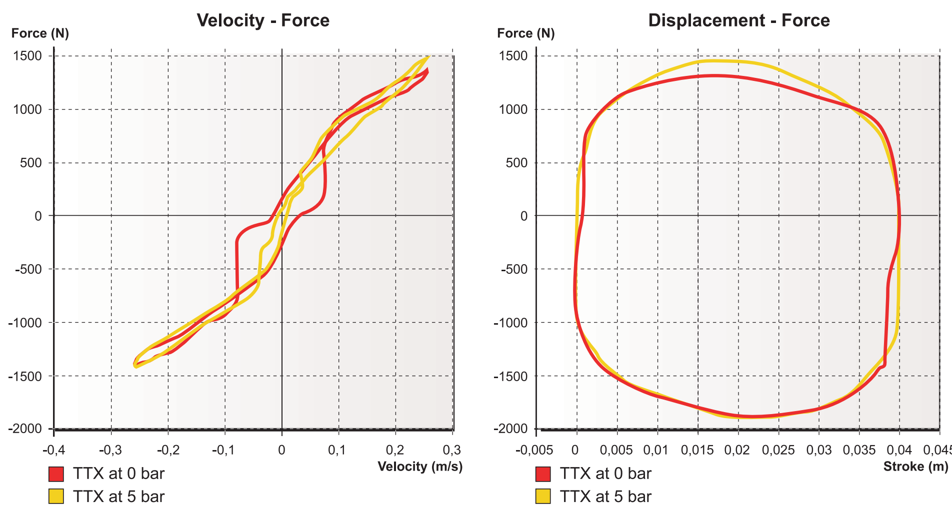


Figure 12.2 *Example of cavitation in a TTX tested at 0 bar of reservoir gas pressure. The curves are produced at continuously force measurement at a sine wave cycle with peak velocity 0.25 m/s.*

In non-TTX type dampers, pressure drops are found at the backside of the main piston and at the outlet of the valves. If the volume of fluid where the pressure drop happens is pressurised high enough, the produced gas bubbles caused by delays in the pressure distribution will disappear right away. As these small bubbles only occur very locally in direct proximity of the piston/valves this phenomenon is sometimes called “local cavitation”. However, cavitation normally refers to conditions where the pressure is low enough to make the small gas bubbles combine to large volumes of gas. If these volumes of gas pass through the valves instead of damper fluid, the damping will go away.

When the gas bubbles are exposed to higher pressure, they will implode. These implosions could cause damage to damper, so called *cavitation pitting*. Often sounds from the implosions can be noticed as a hissing or gurgling sound. As most dampers use shimmed main pistons, damage from cavitation is normally found at the outlet ports of the pistons. The damage is very dependent on the geometry of the piston.

In a conventional damper, the cavitation starts during the compression strokes. If the absolute pressure (see below) on the rebound side drops too much during a compression stroke, gas bubbles will occur. This leads to a stop in pressure drop on the rebound side. The volume increases as the oil changes from fluid to gas and oil from the compression side is dumped into the reservoir. Any additional damping force will come from increased pressure on the compression side. On the following rebound stroke, there will be a lack of oil on the rebound side and this will lead to a delay in the rebound damping force build up.

Most conventional dampers have some type of reservoir valve. This valve is often referred to as a compression valve, but the main purpose of this valve is to improve damper response and reduce the risk of cavitation. By adding a restriction in the reservoir, the absolute pressure on the compression side will be higher for a specific compression damping force. This means that the pressure on the rebound side does not have to drop as far to achieve the same pressure differential across the valve. This shortened pressure drop allows that differential to happen in less time, which equals response time, and also keeps the rebound pressure away from the cavitation limits. This may decrease the risk of cavitation, but it adds hysteresis to the damper.

In a conventional damper with no reservoir valves, the available damping force before cavitation can be estimated from the formula below.

$$\boxed{F_{max\ comp} = p_{reservoir} \bullet A_{rebound}}$$

$p_{reservoir}$ = reservoir gas pressure (not absolute pressure)
$A_{rebound}$ = area of the rebound side of the main piston (piston area minus rod area).

Always try to stay with the recommended gas pressure and keep the extra margin against cavitation by adding reservoir damping.

