

Contents

| | |
|--|-----------|
| 1. Introduction | 4 |
| 2. How the damper works | 5 |
| 3-way configuration compression cycle | 5 |
| HSC configuration compression cycle | 7 |
| 3-way configuration rebound cycle | 7 |
| HSC configuration rebound cycle | 7 |
| 3. Damping curves terminology | 8 |
| 4. External adjustments | 9 |
| External adjusters summarized | 9 |
| Low speed adjusters | 11 |
| Reservoir compression adjuster | 11 |
| High speed compression adjuster | 11 |
| Hydraulic spring preloader | 13 |
| 5. Internal adjustments | 14 |
| Piston | 14 |
| General shim information | 14 |
| Sealing shim | 15 |
| Ring shim and centering shim | 15 |
| Preload shim | 16 |
| Clamp shim | 16 |
| Valve stop | 16 |
| 6. Shim stack theory | 17 |
| Stack stiffness | 17 |
| Stack preload | 17 |
| Knee and slope guidelines | 18 |
| 7. Hysteresis | 19 |
| 8. Combining main piston and reservoir damping | 21 |
| 9. Damping functions | 23 |
| 10. Factory recommended damper set-ups | 25 |
| 11. Damping guideline | 26 |
| 12. Work section | 27 |
| Service tools, oil & grease | 27 |
| Main piston reshimming, basic method | 28 |
| Main piston reshimming, alternate method | 34 |
| Reservoir reshimming | 35 |
| Optional HSC adjuster installation | 38 |
| Optional HSC adjuster spring change | 41 |
| Hydraulic spring preloader refilling | 41 |
| Gas pressure | 41 |
| Routine maintenance | 41 |
| 13. Adjustment and valving charts | 43 |
| 14. External dimensions and damper identification | 51 |
| 15. Optional new parts | 53 |
| 16. Spare parts | 54 |
| Damper body | 54 |
| Piston shaft | 55 |
| End eye | 55 |
| Reservoir | 56 |
| HSC adjuster | 57 |
| Hydraulic spring preloader | 58 |
| Main piston shims | 59 |
| Reservoir shims | 59 |
| 17. Addresses | 63 |



The Öhlins TT44, the winning damper in formula, sports and touring car racing.

1. Introduction

Congratulations on choosing the Öhlins TT44 formula car shock absorber, the most unique and powerful racing damper available today.

The TT44 damper design is the culmination of two decades of Öhlins successful participation in world championship events. This damper draws on all the expertise developed by Öhlins while winning more than 60 World Championships. The TT44 damper is designed to handle the demanding damping characteristics needed for all types of tracks, from street courses to super speedways. The ability to create high damping forces at very short strokes, combined with powerful adjusters, will give you outstanding performance and offer many new possibilities.

The Öhlins TT44 features a patented concept with a unique double wall design and two adjustable bleed valves to control the flow between these tubes. These valves control the initial compression and rebound damping and are check-valved to be completely independent of each other. They meter the oil flow created by the main piston area,

not the flow created by shaft area displacement. This translates to low internal pressure during the compression stroke.

Even though damping force builds rapidly, the low internal pressure prevents high friction from the shaft seal. The result is excellent short stroke/high force performance.

The temperature stability is maintained by using a flow restriction design in the bleed valves that create a turbulent flow at very low piston velocities. Materials with different thermal expansion rates are used to compensate for the viscosity change of the oil caused by changes in temperature.

Thanks to the unique design of the bleed valves (they are not tapered needles working in a fixed orifice) every step (click) of the adjusters produces equal and predictable changes in force in the normal operating range. Optimum settings are easy to find.

The Öhlins shim system with the ring shim preload device offers infinite combinations of shim stacks of very huge spectrum of different character with one

and the same piston.

The reservoir has a traditional adjustable needle bleed valve in parallel with its own shim stack, which works only during the compression cycle. The whole system is pressurized by nitrogen gas behind a floating piston to ensure separation of the gas and oil.

As an option there is an additional compression control adjuster available, providing a total of 4 external damper adjusters. This pressure regulating valve, unique in many ways, controls only the pressure over the main piston. It gives new possibilities in shaping damping curves. For instance "double knee" curves can be produced easily.

In keeping with Öhlins long tradition of perfection, performance is outstanding and long life is to be expected.

The Öhlins TT44 damper is a racer friendly damper, easy to set up, dial in, service and rebuild.

All dampers are dyno tested before they are delivered to the customer.

Support is always available from the Öhlins factory and Öhlins distributors worldwide. □

2. How the damper works

The following description of the function of the damper is divided into four different situations: compression and rebound damping cycles with and without the high speed compression (HSC) adjuster.

The compression damping cycle describes the situation when the shaft-piston unit moves into the damper body, while the rebound damping cycle describes the situation when the shaft-piston unit withdraws from the damper body.

3-way configuration stands for the basic TT44 damper with three external adjusters. The HSC configuration refers to the basic TT44 damper with the optional external high speed adjuster installed. It gives a total number of 4 external adjusters. Dampers with this configuration are often called "4-ways".

3-way configuration compression cycle

For a start we assume that the adjustable reservoir compression bleed valve (figure 1-E) is fully open. In this case, the pressure on the compression side of the piston (figure 1-G) will remain almost constant and the same as the gas pressure during the whole stroke, though some small change of the pressure will occur because of the change of gas volume caused by shaft displacement.

As long as the reservoir compression bleed valve is set to fully open, only the pressure drop on the rebound side of the main piston (figure 1-H) causes the oil to move from the compression side to the rebound side of the piston. This means that the initial nitrogen pressure in the reservoir has to be high enough to handle the compression forces. If the pressure is too low there is a risk of dumping more oil than the shaft displaces into the reservoir and cavitation will occur in the rebound chamber.

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. In the initial part of a compression stroke, when the velocity of the piston is low, the oil flow bypasses the main piston by travelling through the adjustable low speed compression bleed valve (figure 1-A), the compression check valve (figure 1-B) and then flows between the two tubes (figure 1-C). The oil re-enters the main tube on the rebound side through a port near the end cap.

As the piston velocity increases the pressure drop across the main piston will increase. Observe that it is the pressure

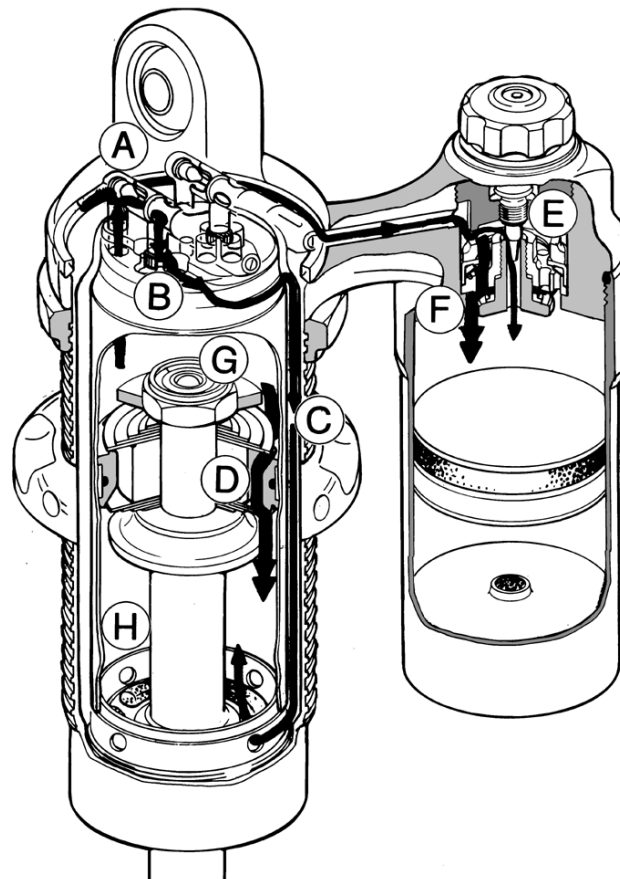


Figure 1. Oil circuit. 3-way configuration during compression cycle.

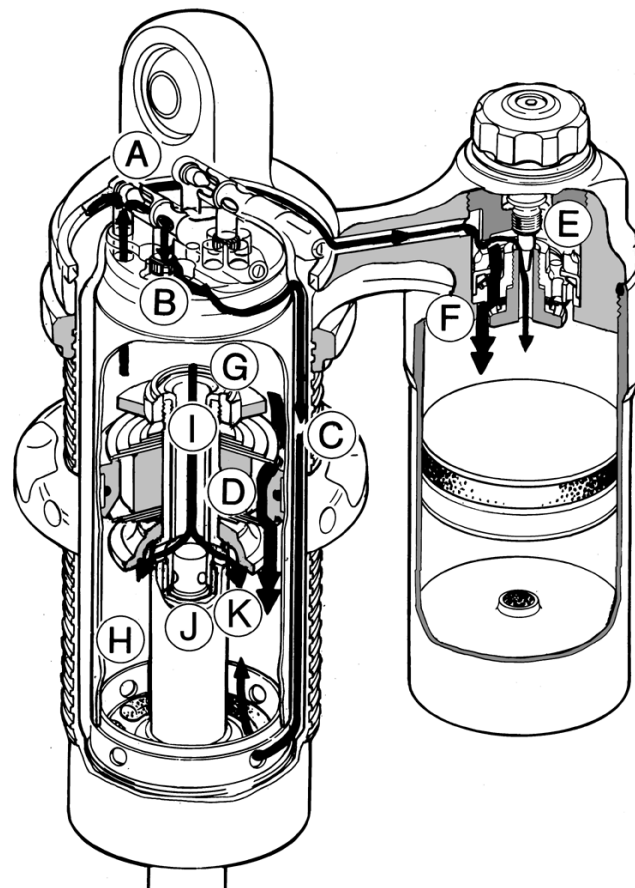


Figure 2. Oil circuit. 4-way configuration during compression cycle.

2. How the damper works

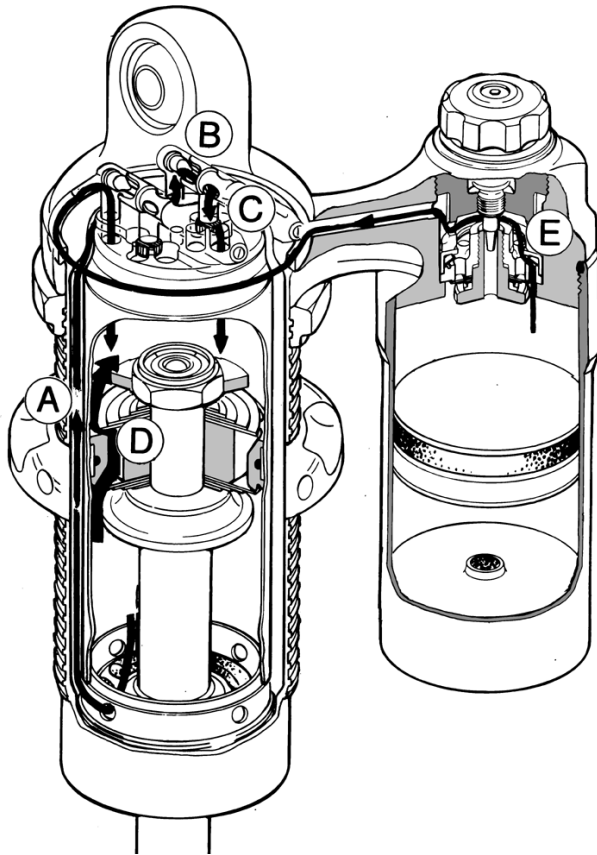


Figure 3. Oil circuit. 3-way configuration during rebound cycle.

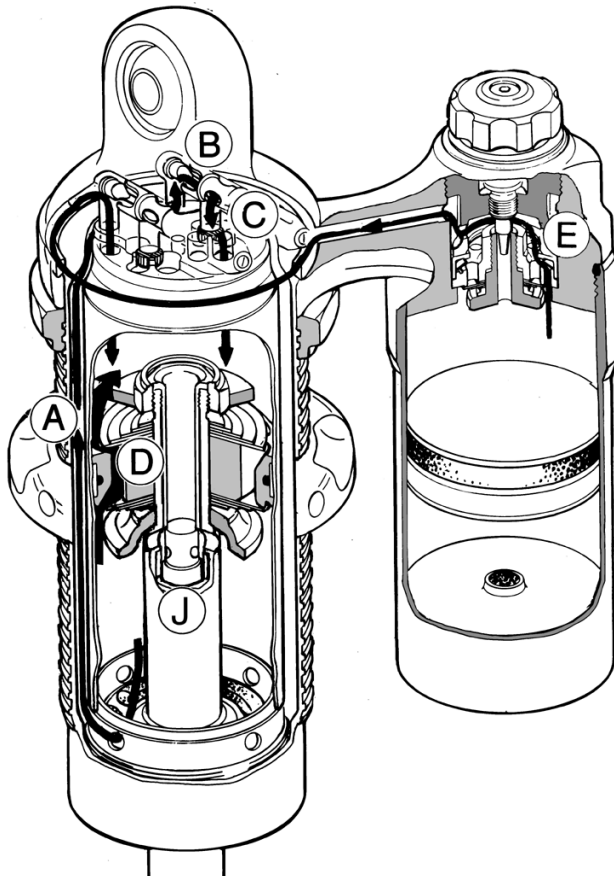


Figure 4. Oil circuit. 4-way configuration during rebound cycle.

drop on the rebound side of the piston that causes this, not the increase in pressure on the compression side. Depending on the combination of shims, the compression shim stack (figure 1-D) opens at a certain pressure. This provides a direct pathway for the oil, allowing it to pass through the stack to the other side of the piston. Depending on the shim stack, the opening will be something from abrupt to gradual. As long as the piston is moving and the bleed valve is not fully closed, some oil will always flow through the bleed valve.

Note: In practice many strokes never reach a velocity high enough to cause enough pressure drop across the main piston necessary to cause the shims to open.