In the TTX, the risk of cavitation is always minimised. Check valves prevent the pressure from dropping below the set gas pressure. This results in always having the same good margin for cavitation, no matter how high the damping forces are. Cavitation in the TTX can only occur if the oil isn't pressurised. Even a tiny pressurisation of 1-2 bar (15–30 psi) is enough. There will be no pressurisation of the oil if

- The reservoir isn't pressurised
- There is too little oil in the damper.

If the oil level is to low, the separating piston won't be pushing on the fluid and can't add any pressure to the damper fluid. The reason why some pressure has to be added to the damper fluid to avoid cavitation in the TTX is that there will always be small areas with pressure drops in channels and check valves.

The easiest way to study cavitation in a dynamometer is to reduce the reservoir gas pressure. Continuous measurement is the best way to identify cavitation, as the acceleration and deceleration part of the damping curve separates due to the delayed damping force build up. In a conventional damper the delay will mainly be noticed on the rebound side. While in a TTX the delay will very much less. In a TTX there will be about the same delay on compression and rebound. Even with constant velocity or peak velocity measurement, cavitation can sometimes be observed. However it will be more difficult to know if the loss in damping force actually is due to cavitation or some other factor.

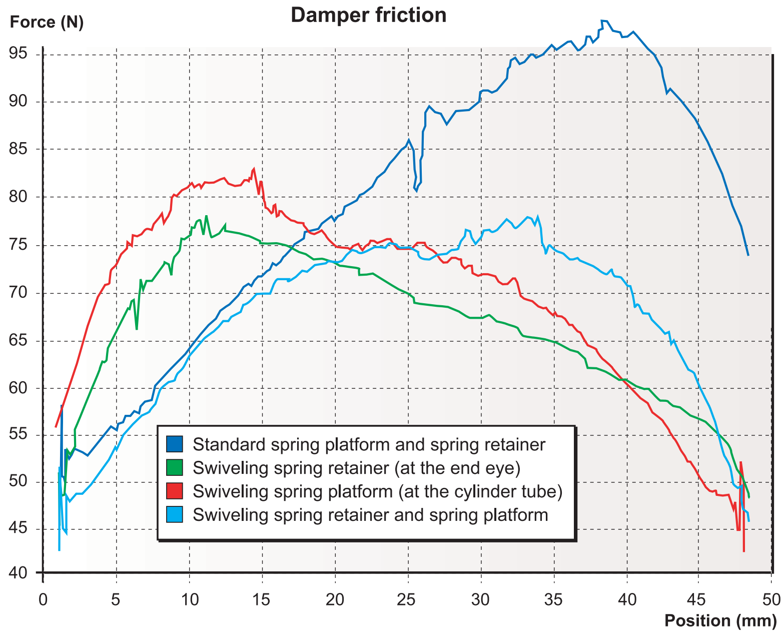


***Figure 13.0** Damper friction verses stroke in a TT44. The higher the position number is, the more extended the damper is.*

13 Friction Levels

For some period in history, damping came mainly from friction. For example, leaf spring friction on trucks gives damping. As the damping characteristics of friction are maximum force at the start of the movement and normally with little or no relation between friction force and velocity. Friction doesn't result in a desired damping characteristic for a vehicle. The level of friction is also hard to control. As racing teams always try to optimise everything to gain performance, the levels of friction can not be neglected.

It is normally hard to quantify the level of friction in the suspension as it depends on the loads. When it comes to the dampers, the spring element used is a determining factor. Coil over springs always cause bending forces to the dampers, as the springs are not perfect. Here the main friction comes from the bushings. The design of the TTX will guarantee minimum friction forces when subjected to side loads (See section *Piston rod guide/seal*).

There are always some variations in friction between individual dampers. After the dampers have been run in, the friction will be reduced.

Without side loads, the friction levels of a TTX could be expected to be in the region of about 15 - 17 N in sliding friction and about 22 - 24 N in starting friction. One reason for these low values is the low gas pressure. Another reason is that the separating piston doesn't move as the piston rod displaces oil. Therefore, there is no friction from the separating piston.

If side loads are added, it is hard to give some exact numbers of how the TTX perform friction wise compared to for example a TT40, as is very much depends on the spring used and the position of the piston rod, but a reduction of friction with about 50% can easily be found.