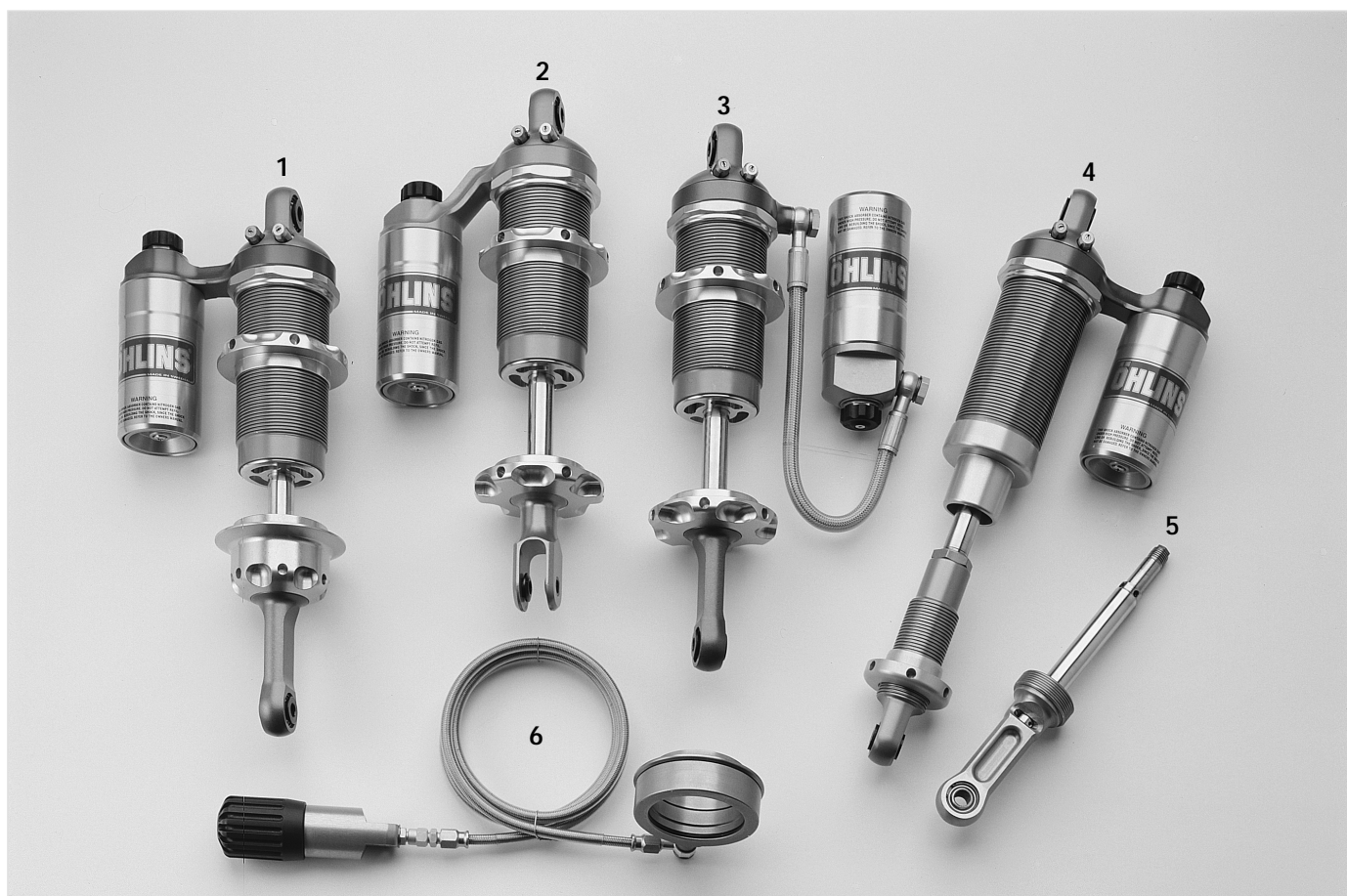
During the compression stroke, the oil displaced by the volume of the piston shaft as it enters the main body is forced into the reservoir. This causes a small increase in system pressure due to the piston shaft volume displacement moving the floating piston and thereby compressing the nitrogen.

In the passage connecting the main body and the reservoir, there is another compression valve system (one bleed valve in parallel with a shim stack) similar to the system found in the main body. By closing off the compression reservoir bleed valve (figure 1-E), the reservoir valve system contributes to the total compression damping.

The reservoir compression bleed valve regulates the flow at low piston velocities. At higher piston velocities, the reservoir compression shim stack (figure 1-F) opens to provide a pressure blow off.

The pressure differential across the main piston will build up quicker if there is some damping from the reservoir, because there is not only a pressure drop on the rebound side of the main piston but also a pressure rise on the compression side. This also reduces the risk of dumping oil into the reservoir. However, the amount of reservoir damping needed to achieve a quick damping force build up is small. Excessive reservoir damping will cause hysteresis.

High hysteresis is caused because the damping force from the reservoir is created by the pressure acting only on the cross section area of the piston shaft. For equal damping force the pressure needed per unit of damping is much higher compared to the pressure needed from the much larger main piston. Higher pressure compresses the oil more, even the best hydraulic oils have a certain compressibility, and also expands the damper slightly more. In a dyno graph this can be seen as a difference in force during acceleration compared to deceleration. The force will be lower during acceleration than during deceleration. This is most evident at low velocities. The



Öhlins TT44 in different configurations: 1. "straight piggyback", 2. "offset piggyback", 3. "remote reservoir", 4. "straight piggyback" in 36 mm spring configuration, 5. optional HSC adjuster, 6. hydraulic spring preloader.

force differential between acceleration and deceleration is referred to as hysteresis. One of Öhlins design goals for the TT44 damper is to have the possibility to run the damper with low hysteresis. See chapter "Hysteresis" on page 19.

Because of the restriction in the reservoir, the pressure in the whole damper body will rise. This will reduce the risk of cavitation.

The compression damping curve is the sum of the forces created in the main body plus those created in the reservoir. See chapter "Combining main piston and reservoir damping" on page 21.

HSC configuration compression cycle

What has already been described here about the 3-way adjuster compression cycle is still valid with the addition of the HSC option, but now there is also a poppet valve in parallel with the compression shim stack of the main piston. The poppet valve is pushed against its seat by a preloaded coil spring. The amount of preload can be externally adjusted. This preload determines what pressure differential across the main piston is necessary to make the poppet valve open.

The HSC configuration provides an additional pathway to the other side of the piston. The new pathway goes through the center of the shaft (figure 2-I)

and then encounters the poppet valve (figure 2-J). As the piston velocity increases, the pressure drop across the main piston rises. At some velocity the movement of the piston creates a pressure drop across the main piston, that equals the predetermined pressure required to open the valve.

After passing the valve, the oil exits the cross drilled holes (figure 2-K) in the shaft, on the other side of the piston. Depending on the combination of shim stack and poppet valve preload, the poppet valve can open before, at the same time, or after the main piston shim stack. This determines the character of the damping curve.

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

3-way configuration rebound cycle

This cycle is very much the same as the compression damping cycle (3-way configuration) of the main piston.

During the rebound stroke, the pressure of the rebound side of the main piston is increasing, while the pressure of the compression side is kept almost constant. This causes the oil to move back across the piston. When the piston velocity is low, oil will initially flow between

the two tubes (figure 3-A) and arrive at the low speed rebound bleed valve (figure 3-B).

After the valve has metered the flow, the oil will open the rebound check valve (figure 3-C) and travel to the compression side of the piston.

When the opening pressure of the main piston shim stack is reached, oil will also pass through the stack (figure 3-D).

To compensate for the displacement change caused by the shaft leaving the body, a check valve in the reservoir (figure 3-E) will open so that oil can freely return to the compression side of the piston. This will cause a small decrease in system pressure, due to the reduction of piston shaft displacement moving the floating piston and therefore expanding the nitrogen.

HSC configuration rebound cycle

The rebound damping cycle is the same whether the damper is of 3-way or HSC type (figure 4).

The poppet valve of a HSC adjustable damper (figure 4-J) has the function of a check valve. It will seal against the valve seat during the rebound stroke and therefore prevent oil from flowing back through the shaft. The oil passage will become identical to the standard 3-way type damper during the rebound stroke. □

3. Damping curves terminology

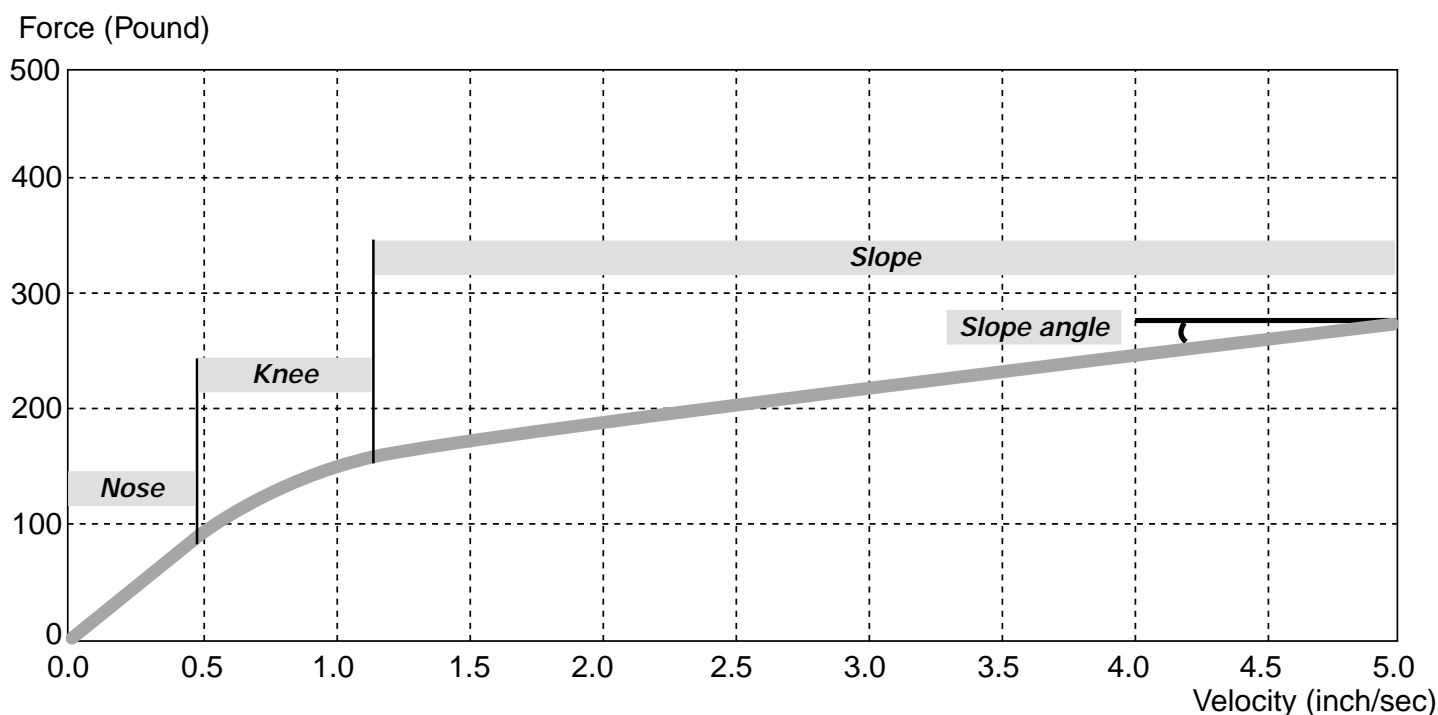


Figure 5. Terminology.

In order to understand the next part of this manual we must all speak the same language. In the damper industry there are some slang words used to describe different zones to interpret damping curves. 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 dyno curves. Later chapters will show how to manipulate the damper to produce alternate dyno curves. This chapter is relevant for both compression and rebound damping curves.

Note: In this part of the manual the speeds and forces shown are not actual and just chosen for illustration purposes.

Later on, these values will be real where adjustment and valving combinations are catalogued. See chapter "Adjustment and valving charts" on page 43.

Because a high percentage of race teams have and use Roehrig damper dynamometers, Öhlins has chosen to format all graph figures in this manual in the same proportion as the latest Roehrig dyno graph printouts, and in the chapter "Adjustment and valving charts" the figures are of the exact size.

There are different ways of presenting dyno curves produced by a Roehrig dyno.

Note: Unless otherwise indicated, the curves shown are compression open (measurement during acceleration) and rebound closed (measurement during deceleration).

To explain the damping curve terminology it is not necessary to include the damping in the reservoir or the influence of the optional high speed compression adjuster at this time.

Take a look at figure 5 and notice the first portion of the damping curve that starts at 0.0 inches/second damper speed and ends at about 0.5 inches/second. This speed zone 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) and is often called bleed. The design and size of the bleed determines the character and shape of the nose.

See "Low speed compression and rebound adjusters", on page 11 for further information.

The finish of the nose zone coincides with the beginning of the knee zone. This point is determined by the initial opening of the piston shim stack. Its location in the curve can be found by identifying where the upwards curve first begins to level off into the radius that transitions into the straight line (in this case about 0.5 in./sec.).

The knee portion extends until the shim stack has transitioned from closed to open (in this case about 1.2 in./sec.). 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 low speed adjustment in combination with

the shim stack's properties determines the position and shape of the knee.

The slope, also referred to as high speed, is determined by a combination of the shim stack stiffness and the size, shape, quantity and placement of the piston ports adjacent to the underside of the shims. In most cases the slope will continue rising in a straight line to damper speeds well beyond those found on most racing dyno charts. Eventually the slope will end and the curve will again turn upwards. This happens when the size of the piston holes begin to restrict oil flow (piston holes 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).

See "Knee and slope guidelines", on page 18 for further information.

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

Note: A nose, a knee and a slope can also be identified in the reservoir damping system. This system is of the same type with a fixed orifice valve in parallel with a shim stack. See chapter "Combining main piston and reservoir damping" on page 21.

If the high speed compression adjuster is used a second knee can be achieved. See "High speed compression adjuster" on page 11.

The speed of the damper refers to the speed of the shaft movements, not to the speed of the car. □

4. External adjustments

Most dampers that are externally adjustable have some type of low speed adjuster. Low speed adjusters are almost always externally adjustable variable orifices that become fixed orifices after adjustment. In the damper industry, these low speed orifices are often referred to as bleeds.

Unlike shim stacks, bleed orifices do not change size in response to changes in pressure. Because oil will always travel to 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 damper shaft 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 adjustment, when it comes to external high speed adjusters, 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 tolerance zones of the parts have to be reduced to keep the precision 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 summarized

The TT44-damper from Öhlins comes normally as a 3-way externally adjustable damper. The HSC adjuster is then sold separately as a kit. However, for some cars the dampers are delivered with the HSC adjuster already mounted.