Many times but not always, a swivelling spring retainer and/or spring platform reduce the friction from side loads caused by coil springs. In the figure below shows how the friction force varies over the stroke in a TT44 with different combinations of spring retainers and spring platforms. The best combination depends on the position of the piston.

NOTE:

The figure here is only for illustrating the trends for a non through rod damper. The numbers are only true for this particular hard ware combination.

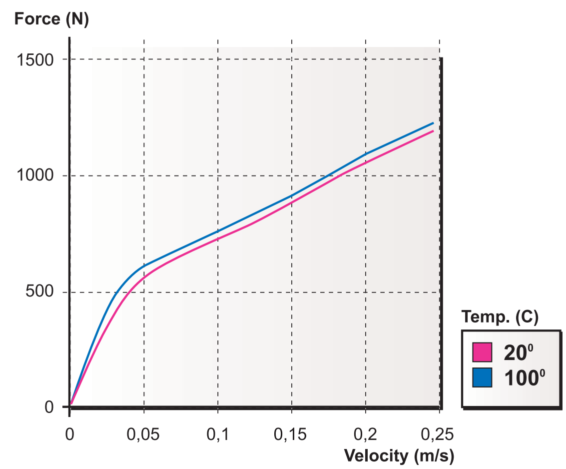


Figure 14.0 *Example on how the damping force can drop in a TTX if the temperate raised from 20º C to 100º C.*

14 Temperature Stability

In all dampers, the damping force changes due to temperature variations. As the viscosity drops when the temperature rises, the damping force will drop unless there is some “compensation”. The word *fading* is often used for this loss of damping. The compensation is normally done by using materials in the bleed needle and housing, so the opening of the bleed path is reduced as the temperature goes up. An enclosed volume of fluid can also be used to push the needle closer to the seat. The lengths of the two parts have to be correct to get an acceptable compensation. However the compensation can never be perfect and any overcompensation is objectionable.

How much focus there is on fading varies. On most applications, the temperature rapidly reaches a constant operating temperature. In single seaters, there is generally a fairly low amount of energy absorbed by the dampers. Therefore, the temperatures are normally very stable – in the front they stay low and in the rear they are kept high by the engine. See the chapter *Temperature range* for recommended temperatures. In applications were a lot of energy is absorbed by the dampers, fading might be more of a concern.

The amount of fade will depend on the type of flow through a valve. Flows are divided in to two different categories – laminar and turbulent. In general, laminar flow occurs where the area change is gradual and the restriction is extended in length. Turbulent flow results from an abrupt area change. Also, the velocity of the fluid moving through the valve has an influence on the type of flow. Increased velocity increases the amount of turbulent flow. At laminar flow, the restriction depends on the viscosity, while turbulent flow is not as sensitive to viscosity changes.

The drop in damping force is not linear with respect to temperature and depends on the fluid used. The Öhlins 309 oil for instance drops little in viscosity above 50ºC.

It is very hard to give correct values on how much fading there is in a damper as there are huge variations, both in force and in percentage of force, due to settings and velocity. The settings include both hardware and how the adjusters are set. Expressed as percentage of drop in damping force, fading will always go down as the velocity increases. In general, compared to a TT44, the TTX fades more at lower damper velocities while the TT44 drops more at higher velocities.

NOTE:

For simulations, keep in mind that the actual damping force you have on the car might be lower than what the dynamometer testing at room temperature indicates. For this reason dynamometer testing for this purpose should be done at operating temperature. When only comparing settings, dynamometer testing can be made at room temperature if it is always done this way.

Figure 15.1 *Jason Bright, Ford Performance Racing.*

15 Damper Functions

by Bruce Burness

Historically, dampers were asked only to provide a comfortable ride. If you were lucky, driver controllability was enhanced at the same time. With the advent of ground effect aerodynamics in the late seventies, racing engineers discovered that damper settings are a valuable tool for optimising "aero" effects. At the same time, the tire companies found they needed to redesign their tires to take advantage of the downforce created by the new ground effect aerodynamics. The mechanical grip of these new tires also turned out to be extremely sensitive to damper settings. These developments doubled the number of duties required of dampers of today. The priority list today for racing damper functions is "areo" management, mechanical grip, tire wear, driver controllability and ride comfort. Dampers have a powerful influence on the performance of your car.