- **Low speed compression damping adjuster (LSC)**

Type of adjuster:

Bleed adjuster.

Effects:

The flow from the main piston during compression strokes only.

Identification:

Gold knob at the head of the cylinder body.

Number of positions:

Approx. 38

- **Low speed rebound damping adjuster (LSR)**

Type of adjuster:

Bleed adjuster.

Effects:

The flow from the main piston during rebound strokes only.

Identification:

Silver knob at the head of the cylinder body.

Number of positions:

Approx. 38.

- **Reservoir compression damping adjuster (RC)**

Type of adjuster:

Bleed adjuster.

Effects:

The flow from the displacement of the shaft during compression strokes only.

Identification:

Black knob at the top of the reservoir.

Number of positions:

Normally within 20-25. Depends on the shim stack used.

- **Optional high speed compression damping adjuster (HSC)**

Type of adjuster:

Poppet valve preload adjuster.

Effects:

The flow from the main piston during compression strokes only.

Identification:

Gold wheel in the end eye.

Number of positions:

Approx. 55.

Note: All Öhlins external adjusters, including the optional high speed compression adjuster, are "fully hard" when turned clockwise until they stop.

It is very important that the clicker position is always counted from "fully hard". The reason is "full hard" is always an absolute position. "Fully soft" will vary depending on tolerances. In the case of the reservoir, "fully soft" will also vary with the shim stack chosen and in the case of the optional high speed compression adjuster the detents will become gradually less pronounced.

Normally the first click and/or detent is counted as "zero" position.

To match damping curves of a pair of dampers in the dyno, sometimes the clicker numbers will end up +/-1 click from each other.

Because of the high number of clicks of the HSC adjuster, the click position numbers can differ even more.



Figure 6. Standard external adjusters. - = soft. + = hard.



Figure 7. Optional high speed compression damping adjuster. - = soft. + = hard.

4. External adjustments

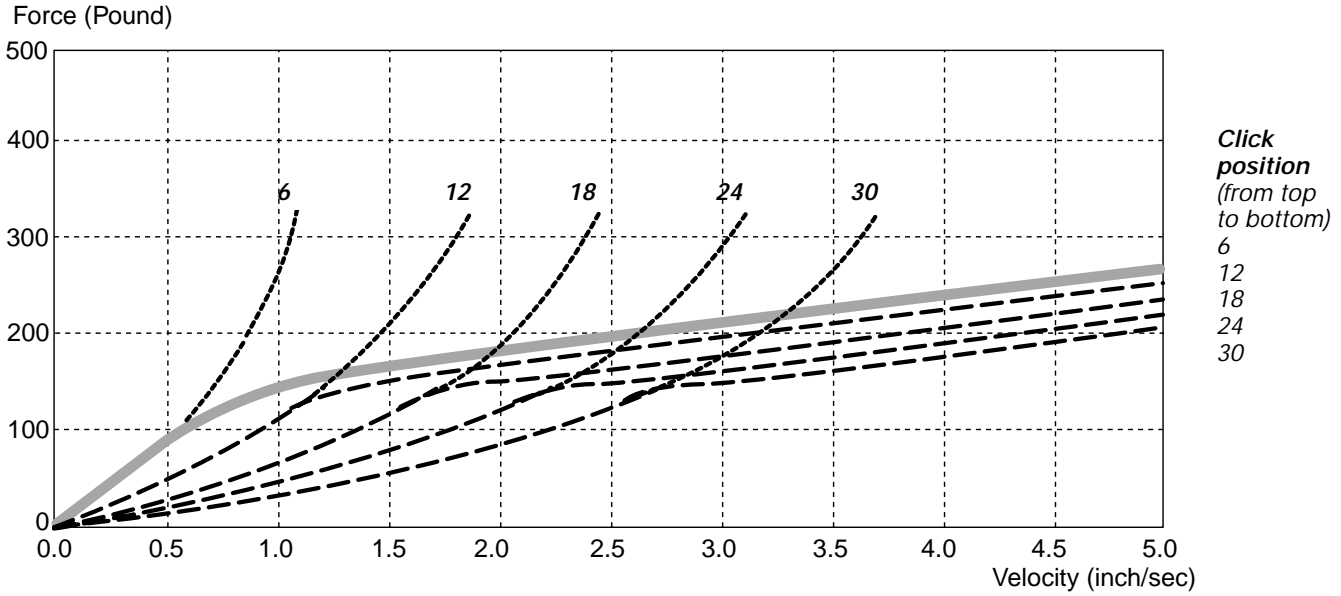


Figure 8. Influence of the low speed adjusters. The graph represents both the LSC and the LSR.

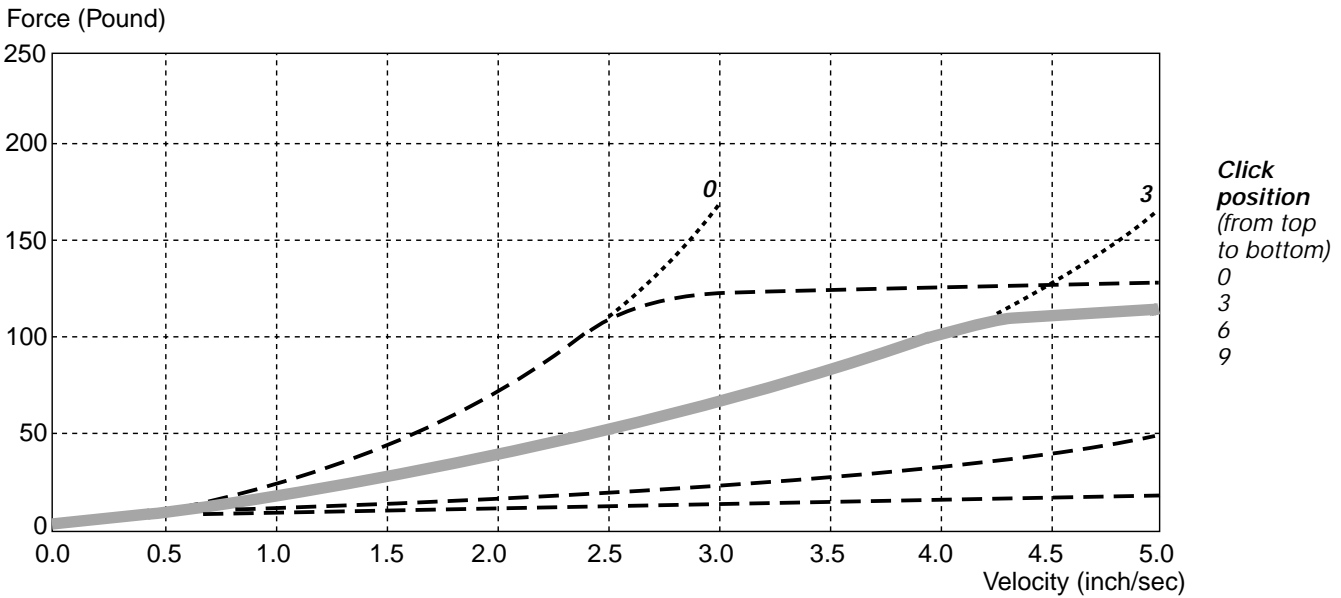


Figure 9. Influence of the standard reservoir compression adjuster (2.5 mm type). The graph illustrates the reservoir damping only. **Note:** Different scale.

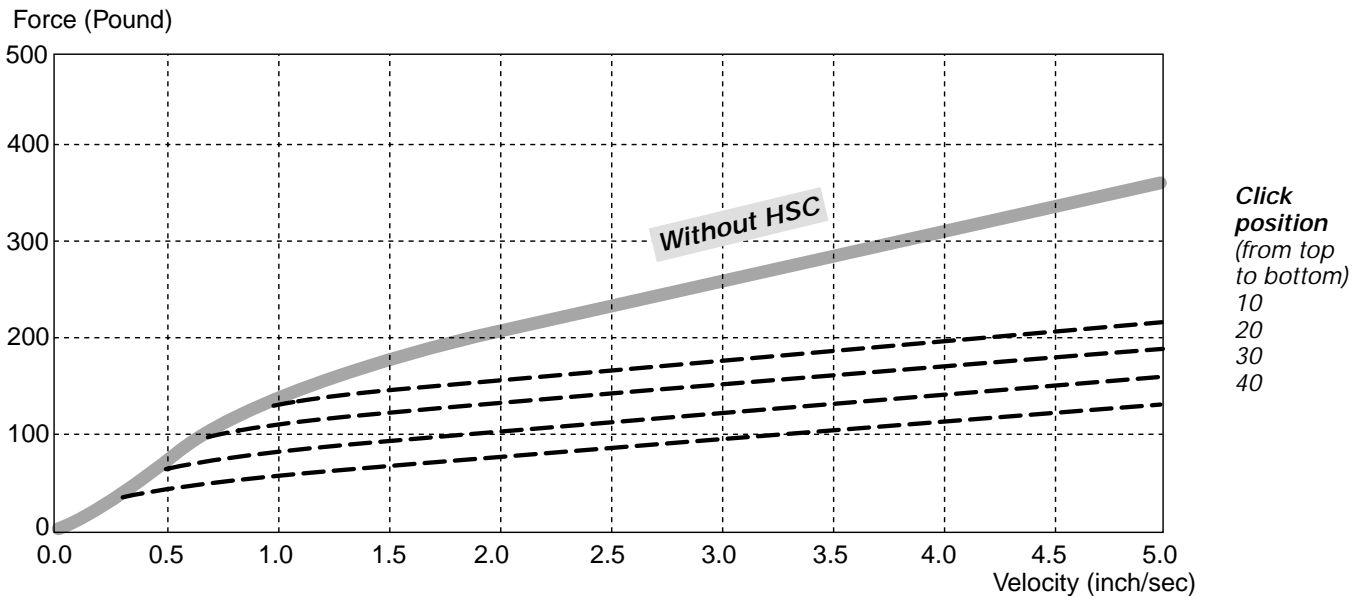


Figure 10. Influence of HSC adjuster (1 mm wire spring). Here LSC is set to click position 10. No oil passes the main piston compression shim stack. **Note:** The base curve is different due to a different shim stack.

Just remember maximum clockwise is “full hard” for all adjusters.

Low speed adjusters

The two low speed adjusters LSC and LSR have the same design and are uniquely designed so that in the normal operating range each click of the knobs will change the damping in equal steps. They are not tapered needles working in fixed orifices, where the damping force increases progressively per click as the needle is closed. The adjusters are powerful over the whole range making it easier for you to find optimum settings.

The LSC and LSR have left-hand threads and the knobs move away from the body when adjusted clockwise. Do not let the outward motion mislead you. All the way out is full hard!

Both knobs have a range of approximately 38 clicks. To match a pair of dampers in the low speed area, the click positions shall not differ more than 2 clicks.

The knobs can be adjusted either by hand or with a small screwdriver. Do not use too much torque when closing the bleeds completely.

As the adjuster is turned counter-clockwise the clicker numbers get higher.

Temperature stability is maintained in the low speed area because of the unique design of the bleed valves, which creates a turbulent flow at very low piston velocities. Also materials with different thermal expansion are used to compensate for the viscosity change of the oil caused by temperature changes.

Depending on the situation, different starting settings are recommended. Generally it is better to start with the adjusters a little more open and gradually close them off. Öhlins recommend setting the LSC to a click position between 5 and 15 and the LSR to a click position between 10 and 25.

Low speed adjusters effect on damping curve

Assuming the LSC and the LSR are adjusted the same and the piston velocity is the same and that there is no oil flowing through the shim stacks, exactly the same amount of oil will flow through both valves. The LSR has exactly the same effect on the nose of the rebound damping curve as the LSC has on the nose of the compression damping curve.

Figure 8 shows the influence of the LSC. As described above, it could just as well be a graph showing the influence of the low speed rebound adjuster. In the graph the original baseline compression curve of figure 6 can be seen together with four alternate curves. These extra curves are achieved by adjusting only the gold knob. The numbers in the graph represent various low speed adjuster clicker settings. The dotted curves pointing up-

wards indicate the theoretical shape of the curves if there was only a bleed valve and no high speed shim stack to open. See chapter “Adjustment and valving charts”, on page 43 for real values.

As the bleed is opened more and more, the damping is reduced. The speed at where the knee begins increases and the nose is stretched longer and longer. Notice, the force at where the knee and slope begins is always the same. The slopes also remain parallel to each other. This is because the shim stack determines the knee and the slope forces and we have not changed any shims yet.

Conversely, if you wish to keep the slope starting speed constant, the shim stack preload has to be reduced every time the low speed adjuster is opened.

Reservoir compression adjuster

The RC is the reservoir counterpart to the gold knob on the main body. It also adjusts bleeds and therefore the low speed damping of the reservoir. It is a tapered needle working in a fixed orifice.

The needle valve is available in two different sizes, one for the 1.5 mm needle seat and one for the 2.5 mm needle seat. The 2.5 mm needle/needle seat is standard. For more information about the 1.5 mm needle, see chapter “Optional new parts”, on page 53.

The RC has a standard right-hand thread so it moves into the reservoir body when turned clockwise. In this case “all the way in” is full hard.

This adjuster has a range of approximately 20 clicks. The full range varies depending on the reservoir shim stack chosen. The total thickness of the shim stack effects the number of click positions, as the needle seat is also the bolt that clamps the shims. Different stack thicknesses create different needle seat heights. The thinner the stack is, the less click positions there will be. The lost clicks will be on the full open side of the range. However, even with the thinnest stack possible, there are still plenty of usable clicks available.

The reservoir adjuster shall be adjusted only by hand.

Your designated starting clicker setting can also vary greatly with different style of shim stacks. Generally a good starting position is with the RC with the 2.5 mm needle at click position 8 to 12 for non-preloaded reservoir shim stacks and click position 1 to 5 for preloaded, low blow off shim stacks. There will be more specifics about the different type of shim stacks when we move on to chapter “Internal adjustments”.

Reservoir compression adjuster effect on damping curve

Figure 9 shows the influence of the RC. In the reservoir there is a valve system very similar to those inside the main

body. But it only operates during the compression cycle and meters only the oil driven to the reservoir by the piston shaft displacement. It works in series with the main compression system. The compression damping curve becomes the sum of the damping created in the main body plus the damping created in the reservoir. See chapter “Combining main piston and reservoir damping”, on page 21.

The reservoir valves are however not identical to the main valves and therefore create damping curves with slightly different character. Here, even if the bleed is closed, the pressure build up is very much delayed because of hysteresis. See chapter “Hysteresis”, on page 19 for more information.

See chapter “Adjustment and valving charts”, on page 43 for real values.

High speed compression adjuster

In order to extend the performance of our TT44 damper we offer this additional compression control adjuster. With this feature we now have three distinct compression adjustments and combined with the existing rebound adjuster there are now a total of four adjusters. This modification has just a few new parts and can easily be retrofitted to most existing TT44 dampers.

The concept for this new option is to provide race team engineers and technicians with more external control over the compression damping to provide more usable time during practice and qualifying and to provide a system to shape damping curves in a way not otherwise possible. Even if the HSC adjuster is used, this does not eliminate the need of reshimming the compression shim stack to optimize the setting.

The main parts required to convert or add this HSC adjuster are:

- *New end eye incorporating a window to give access to the adjustment wheel.*
- *New hollow shaft to house the poppet valve and adjuster mechanism.*
- *Threaded insert valve seat.*
- *Bronze valve guide.*
- *Poppet valve.*
- *Valve spring.*
- *Valve.*
- *Adjuster mechanism with threaded seal cap and affixed adjuster wheel.*

No other part than the shaft and the end eye, need be replaced. Installation is quite easy. By following the steps described in chapter “Optional HSC adjuster installation”, on page 41, you can avoid pitfalls.

The HSC changes the preload of a poppet valve in parallel with the compression shim stack. A coil spring pushes the poppet valve against its seat. The preload of the spring determines the

4. External adjustments

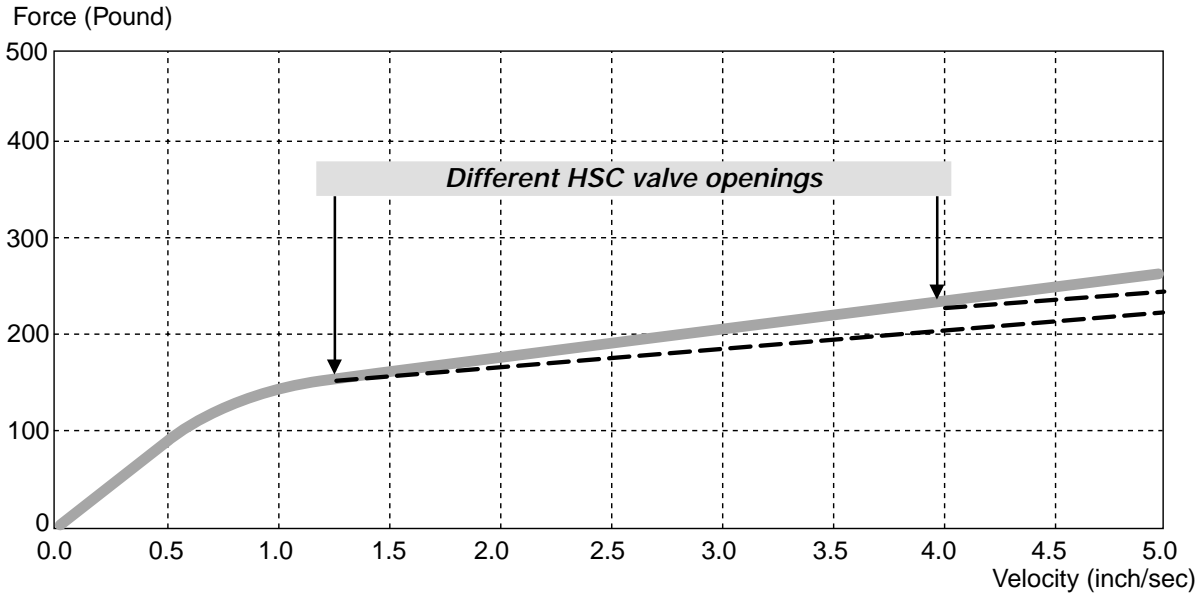


Figure 11. Situation A. The original compression shim stack is used and opens first followed by the HSC adjuster valve opening.

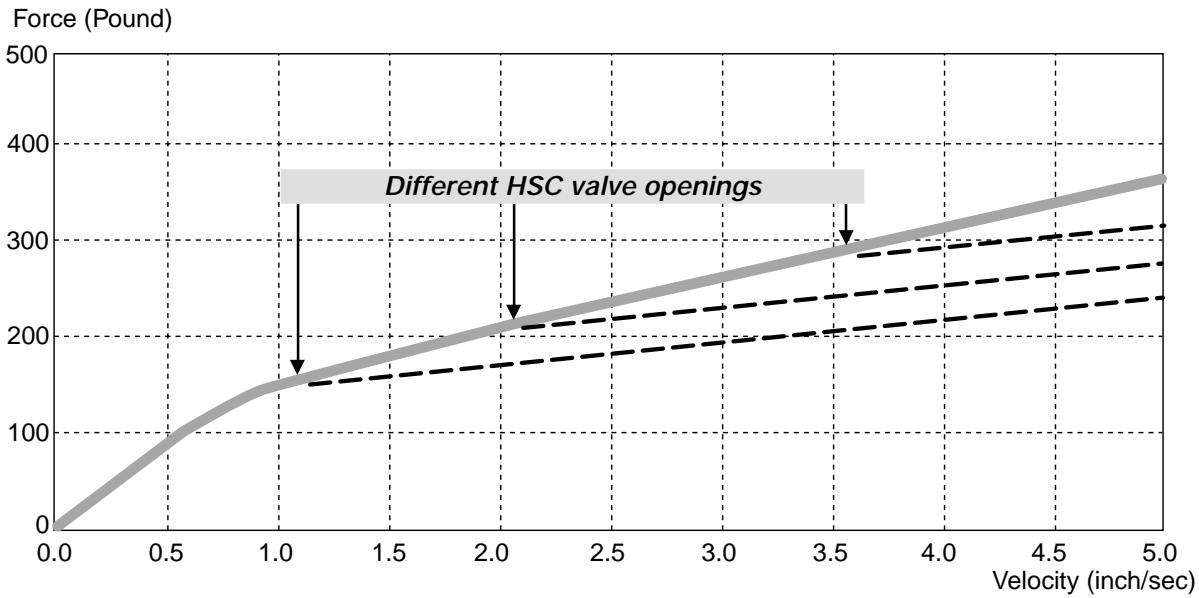


Figure 12. Situation B. A modified stiffer compression shim stack is used and opens first, followed by the HSC adjuster valve opening.

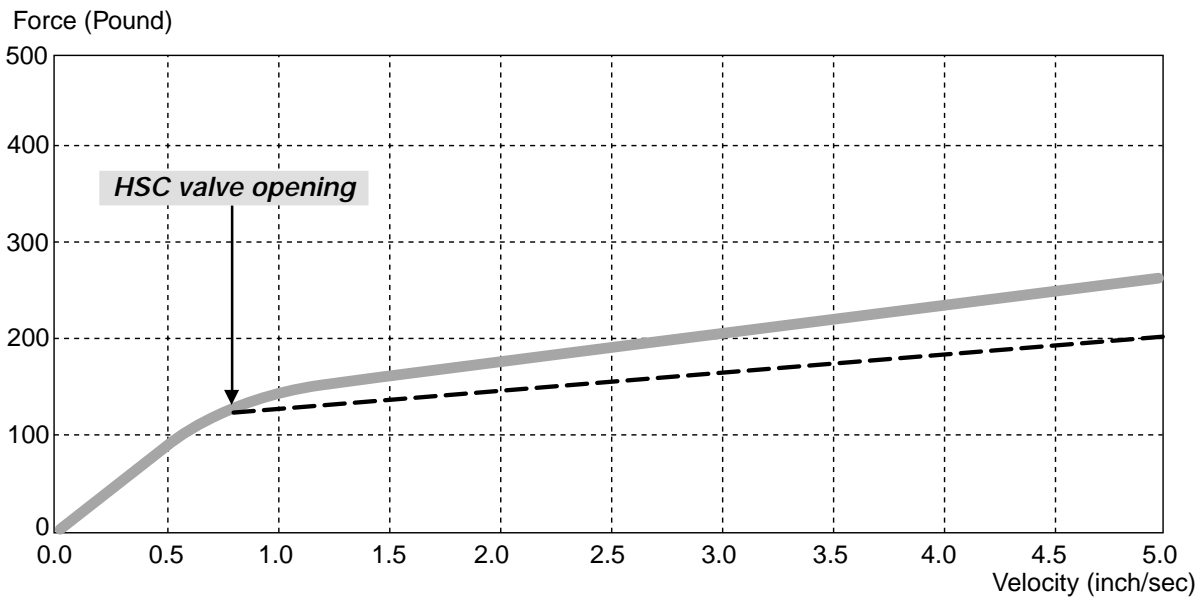


Figure 13. Situation C. The HSC adjuster valve opens simultaneously with the original compression shim stack.

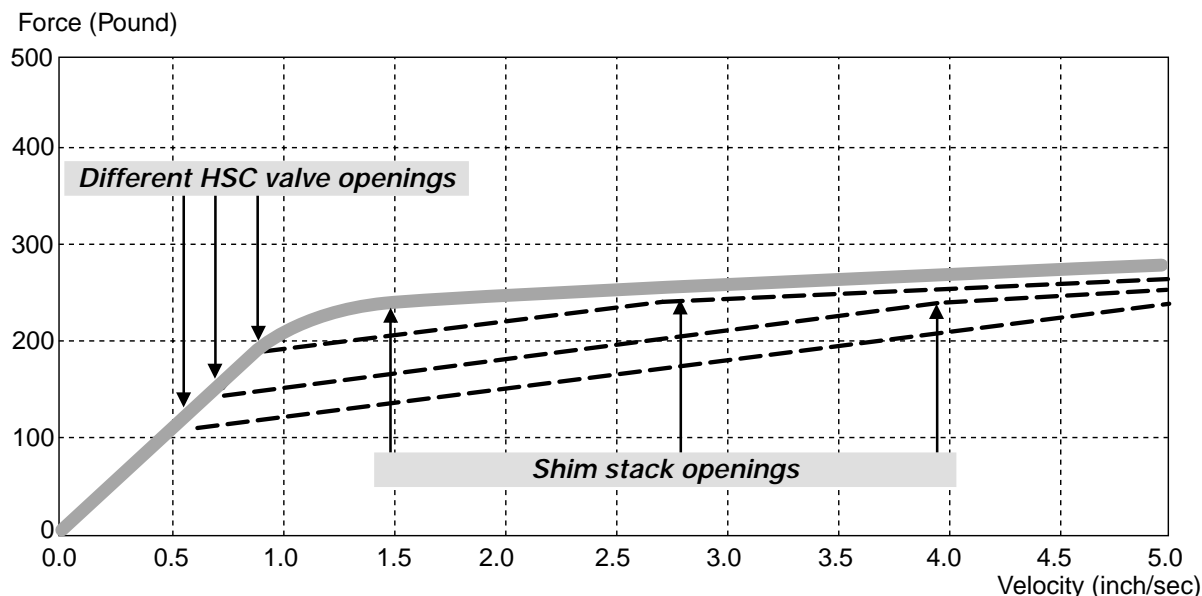


Figure 14. Situation D. The HSC adjuster valve opens first, followed by a modified compression shim stack opening.

pressure differential to open it.

There are two different springs available for the HSC, one with 0.8 mm wire and one with 1 mm wire. The HSC adjuster is delivered with the stiffer spring (1 mm) mounted and the softer (0.8 mm) as a supplemental part.

By turning the adjuster wheel clockwise (viewed from the end of the damper at the shaft side), the preload of the poppet valve increases.

As on all adjusters on the TT44, the clicks are counted from maximum clockwise position (max preload = max force). There are a total number of approximately 55 clicks for both the 0.8 mm and the 1 mm spring. At the full soft end of the adjustment range, when the adjuster is turned fully counterclockwise, there will be a couple of turns without detent clicks. The adjuster should not be used in this area.

When the preload of the poppet valve increases, the space for the valve to move will decrease, as the preloader (part # 05464-01) moves closer to the valve. The risk of the valve bottoming against the preloader increases with damper speed and is also higher with the soft spring (part # 05473-01).

Note: To avoid bottoming of the poppet valve, the adjuster should not be set to a click position less than 10.

Turning the adjustment wheel changes the preload of the poppet valve. This is done by using a 2 mm diameter pin through the window of the end eye. A maximum of two clicks can be made per sweep.

No start setting can be recommended, as it will vary depending on type of car, ratio, shim stacks etc. The HSC adjuster blow off starting point is best set with the help of a damper dyno. We recommend click position 30 for the first “dyno” run.

Generally the blow off point is set no sooner than 2 inches/sec.

High speed compression adjuster effect on damping curve

Figure 10 shows the influence of the HSC. As described earlier, two different springs are available. By using the 1 mm wire spring you can run a larger adjustment range than you can if you use the 0.8 mm spring. The minimum force for the two springs is about the same, but the maximum force will be higher and the damping force change per click will be larger with the stiffer spring. See chapter “Adjustment and valving charts” on page 43 for real values.

There are several distinct ways to use the HSC adjuster in conjunction with the other external adjusters and the shim stacks. Keep in mind that the HSC functions in some ways exactly like a compression shim stack. Both are pressure regulators that control oil flow by opening at a pre-determined pressure thereby providing a path for the oil to flow. This additional pathway allows the oil to reach the other side of the piston with less resistance. Lower resistance always equates to lower damping force. Whether these two valves open simultaneously, or one after the other, and which valve opens first, is your option.

Following are descriptions of different methods of using the piston shim stack in conjunction with the HSC adjuster poppet valve:

- A.** The original compression shim stack is used and opens first, followed by the HSC adjuster valve opening (figure 11).
- B.** A modified stiffer compression shim stack is used and opens first, followed

by the HSC adjuster valve opening (figure 12).

C. The HSC adjuster valve opens simultaneously with the original compression shim stack (figure 13).

D. The HSC adjuster valve opens first, followed by a modified compression shim stack opening (figure 14).

All of these situations have been tested during the development. However, each race team has to decide which configuration will best suit their needs.

Hydraulic spring preloader

The hydraulic spring preloader (“weight jacker”) makes it possible to adjust spring preload from the cockpit. The rate is 0.5 mm/turn and the stroke is 6.5 mm.