These five damper functions are all interrelated but at the same time optimising one of these functions can sabotage another. A compromise between function goals is many times unavoidable. Finding the most effective compromise is the overall goal and will pay dividends on the racetrack.

Comfort, grip and control

These five goals are so tightly interwoven that most of the time it is very difficult to make a damping change and then properly assign the performance gain or loss to the correct category. For example, let us say you have added some extra compression damping to the front dampers and now the front tires have gained grip. The question is did we create pure mechanical grip from the tires or is the gain from improved aerodynamics or from better dynamic fore/aft pitch control or possibly a higher dynamic ride height or centre of gravity which could also change dynamic roll centres. It is not essential to know the exact cause and effect, but it is possible through a cleverly planned sequence of subsequent tests to better isolate the gain and assign it to the correct category. If this is achieved, the focus of further testing will be more on target and the possibility of a wayward theory will be minimised.

All this may sound too hypothetical but rest assured if you optimise the aerodynamic potential without compromising the grip and then find the mechanical balance by adjusting the springs, sway-bars, etc. the driver controllability will most likely be there automatically. The ride comfort may be compromised but do not be too concerned. Even though Öhlins dampers generally produce an improved ride quality, we have found that damper settings that give too much comfort cannot provide optimum grip or controllability.

Aerodynamic management

In this ground effect age, dampers can maximise the amount of downforce generated by the underside of the car by assisting in maintaining a constant air gap between the underside and the ground. Today we have basically two types of formula car ground effect configurations, *tunnels* and *flat bottoms***.** With both types the clearance between the underside and the ground is very critical. Generally there is a ride height ”sweet spot” that is favourable for generating high downforce with a minimum of ”aero” drag. The problem is that this ”sweet spot” is very close to the ground. Good damper settings will keep the car at this ideal ride height a higher percentage of the time through most dynamic conditions without allowing the underside to contact the ground (bottoming).

With both tunnel and flat bottom cars the centre of the downforce is found where the geometry of the underside comes closest to the ground. With either type of car the centre of downforce migrates with any change in pitch angle in relation to the ground caused by braking, cornering or acceleration. This migration of the centre of downforce alters the handling balance by increasing the downforce towards the direction of migration and reducing the downforce away from the migration. Therefore, added tire grip will occur at the end of the car that moves closer to the ground.

Tunnel cars have far less downforce migration than flat bottom cars because the contour of the tunnel is curved in the shape of a venturi with a raised entry that curves down to a short flat area followed by a long, slowly enlarging exit. Tunnels are generally positioned near the vehicle centre of gravity. The tunnel flat part is in the closest proximity to the ground and that is where the centre of downforce occurs. When the car pitches fore or aft this part of the tunnel primarily rocks back and forth and does not raise or lower significantly. Tunnels minimise downforce migration.

Flat bottom with diffusers

On the other hand, flat bottom cars with diffusers can have the downforce migrate from just ahead of the diffuser at the back all the way to the tip of the nose under braking. For cars with raised noses the migration will essentially stop where the underside begins to move away from the ground. Flat bottom cars are much more sensitive to static and dynamic pitch changes than ground effect cars. Damper settings for flat bottom cars therefore need to be biased more towards pitch control than the settings for tunnel cars.

Both tunnel and flat bottom cars can also benefit by keeping the underside parallel to the ground, side to side, while cornering. In this case, the downforce migrates from side to side but also will diminish substantially if the inside of the car raises away from the ground. For cars that turn only one direction as on an oval, sometimes higher corner speed can be achieved by increasing the compression damping and reducing the rebound on the right side (outside) and the opposite on the left side (inside).