The spring preloader is designed for 2” inner diameter (i.d.) springs and comes with a 1500 mm long hose. For more dimensions, see drawing in chapter “External dimensions and damper identification” on page 51.

The spring preloader can be adjusted by hand to approx. 1200 lbs (5500 N) of load, which is the maximum continuous load.

By changing some of the parts in the master cylinder, there is a possibility to rebuild the spring preloader, so you get a rate of 0.75 mm per turn. However, this will increase the torque needed for a specific load. Contact Öhlins or an Öhlins distributor for further information.

If the spring preloader needs to be taken apart for whatever reason, it is very important to get all the air out of the system when it is assembled. If not the stroke will be reduced and the flexing will increase. See chapter “Hydraulic spring preloader refilling” on page 41 for guidelines. □

5. Internal adjustments

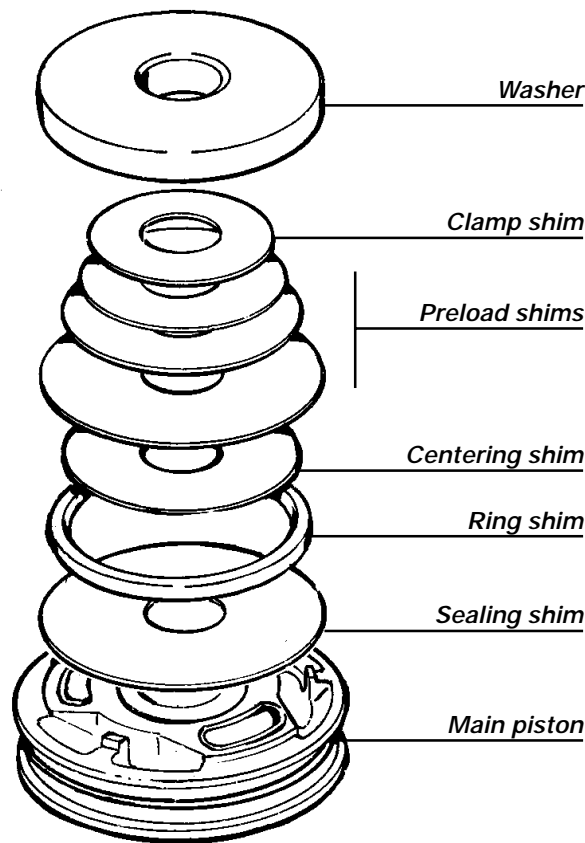


Figure 15. A typical Öhlins main piston shim stack.

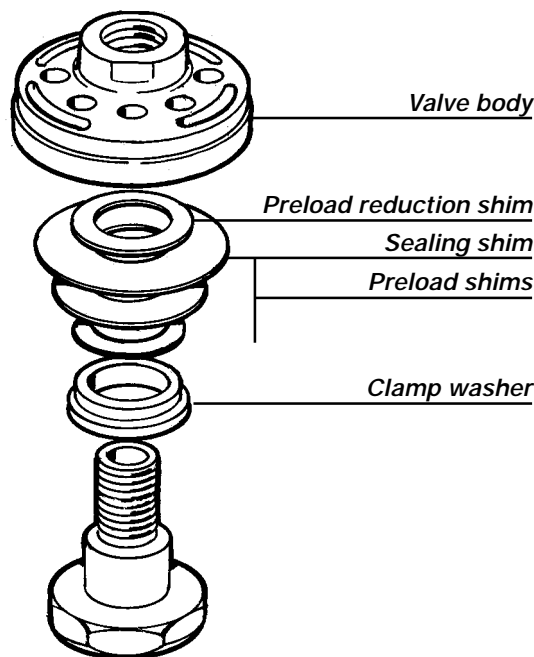


Figure 16. A typical reservoir shim stack with the standard cupped valve body (part # 01244-01).

We have just learned about the external adjusters. We now need to look inside to see what tools are available to influence the damping. (The two different springs available for the high speed compression adjuster have already been covered in "High speed compression adjuster" on page 11).

Piston

The piston (#5415-11), the heart of the damper, has three ports in each direction, is made out of sintered steel and is specially developed for racing purposes. Until the end of 1998 it was a machined steel piston (#5415-01). The function of the machined piston and the sintered piston are identical. However, the sintered piston is 25 % lighter and the tolerances are smaller making it easier to match dampers.

Thanks to the large piston diameter (44 mm), a quick damping response can be achieved.

The piston is flat on both sides providing easy control of the condition of the sealing surfaces.

The rebound side of the piston has a machined groove in the surface that has no function other than making it easier to identify the rebound side. The compression side has no groove.

General shim information

Öhlins shim stack system will offer us almost endless possibilities.

Depending on where a shim is positioned in the stack it will have different functions and effects on the damping curve. The shims are named by their position. See figure 15 and 16.

Some information is always valid, no matter what type of damping curves you are looking for.

Usually the shims in the stack will be smaller and smaller the farther they are positioned away from the piston. The farther away the shims are positioned from the piston, the less effect they will have on the initial part of the damping curve (at lower velocities).

Shims with uniform thickness and diameters that get smaller in even steps will reduce stress concentration in the shims, and reduce friction between the shims when flexing. However, this ideal is not always possible.

The shims in the reservoir are smaller than the ones on the main piston. The i.d. of the shims on the main piston is 12 mm, and in the reservoir the i.d. is 8 mm. The standard thicknesses are the same: 0.15, 0.20, 0.25 and 0.30 mm. See chapter "Spare parts", on page 54 for the different o.d.s available.

With the diameters available for the

main piston it is possible to create stacks with 2, 4 and 6 mm increments of diameter change in even steps:

- For 2 mm spacing use:
38,36,34,32,30,28,26, etc.
- For 4 mm spacing use:
38,34,30,26, etc.
- For 6 mm spacing use:
38,32,26, etc.

With some combinations it is not possible to have even spacing all the way to the clamp shim. In those cases it is desirable to reduce the spacing as the shims get closer to the clamp shim.

A similar approach is used concerning shim thickness. Although uniform thickness is desirable, many times it is impossible to achieve a particular damping force without mixing the thicknesses. If this is the case Öhlins suggests the larger shims be thinner and progressively thicker as the shims become smaller.

Sealing shim

After flowing through the piston, the oil first encounters the sealing shim. This shim closes the piston ports and must always be large enough to completely cover them. For this reason the diameter of the sealing shim is not variable. The TT44 main piston design requires a minimum sealing shim diameter of 38 mm for the compression side and 36 mm for the rebound side.

In the reservoir the sealing shim has a diameter of 18 mm.

As the sealing shim does cover the ports, it also acts as a check-valve when the direction of oil flow reverses. The thickness of this shim can be adjusted to give more or less overall damping. The number of sealing shims can also be multiplied for additional force but with some compromise in low damper speed sensitivity. On the main piston the sealing shim is never preloaded so a thickness change here affects mainly the slope accompanied by a slight change to the knee height.

When using the optional cup type main piston, see chapter "Optional new parts" on page 53, or the standard valve body (part # 01244-01) in the reservoir, even the sealing shim can be preloaded. See figure 18.

Ring shim and centering shim

The next shims are the centering shim and the accompanying ring shim. These shims are only used if preload is desired.

The outside diameter (o.d.) of the main piston centering shims is 34 mm as standard for both the compression and the rebound stack.

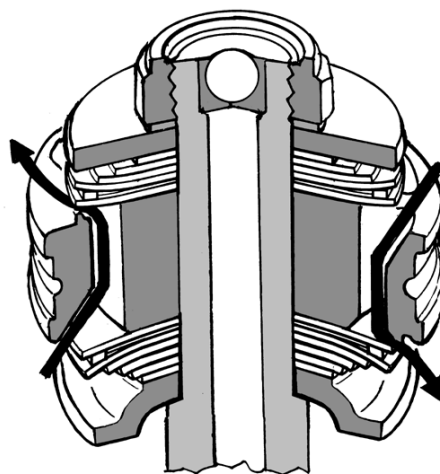


Figure 17. Cross section of compression and rebound shim stacks with ring shim configuration. The preload in the figure is exaggerated.

If a ring shim system is used in the reservoir this diameter is 15 mm. See below for a description of how the stack is preloaded if the standard valve body is used.

The ring shim is positioned concentric to the centering shim and has the same i.d. as the o.d. of the centering shim (34 mm). The o.d. of the ring shim is the same as the sealing shim. Because of the different o.d.:s of the sealing shims used on the compression and the rebound side, the two ring shims also have different o.d.:s, 38 mm for compression and 36 mm for rebound.

Ring shims with i.d. 30 mm and o.d. 34 mm are also available. In using these ring shims, their o.d. will be smaller than the sealing shim and this will result in the knee of the damping curve getting more rounded. Figure 17 illustrates preloaded compression and rebound stacks.

The reservoir ring shim i.d. is 15 mm and the o.d. 18 mm.

Ring shims are available in 0.20, 0.25 and 0.30 mm thicknesses. Centering shims are available in thickness 0.15, 0.20, 0.25 and 0.30 mm. This will give a maximum preload of 0.15 mm.

The static preload is calculated by subtracting the thickness of the centering shim from the thickness of the ring shim. Example:

A 0.30 mm ring shim minus a 0.20 mm centering shim equals a 0.10 mm preload.

As the centering shim is an additional shim in the stack the overall slope of the damping curve will be slightly increased if ring shims are used (the stiffness of the stack increases). To minimize this side effect, the same static preload can be achieved by using a thinner centering shim with a thinner ring shim, which equals the same preload. For example a 0.15 mm centering shim can be used

together with a 0.20 mm ring shim, instead of a 0.20 mm centering shim and a 0.25 mm ring shim.

If you require a higher preload than 0.15 mm, a secondary ring shim combination can be used. (This is only valid for the main piston.) To do this, another shim with the same o.d. as the sealing shim must be used as a divider between the two ring shim combinations. It is not necessary that both ring shim combinations are identical. The total preload can be calculated by combining the individual preload at each ring shim.

Note: With 0.30 mm thick preload shims we recommend that the maximum total preload is kept below 0.20 mm at the main piston and 0.10 mm in the reservoir. For thinner shims more preload is acceptable. In either case more preload can be tried but the shims need to be monitored for permanent bending.

If the standard valve body is used in the reservoir, the preload is achieved by the offset between the shim sealing seat and the center land where the preload shims are clamped (see figure 18). Nominally this offset is 0.40 mm. This distance is reduced to the wanted preload by adding a stack of shims (preload reduction shims). For this purpose use only shims with o.d. 12 mm. Example: 0.05 mm preload is built into the stack by adding 1x0.20 mm and 1x0.15 shim mm giving a total height of 0.35 mm. 0.40 minus 0.35 equals 0.05 – the desired preload.

If you require *negative preload* (a two-stage stack) on the main piston, it is possible on both compression and rebound side. Figure 19 illustrates a negative preloaded compression shim stack. Negative preload means that there is a gap between the ring shim and the following shims in the stack. These following shims are sometimes called the second stack. Negative preload is achieved by using

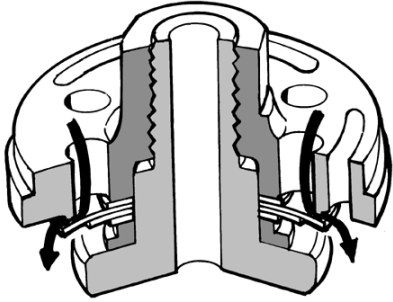


Figure 18. Cross section of preloaded reservoir shim stack with standard cupped valve body. The preload in the figure is exaggerated.

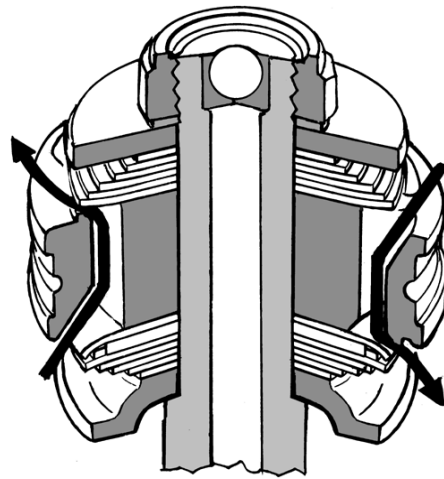


Figure 19. Negative preloaded compression shim stack with a rebound shim stack without preload. The negative preload in the figure is exaggerated.

an extra shim that together with the centering shim has a thickness exceeding the ring shim thickness. This extra shim can be placed between the centering shim and the sealing shim or after the centering shim if you wish to use the stiffness of the centering shim in the first stack. This extra shim can be of any o.d. smaller than the centering shim. The first stage stiffness is determined by the first stack only. The second stage stack will have a stiffness resulting from the whole stack. Generally a gap of 0.05 mm is enough. In some cases more gap can be used. There are two shim combinations that give a 0.05 mm gap. Both combinations use a 0.30 mm ring shim. First combination uses a 0.20 mm extra shim and a 0.15 mm centering shim for the sum of 0.35 height. Second combination uses a 0.15 mm extra shim and a 0.20 mm centering shim. The sum is again 0.35 mm. The second combination is preferable because the ring shim has more engagement with the centering shim. See also "Stack preload" on page 17 for further information.

Note: Always maximize engagement between the ring shim and the centering shim.

Preload shim

A preload shim is just as the name says always preloaded, even in its static position. The preloading force produced by the preloading shims is the result of their spring rate multiplied by the distance they are prebent. As the ring shim must be followed by a shim of the same o.d., the first preloaded shim will always have the same o.d. as the ring. Then comes a variable quantity of shims of various thickness and diameters. All the shims following the ring shim except the clamp shim will be prebent when the shaft nut is tightened because their centers will be

clamped solidly to the centering shim and their edges will rest on the thicker ring shim. The numerical amount of static preload is only accurate for shims of the same o.d. as the ring shim. All smaller diameter preloaded shims are bent less as the o.d.'s get smaller. This is why changing a larger shim has more effect than changing a smaller shim.

Quite often the majority of shims in the stack are in the preload category.

In the case of the reservoir, when the standard valve body part is used the 12 mm diameter shims under the sealing shim control the preload or lack of preload. These shims are only spacers. All other shims in the stack are bending shims. So here even the sealing shim can be preloaded. The same is true of the optional cup-type main piston with the exception of the additional clamp shim.

The static force produced by the preloading shims is determined by their stiffness times their static preload.

Clamp shim / clamp washer

The last shim in the compression and rebound stack of the main piston is the clamp shim. Unlike all the previous shims, the clamp shim never bends or moves. As its name implies, the function of this shim is to clamp the centers of all the other shims. It is really just a solid spacer and when the shaft nut is tightened it creates a solid column, the size of its o.d. through all the other shims clear to the piston surface.

This solid column determines the fulcrum that all the flexing shims bend about. The power of the clamp shim is in its variable o.d.. Increasing the clamp shim diameter moves the fulcrum out and reduces the amount of unclamped, flexing part of all the other shims. Moving the fulcrum out towards the piston ports effectively reduces the amount of

leverage the oil has on the shim, therefore increasing the damping. When reducing the clamp the opposite is also true.

Varying the clamp shim affects the entire stack. The knee and slope are changed simultaneously by varying the clamp shim. Changing the clamp shim is very useful if you want to raise or lower the damping curve without changing the characteristic of the shim stack.

Note: A maximum clamp diameter of 23 mm for compression and 26 mm for rebound is recommended.

Sometimes if there are few shims in the stack, multiple clamp shims can be used as spacers to allow the stack to open fully without contacting the valve stop. For the same reason we suggest a minimum clamp thickness of 0.30 mm.

The clamp shim diameters are available in 1 mm increments. However, the percentage of damping change for each 1 mm step increases as the diameters increase. The reason for this is that, even though the diameter steps are uniform, the percentage each 1 mm step encroaches on the remaining unclamped shims will be greater and greater. Therefore a change from a 20 mm to a 21 mm clamp will not be as significant as a change from 21 mm to 22 mm.

In the reservoir there is no clamp shim. Instead the clamp function is integrated in the stepped washer (part # 00641-01), that also has the function of a valve stop. The stepped washer diameter is 10 mm.

Valve stop

This part provides a rigid base for the shim stack to work against. Once in a while the valve stop is used to limit the amount the shims can bend. It is important that its surface is always flat and should be checked occasionally with a precision straight edge. □

6. Shim stack theory

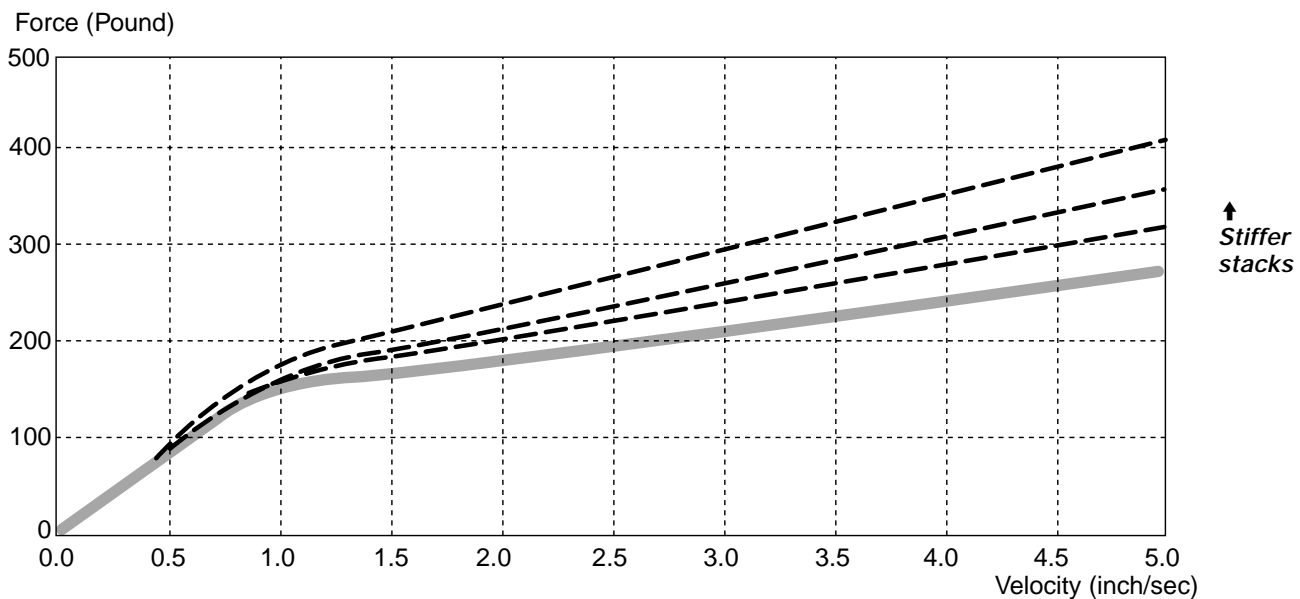


Figure 20. Influence of shim stack stiffness.

In the earlier chapters we have discussed the effect of the external adjuster on the damping curve and what function the different shims have. However, we would like to discuss further what influence different shim stacks have on the damping curve.

Stack stiffness

Shim stack stiffness is the same as shim stack spring rate. In theory the spring rate increases progressively the further the shims are flexed, but in practice the rate is more or less constant.

The stiffness is difficult to measure, therefore it is never quantified as force/distance [lbs/inch] or [N/m]. The slope of the damping curve does reveal the stiffness of the stack, therefore the stiffness is more often given in force/velocity [lbs s/inch] or [N s/m].

Note: To change the slope of the damping curve the stack stiffness has to be changed.

Figure 20 illustrates the effect achieved by altering just the shim stack stiffness. The shim stack is modified incrementally to be stiffer and stiffer. Notice the slope lines are no longer parallel. The stiffer the stack, the steeper the slope.

Changes in quantity, clamp diameter and stiffness of the individual shims all affect the stack stiffness (spring rate):

- The stiffness is directly related to the number of shims. Varying the number of shims is used to change the slope if just a small change is needed.
- The stiffness increases progressively with the increase of the clamp diameter. A change of the clamp diameter

is the most powerful tool in changing the slope of the damping curve and is only used when larger changes are needed. See "Clamp shim/clamp washer" on page 16.

- Changing the stiffness of the individual shims can theoretically be done in two different ways: change o.d. and/or the thickness. Changing the o.d. is seldom used except for fine tuning. Changing the thickness is the second most effective way of changing the slope. A lot can be gained by understanding how the stiffness is related to the thickness of the shim. Read the next few paragraphs carefully!

The stiffness of a shim is not linear in relation to its thickness. If two 0.15 mm shims are stacked together the stiffness will not equal one 0.30 mm shim. In fact it takes a little more than eight 0.15 mm shims to equal one 0.30 mm shim. To calculate how many thin shims it takes to equal one thicker shim the formula is approximately the same as for comparing the stiffness of dissimilar constant section beams. The procedure is to divide the thicker shim by the thinner shim to get the ratio between the thicknesses of the shims. Then if you cube the thickness ratio you will have an idea of how many thin shims it takes to equal one thicker shim.

Example 1:

$0.30 \div 0.15 = 2.00$
 $2.00 \text{ cubed } (2.00 \times 2.00 \times 2.00) = 8.00$
 Hence:
 $0.15 \times 8.0 \text{ shims} = \text{one } 0.30 \text{ shim}$

Example 2:

$0.25 \div 0.15 = 1.67$
 $1.68 \text{ cubed} = 4.66$
 Hence:
 $0.15 \times 4.66 \text{ shims} = \text{one } 0.25 \text{ shim}$

Example 3:

$0.20 \div 0.15 = 1.33$
 $1.33 \text{ cubed} = 2.35$
 Hence:
 $0.15 \times 2.35 \text{ shims} = \text{one } 0.20 \text{ shim}$

Because the shims do not bend exactly like constant section beams, this formula gives an answer that is not entirely correct. In reality, the equivalency factor is slightly more than the result yielded by the formula. Understanding shim equivalency is very useful if you are trying to hit specific damping forces. As you can see the equivalency factors are mostly uneven numbers. This can be used to your advantage if you need to split the difference between adding one identical shim or leaving the stack unchanged. This knowledge will also help you select the appropriate shim thickness for the damping change you are looking for.

Stack preload

The preload of a shim stack is the distance the shims in the stack are prebent by ring shims or cup pistons.

The preload always refers to the amount of prebending of the shim that is prebent the most. (Depending on the position in the stack, the shims can have different amounts of prebending. If a double ring shim arrangement is used, only the shims after the second ring shim will be bent the sum of both preloads.)

In the case of ring shims, preload is

6. Shim stack theory

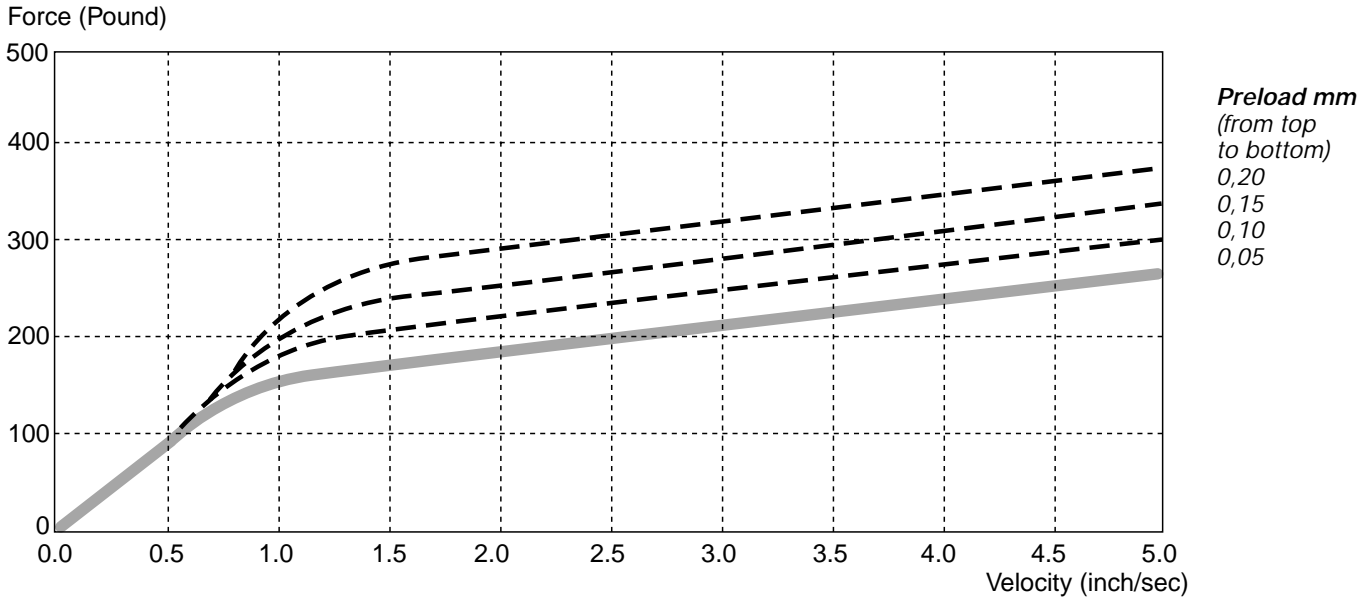


Figure 21. Influence of shim stack preload.

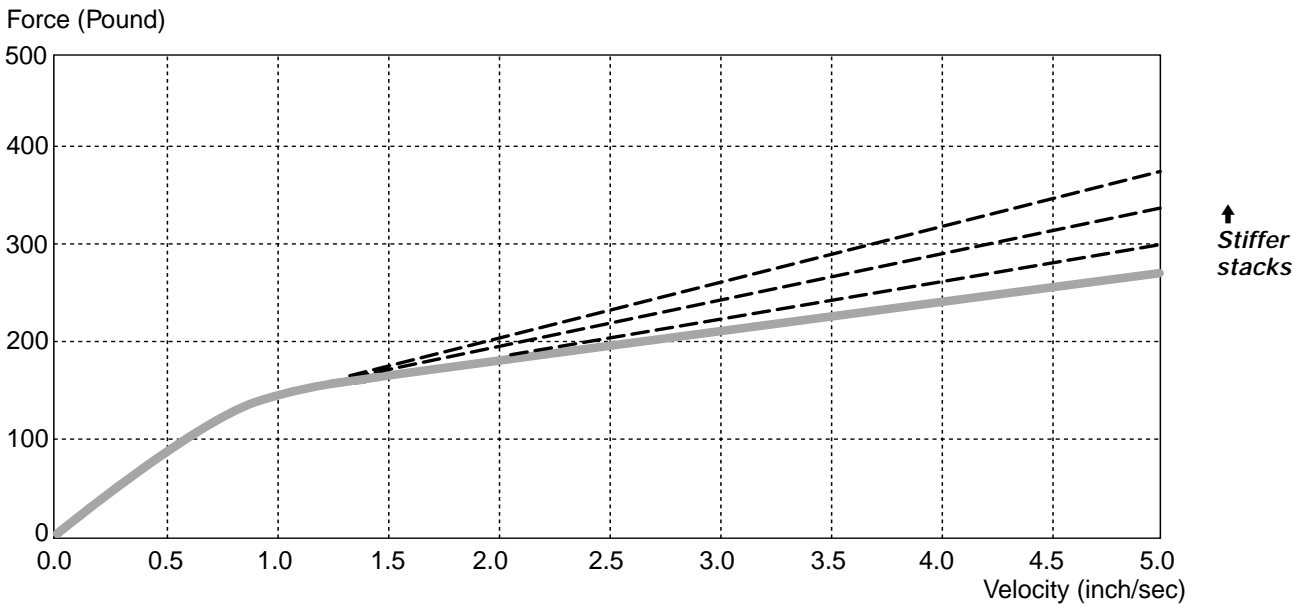


Figure 22. Influence of shim stack stiffness with preload compensation.

created when the ring shim is thicker than the centering shim. In the case of the standard reservoir valve body or a cupped main piston, the preload is achieved by the offset between the shim sealing seat and the center land where the preload spacer shims are clamped.

The term *negative preload* is used in the case of a double stack, where it tells how much the shim closest to the piston has to open until it bumps into the remainder of the shims.

Figure 21 illustrates the effect of varying only the ring shim. All the shims in the stack except the ring shim are left the same as the original baseline. The numbers shown at the right edge of the figure represent the amount of preload (the static bending on all the preloading shims with the same o.d. as the ring shim caused by the difference in thickness of the ring shim and its centering shim). Increasing the bending of the shims be-

hind the ring shim raises the pressure on the sealing shim positioned between the piston and the ring shim. This causes the knee to move higher and higher up the theoretical low speed curve. Because the knee follows the low speed curve, the knee is not only higher but starts at a slightly higher damper speed for each preload change.