Mechanical grip

Compared to aerodynamics, understanding the dynamics of tire grip is more elusive and the perceived rules change from one type of tire to another. It seems tire grip is created when the tire is pressed into the track surface enough to cause the rubber to interlock with the grain of the pavement. Not enough compression damping allows the tire to move freer and ride up on top of the pavement grain, metaphorically similar to “dry aquaplaning”. As the compression damping is increased, the tire will interlock with the pavement and grip will increase. If the damping is further increased incrementally, eventually the grip will stop improving and begin to go down. This is mainly caused by too much pressure from the suspension that overheats the tire or compresses it too much, giving unduly high tire load variations. Keep in mind that the suspension pressure the tire feels is the sum of the compression damping, the spring rate, the sway-bar rate and possibly the torsional rigidity of the chassis. If the pressure sum seems to be optimised for grip but for other reasons it is indicated that one component of the sum needs to be increased, another component may need to be reduced. For instance, a higher spring rate may be necessary to reduce fore and aft pitching. In order to make the stiffer spring work properly, the compression damping may need to be reduced. In another case, one car might have less torsional stiffness in its chassis than another. To compensate for this, the car with lower chassis stiffness will require more compression damping to make the suspension pressure sum high enough. An indicator of too much suspension pressure is controllable sliding at all speeds and all phases throughout the turns (flat sliding).

Grip and rebound damping

Grip in relation to rebound damping works in a slightly different manner. Rebound damping only occurs after there has been some compression of the damper and spring. The pavement grain constantly causes small wheel movements of the suspension system. The rebound damping controls the expansion in these small displacements. If the rebound damping is excessive, the expansion will be too slow leading to a loss of grip. This type of grip loss will be particularly noticeable in rear tire forward traction with the application of power. Cornering grip will not be as dramatically effected as forward traction.

If a lot of rebound damping is used the suspension will be dynamically pumped down which can improve the aerodynamic downforce. If there is enough ”aero” gain it can more than offset any loss of grip due to slow rebound recovery. When this approach is used compression damping is generally reduced at the same time to help the pumping down. We have seen success with this approach, but today most teams are pursuing the high compression, low rebound technique with even better results. Both philosophies have their place. It seems that in the classes where the downforce potential is much less, the proportion between compression and rebound damping leans towards less compression and more rebound damping.

Driver controllability

In most cases vehicle stability will be quite acceptable when the damper has been adjusted for optimum “aero” and grip management. Sometimes “aero” and grip need to be slightly compromised in order to adapt to the style of different drivers. In the final analysis a car that is more driver friendly will prevail over a car with ultimate grip that is also nervous.

Tire wear

Sometimes settings that are good for qualifying can be too hard on tires after a lot of laps. Our experience suggests slightly more compliant damper settings for the race than those used during qualifying.

Ride comfort

One final word about ride comfort. Harshness is either from a suspension that is too stiff to comply with bumps or from a suspension that shakes because of inadequate damping. Deciding which condition exists in your car plus a review of your damper settings can guide you in solving harshness problems.

Figure 16.0 *Öhlins Racing AB, Upplands Väsby, Sweden.*

16 Factory Damper Setting

Racecar set-ups as well as track conditions can vary in an endless number of ways. There is no information available about the optimum damper set-up for just your car. However, to help you, Öhlins has installed a valve configuration that seems to be one of the most useful and results in a huge adjustment range.

The valve components, the same on both compression and rebound, consists of

- 04104-02 Valve seat ∅12
- 04103-06 Poppet valve
- 04107-04 Spring c=40 N/mm, t = 2.20 mm
- 01415-14 x 3 Nose shim, 0.15x14

In the first batch of dampers, the poppet valves and springs were different. These dampers are marked TTX NE02**0** and TTX NE04**0**. In these dampers, the parts used are

- 04103-04 Poppet valve
- 04134-04 Spring, c = 40 N/mm, t = 2.40 mm

See section *High speed adjusters* for more information.

For quality control, all Öhlins TTX dampers are dynamometer tested at least once before they are delivered to the customer. They are first tested at the Öhlins factory in Sweden. At the test at the factory, all dampers are set the same. The clicker positions are set to

LSC/HSC: 10/30
LSR/HSR: 10/30

NOTE:
As the compression and rebound valving are the same, the compression and rebound forces turn out the same.

Figure 17.0 *K-Mart Racing in the V8 SCCS was one of the first touring car teams to test the TTX.*

17 Damping Guidelines

It is not our intention in this chapter to cover all questions about how to find a good damper set-up. However, we offer some basic rules to help you set up the car.

The first time the TTX is installed on the vehicle our recommendation is to match the damping curves previously used. However, as the TTX will have at least as quick damping force build up as the damper you are replacing, you might have to lower the damping force on the TTX to find a better set-up.