The baseline curve is shown as having a preload of 0.05 mm. This could be achieved with a centering shim of 0.15 mm thickness combined with a ring shim of 0.20 mm thickness giving a differential of 0.05 mm. The first broken line could be achieved by increasing the ring shim to 0.25 mm, which gives a differential of 0.10 mm, etc.

As only the ring shim is changed, the stiffness of the shim stack will not be affected (the spring rate of the stack remains the same) and the slope lines will remain parallel to each other. However,

for extreme changes in preload, some change in the slope would be seen, as the rate of the stack is progressive.

After all this discussion about preload, it is important to remember that Öhlins offers as many non-preloaded shim combinations as those that are preloaded.

Knee and slope guidelines

The position of the knee depends on the stiffness of the stack plus the preload of the stack. The slope is controlled only by the stiffness of the stack.

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

Note: Even with zero preload there is still some knee as it still takes a rise in pressure to open the shim stack.

The zero preload knee is much less pro-

nounced than the knee from a preloaded stack, but it is still of importance.

When the stack stiffness is increased and the static preload remains constant, there will still be an increase in pressure applied to the sealing shim. The stiffer shims behind the ring shim are now more resistant to bending, so more preload force is created. This also occurs to a lesser extent with non-pre-loaded shim stacks. There will also be an increase in slope as a result of the stiffer shims.

Whether it is changes to preload or stack stiffness that rises or lowers the knee, the starting point will always follow the theoretical low speed curve.

Because the low speed curve does not rise vertically this means a higher knee will occur at a slightly higher damper speed, and a lower knee at a slightly lower damper speed. The only way to

keep the knee opening speed constant is to compensate by closing the low speed adjuster a small amount.

If you desire an increase in slope but not an increase in knee force, this can be achieved by increasing the stiffness of the stack (see "Stack stiffness" on page 17) and then compensate for the additional preload force by lowering the static preload. The opposite is also valid.

Sometimes compensation can be achieved by changing the diameter of the clamp shim while changing the thickness and/or quantity of large diameter shims at the same time. This technique works for both preloaded and non-preloaded shim stacks but is the only option for non-preloaded shim stacks. The exact proportion for compensating is best determined with the help of a damper dyno.

Example: a larger clamp diameter together with a thinner sealing shim and a smaller diameter of the following shim will give more slope with a similar knee.

Figure 22 shows the influence of shim stack stiffness with preload compensation.

If you think the knee force is optimized, a slope change should not be performed without including preload compensation.

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.

When a steep slope is needed together with a soft opening characteristic a *negative preloaded shim stack* can be used. This makes the shim stack open smoothly and quickly to small movements.

See also "Ring shim and centering shim" on page 15. □

7. Hysteresis

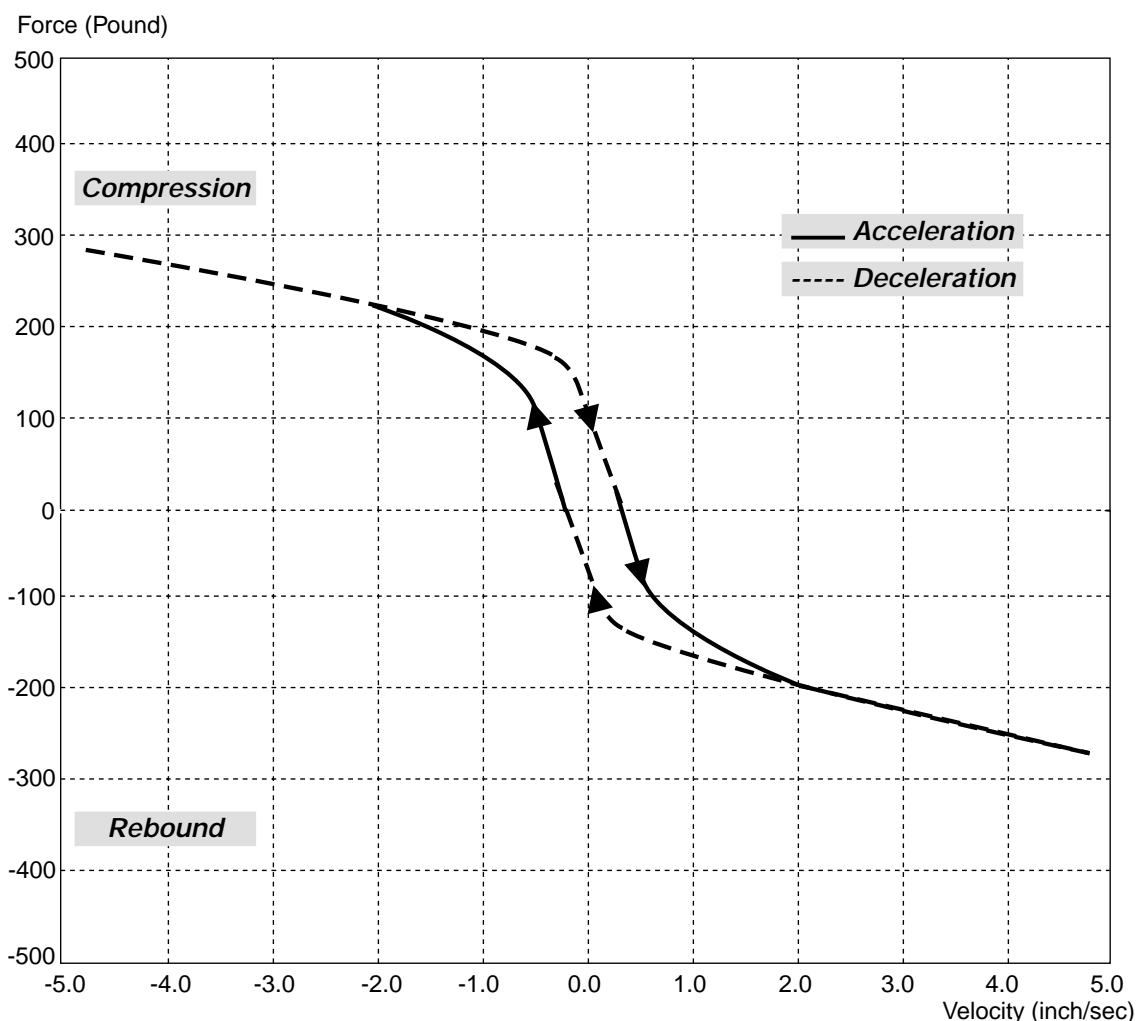


Figure 23. Damping force during a complete sine-wave cycle illustrating hysteresis.

On damper graphs there is always a difference in ascending (acceleration) and descending (deceleration) parts of the curve. We normally refer to this

area as *hysteresis*. Technically, the term hysteresis is related to energy losses but here we are actually storing energy as the damper acts like a spring.

Note: In figure 23, negative velocity illustrates compression, positive rebound.

This delay in damping force build up

7. Hysteresis

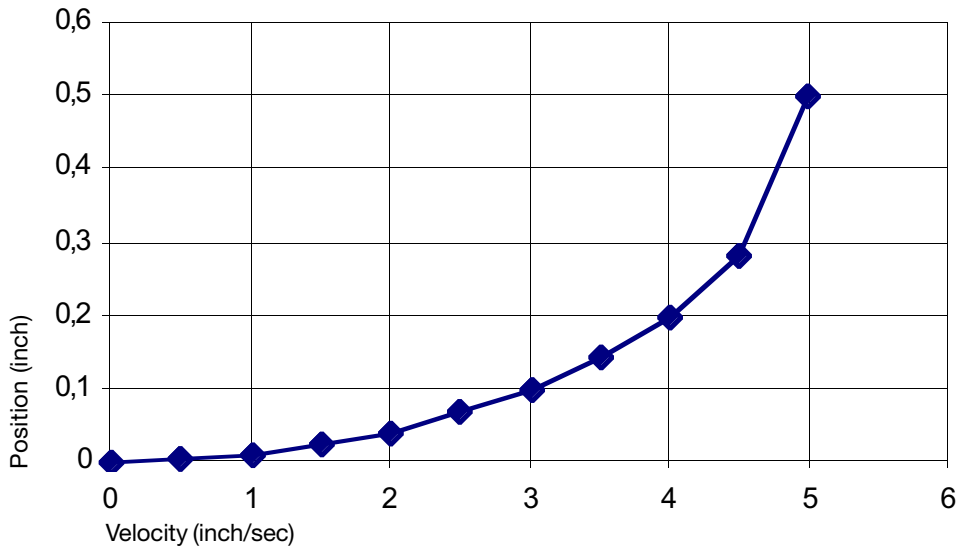


Figure 24. Stroke-velocity relation in a typical dyno run. Peak velocity 5 ips and frequency 2 Hz.



All Öhlins dampers, not just the racing dampers, are dyno tested before they are delivered to the customer.

can always be seen during a complete sine-wave cycle, even if the bleeds are closed. So this delay is not related to leakage but to the hysteresis in the damper.

Hysteresis is actually compressibility in the damper system. This delays pressure rise and pressure drop. Hysteresis affects the performance of the damper. In formula racing where the damper strokes are short, you are dependent on quick damping force build up. Because

of this, hysteresis needs to be kept to a minimum.

The compressibility can be classified into three different groups: elasticity of the damper parts, compressibility of the oil itself and compressibility of the gas in the oil. Also the gas volume causes a hysteresis effect that might be seen on the dyno curve depending on how well the dyno takes care of the gas force compensation. The elasticity of the damper itself is linear.

The compressibility of the oil itself is close to linear. However, the gas pressure will affect this part of the hysteresis.

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.

The gas in the damper is a result of different factors. Air enters the damper during the filling procedure and some air is already integrated with the oil when it is delivered. But even without air there will be gas bubbles at low pressures since the oil contains different additives that boil at different pressures/temperatures.

When the oil is under low pressure those additives change to a gaseous form which create bubbles. There will be even lower pressure locally at the valves than the theoretical pressure calculated from the damping force or measured by a pressure gauge, ie, due to dynamic pressure. With dynamic pressure, the oil volume changes caused by movement of the main piston not having an immediate pressure response throughout the entire volume of oil. Gas bubbles can be prevented by maintaining the recommended nitrogen pressure and by additional damping by the reservoir. See chapter "Combining main piston and reservoir damping" on page 21.

The amount of hysteresis for a certain damping force can be very different depending on the size of the piston (eg, pressure area) and the amount of oil under variation of pressure. The larger the piston is, the easier it will be to reduce the hysteresis. This is explained by the formula $F=pA$. F is the damping force, p is the pressure and A is the pressurized area. For a specified damping force (F) a smaller area (A) will lead to a higher pressure (p). The higher pressure will compress the oil more. This will cause more hysteresis. The piston shaft 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.

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.

In figure 24, the relation between displacement and velocity is shown for a peak velocity of 5 ips and a frequency of 2 Hz.

Note: The displacement at lower velocities is very small and every delay, slack in the system is shown very clearly. □

8. Combining main piston and reservoir damping

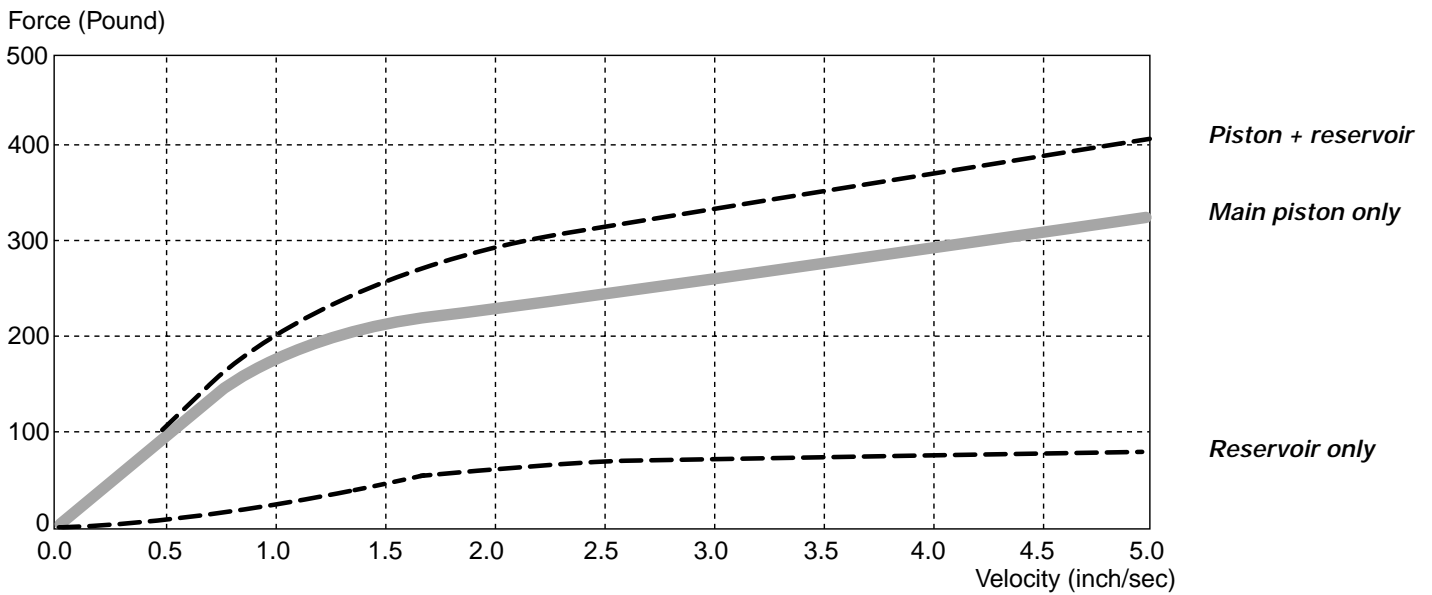


Figure 25a. Combining main piston and reservoir damping. Example 1. (RES-15-05, RC:2.)