If there is no history when it comes to damping forces, some calculations, simulations or shake rig testing might be needed to find the correct amount of damping. If "aero" isn't taken in account, there will be no information about the distribution compression to rebound damping. Also, the damping curves may need to be reshaped from the maximum grip level to achieve an acceptable vehicle handling. Here experience is extremely useful.

In general, our recommendation is to start with linear damping curves and add knee if necessary.

When comparing damping curves, keep in mind that the motion ratio must be the same. The damping forces at a different motion ratio could be estimated by scaling the forces with the square of the quotient of the motion ratios. If the damping curves are not very linear, you have to do a more proper calculation by scaling both the forces and velocity with the quotient. One of them should be multiplied with the quotient the other divided. See the end of this chapter for some more detailed information. Having intimate knowledge of your damper mounting geometry is the key to predicting the proper amount of damping forces.

If your racecar has handling problems, determine first if it is damper related or not. Because dampers have proved to have such a profound influence on handling, some race engineers are in the habit of tuning the dampers before making "aero" or mechanical adjustments. As damper manufacturers, we are flattered, but there are limits to the problems that can be solved by damping adjustments. If a problem can be improved by "aero" or mechanical changes, it is wise to make those changes first.

To make improvements, it is important to understand the function of the dampers. Then through testing, learn how the dampers influence the handling of your car. When making adjustments, keep notes, make adjustments one at a time and in small steps. Always pay attention to changes in conditions like tire wear, track temperatures, time of day, etc. At the end of the test session, go back, if possible, to the starting set-up to double check that an improvement has actually been achieved.

We recommend limiting changing of the low speed adjusters to steps of no more than 6 to 7 clicks at the time. Too large a change can jump right over the optimum setting and sometimes result in similar handling as the original setting. We normally recommend changes of 3 to 4 clicks. When you are near the optimum setting the driver can notice such a little change as 1 or 2 clicks. When both compression and rebound are near optimum a final adjustment might require a trade of one less rebound click for one more compression click or the reverse.

A logical reason for opening only the compression low speed adjuster could be a desire to reduce harshness, to slow down turn-in, or to search for more mechanical grip. The limits to how far the low speed adjusters can be opened are instability, bottoming, lazy turn-in, not enough roll support, braking problems or loss of grip.

Opening the low speed rebound adjuster usually results in more grip especially in the rear during power-down conditions.

As the low speed adjusters are changed, the knee will occur at a different velocity. This affects the high speed forces. The more slope there is on the high speed, the more the high speed will be affected. This can be compensated by changing the high speed adjusters to raise or lower the knee.

Raising the knee (mainly on compression) can result in more support. Raising the knee can also be an effective way to control the underside rake angle, in either the front or rear. Lowering the knee can reduce harshness.

The more pronounced knee you have the more feedback the driver will get. There is often a trade off between feedback and grip/traction. With the TTX the nose shim stack is the main parameter for reshaping the knee without changing low or high speed.

Reducing the compression slope might be called for on bumpy street courses if your car has difficulty absorbing bumps causing harshness. You might want to increase the compression slope if the car bottoms easily or if roll support seems inadequate. This could also be advantageous on bumpy circuits where bumps cause big chassis movements.

Also, if you determine the knee needs to be reduced, increasing the slope at the same time is sometimes a good idea. The converse is also true.

To compensate for motion ratio changes, both high and low speed have to be changed.

It is not easy to have some general rules about in what range the adjusters should be set, as the needs can be very different depending on the vehicle. However for race cars with damper/wheel bell crank (rocker) ratios of around 1.0/1.0, the low speed compression (LSC) very often ends up at a clicker position in the range of 6 to 14. The corresponding range for the low speed rebound (LSR) is 15 to 25.

If your bell crank ratio moves the damper slower than the wheel, the clickers will need to be set to lower numbers to give more low speed damping. The reason is the wheel has mechanical leverage over the damper and the forces from the damper will end up less effective at the wheel. In addition, the leverage also causes the damper piston speed to be lower. Thus the original damping must be multiplied by the change factor and then the new damping force must be moved to a lower piston speed this time dividing the speed by the same factor. To compensate for the leverage changes, tighten the low speed adjuster and add preload with the high speed adjuster.

Conversely, if your bell crank ratio moves the damper faster than the wheel the adjusters need to be set to higher numbers to give less damping and the damping change factor now needs to divide the original damping and multiply the piston speed.