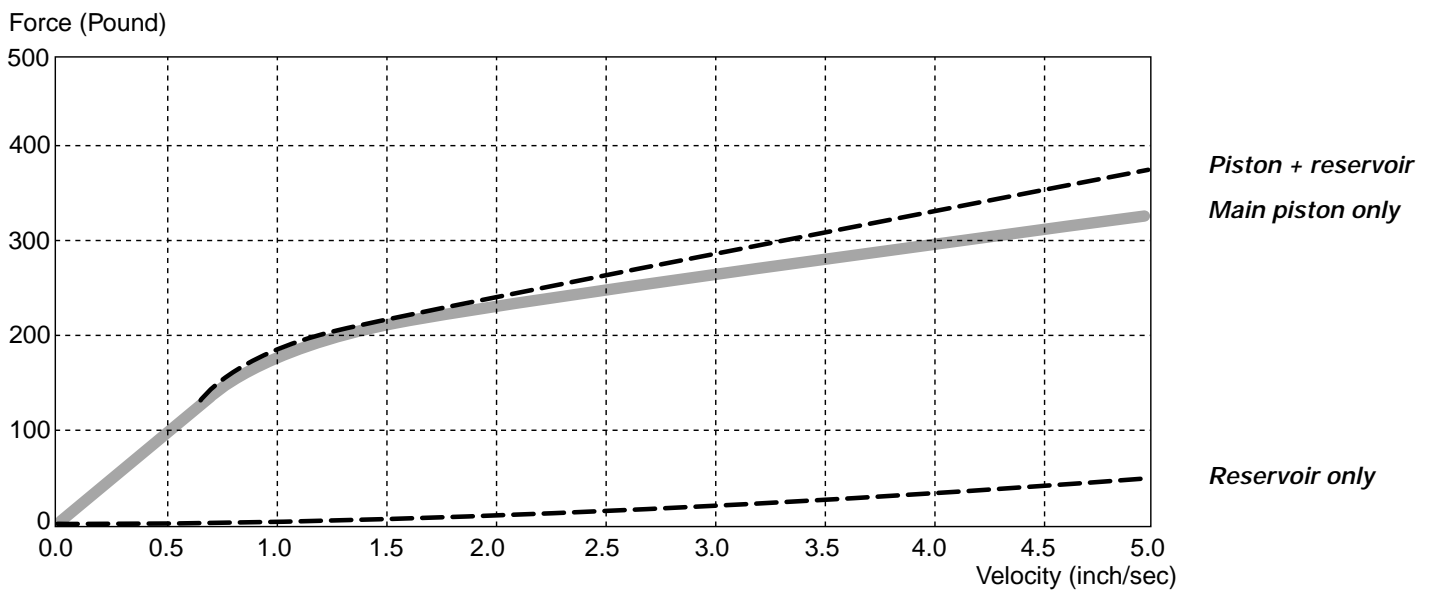


Figure 25b. Combining main piston and reservoir damping. Example 2. (RES-15-05, RC:8.)

The compression damping is the sum of the damping curves from the main piston and the one from the reservoir. Both curves consist of a nose, a knee and a slope.

In the figures 25a, 25b and 25c the compression damping from the main piston is the same (Baseline compression curve).

In the first example the reservoir stack is preloaded 0.05 mm (stack RES-15-05) and the reservoir adjuster is set to click position 2.

In example 2, only the click position of the reservoir adjuster differs from example 1. Here it is at click position 8.

In the last example, the click position is back at click position 2, but now the

preload has increased to 0.15 (stack RES-15-15).

The total damping at a specific speed is calculated by adding together the main piston damping and the reservoir damping at that speed.

As mentioned in the chapter about hysteresis, there will be a difference in force build up depending on the different stiffness of the two systems.

Assuming that there is no restriction in the reservoir, the compression damping force is built up by a pressure drop on the rebound side of the main piston. By using a restriction in the reservoir there will also be a pressure build up on the compression side of the piston. This results in a quicker pressure build up since

we have a pressure increase on one side of the piston at the same time as the pressure is dropping on the other side of the piston. Note that under these circumstances the reservoir restriction reduces the hysteresis. The amount of reservoir damping needed to achieve quick damping build up is small and all excessive reservoir damping will cause high hysteresis.

We can find the exact amount of reservoir damping needed to maximize the compression damping force build up by using the available pressure areas in such a way that the flexing of the damper is minimized. We call it "balancing" the reservoir damping with the main piston damping. The method will be described

8. Combining main piston and reservoir damping

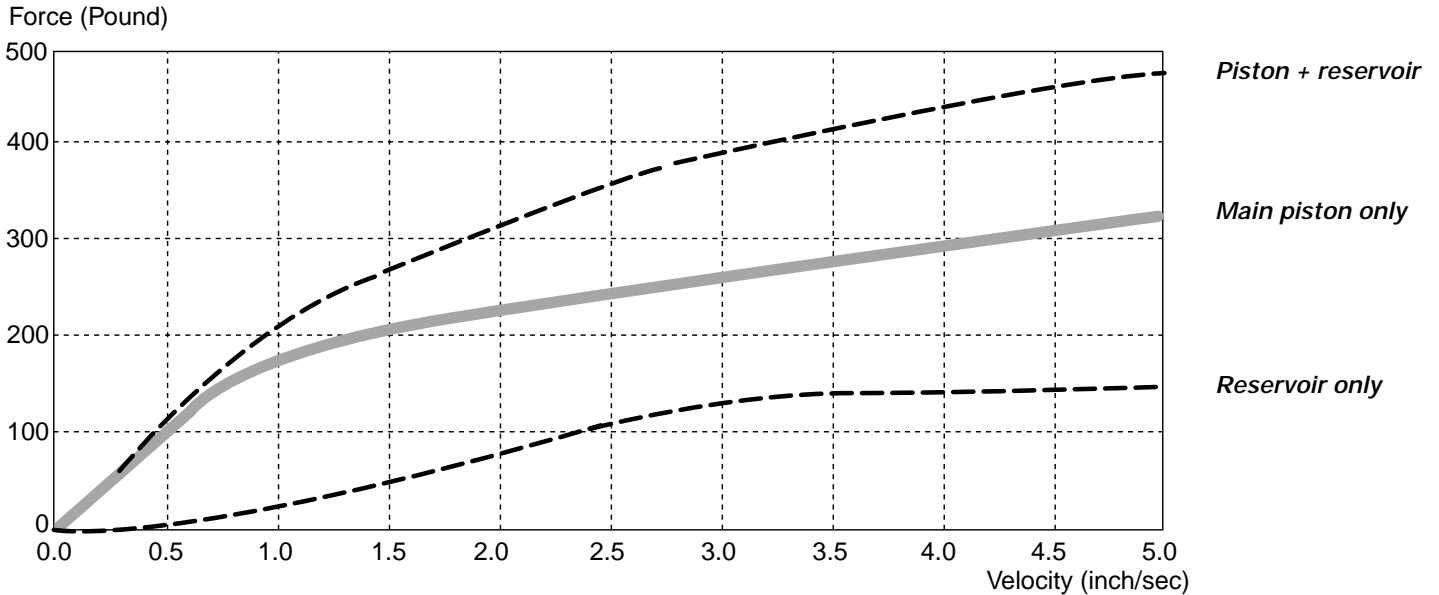


Figure 25c. Combining main piston and reservoir damping. Example 3. (RES-15-15), RC:2.)

in the text below.

A damping force is the result of a pressure difference multiplied by actual area. If we want to maximize the damping

force build up, we have to use all areas available. By spreading the damping between the main piston and the reservoir in relation to their area relations, the

damping force build up can be maximized.

The pressure area of the main piston can be calculated by first calculating the whole area based on the o.d. of the piston. However, because of the presence of the shaft on the rebound side all the oil on the compression side cannot travel across the piston. To find the pressure area at the main piston, A_{piston} , the shaft area, A_{shaft} , must be subtracted from the total piston area. The subtracted shaft area represents the oil that is forced into the reservoir. The formulas below describe the theoretical relation between the total damping, F_{tot} , and the damping from the reservoir, F_{shaft} , if maximum pressure build up is the goal.

$$F_{tot} = Dp A_{shaft} + 2Dp A_{piston}$$

$$F_{shaft} = Dp A_{shaft}$$

$$F_{shaft} / F_{tot} = A_{shaft} / (A_{shaft} + 2 A_{piston})$$

For the TT44 the ratio F_{shaft} / F_{tot} is 0.07.

This means that the damping from the reservoir should be 7% of damping from the main piston. Normally we recommend a setting, where the reservoir damping is more like 15% of the total damping.

If you study the damping force build up more carefully, you would find it to be quite complex. However, even if the discussion above is very simplified the result is very useful as a rule of thumb.

Note: There are sometimes situations where you are not looking for a maximum build up of damping force.

Since the rebound damping is caused only by the main piston oil flow, no balancing between the main piston damping and the reservoir damping can take place. □



The Ohlins four poster rig can reproduce both dynamic and aerodynamic forces and is widely used by race teams and car manufactures.

9. Damping functions



In 1993 Nigel Mansell won the CART championship for the Newman-Haas team with very special Öhlins dampers. The experience gained with these dampers led to the TT44 design.

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 optimizing “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 “aero” 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 optimizing 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 center of gravity which could also change dynamic roll centers. 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 right category. If this is achieved, the focus of further testing will be more on target and the possibility of a wayward theory will be minimized.

All this may sound too hypothetical but rest assured if you optimize 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.

Ground effect cars

In this ground effect age, dampers can maximize 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 center of the downforce is found where the geometry of the underside comes closest to the ground. With either type of car the center of downforce migrates with any change in pitch angle in relation to the ground caused by braking, cornering or acceleration. This migration of the center 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 center of gravity. The tunnel flat part is in the closest proximity to the ground and that is where the center 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 minimize 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



Today the Öhlins TT44 design is used in touring car racing and have even been tested on bikes. The picture is showing a BTCC front strut.

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).

Grip and compression damping

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 com-

pression 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 optimized 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.

Qualifying or race

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.

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.

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. □

10. Factory recommended damper set-ups

ÖHLINS SHOCK ABSORBER SPECIFICATION CARD TT44

FORM 029.002

| | | | |
|--------------------------------------|----------------------------|--------------------------------|--------------------------|
| REYNARD 308 | Model 981-09J | Year 1996-99 | Part No. REA 69000 |
| Shock body 5428-04 | Stroke 73,6 | Type TT 44-4-way | Position REAR left/right |
| Shock oil 302-02 | Cylinder diameter 5411-02 | Shim 5400-04 | Click 98-12-07 |
| Reservoir cap 578-01 | Reservoir type 5415-11 | Shim 5414-01 | Part No. 5469-09 |
| Reservoir lock nut for piston 578-01 | Lock nut for piston 539-01 | Valve seat compression 5414-01 | Part No. 5440-01 |
| Valve seat 5415-11 | Lock nut for piston 539-01 | Valve seat compression 5414-01 | Part No. 5440-01 |

| Compression | Rebound | Reservoir stack | Shim fitting code |
|----------------|-----------|-----------------|------------------------------|
| 1. 530-38 (2x) | 1. 530-36 | 1. 185-12 | Shim fitting code |
| 2. 530-36 | 2. 530-33 | 2. 186-12 | Shim fitting code |
| 3. 530-34 | 3. 530-30 | 3. 185-18 | See notes |
| 4. 530-30 | 4. 530-28 | 4. 185-16 | Any adjustment affecting sh. |
| 5. 530-26 | 5. 530-26 | 5. 185-14 | Pressure inspection sheet |
| 6. 530-23 | 6. 530-23 | 6. 185-12 | Pressure inspection sheet |
| 7. | 7. | 7. | Measuring your clearance |
| 8. | 8. | 8. | |
| 9. | 9. | 9. | |
| 10. | 10. | 10. | |
| 11. | 11. | 11. | |

| | | | |
|---|--|---------------------|---------------------|
| Spring adjuster 5409-01 | Spring cap 5412-01 | External spacer | Compression click 8 |
| Spring 5536-01 | Pin tool | Shim holder 5536-01 | Rebound click 18 |
| Spring collar bush 5536-01 | Shim bearing (cylinder body) 5425-01 (2x) | Shim holder 5536-01 | Rebound click 2 |
| Shim bearing (cylinder body) 5425-01 (2x) | Shim | Shim holder 5536-01 | Rebound click 2 |
| Part No. REA 69000 98-11-02 | High speed compression adjuster. Spring 5473-02 mounted (from wire diameter) Click position 35 | Shim holder 5536-01 | Rebound click 2 |
| Change Cylinder body internal | Assy drawing: TT44-01A1 / HSC-01A1 | Shim holder 5536-01 | Rebound click 2 |
| spaces (only correction of spec. card) | | Shim holder 5536-01 | Rebound click 2 |

Öhlins distributors worldwide have specification cards available with recommended damper settings.

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 distributors worldwide have specification cards available with recommended damper settings. These settings are track proven with good results for racecars with their manufacturer's standard configurations.

Öhlins does not suggest that these settings are perfect and cannot be improved, but we recommend them as a point at which to start testing. The spec. cards give information about valving as well as clicker settings. For most vehicle classes both road course and oval settings are available. Sometimes recommendations for springs are also available.

Note: Dampers delivered by the distributor are usually valved and adjusted for road courses with the exception of IRL dampers (Indy Racing League). The dampers can usually be modified by the distributor to almost any setting the customer desires.

Updated continually

The information on the spec. cards is updated continually. Make sure you have the latest version of the spec. cards. The older the design date on the spec. card is, the greater the possibility of a later, improved spec. card.

For quality control, all Öhlins TT44

dampers are dyno tested at least once before they are delivered to the customer. They are first tested at the Öhlins factory in Sweden. At the Öhlins factory, the dampers are always tested with the settings according to the specification cards. At our distributors the settings may be changed by request or updated, then before delivery the dampers are tested a second time.

Standard or modified

If track testing indicates that your racecar works best at clicker positions far away from the spec. card suggestions, this is a clue indicating that the internal shim stacks are not quite right for your application. Undesirable handling characteristics also suggest that a shim stack modification is in order.

If your car is modified from "standard" it is difficult to predict how much the damping needs to be changed from the standard settings. However, if the modification is to the damper rocker ratio, compensating damping force for a new ratio can be calculated. Having intimate knowledge of your damper mounting geometry is the key to predicting the proper amount of damping forces.

Rockers and click settings

When it comes to proper clicker settings in relation to damper rocker ratios, Öhlins has some general rules. For racecars with damper/wheel rocker ratios of

around 1.0, we recommend for low speed compression (LSC) a clicker range of 6 to 12 clicks. For low speed rebound (LSR) 15 to 25 clicks is a good starting range.

If your rocker 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 damping forces at 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.

Tightening the clicker will achieve more damping at a lower speed. However, more often than not, shim settings need to be changed to compensate for leverage changes.

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

Öhlins technicians can help you calculate damping curves necessary to compensate for changes in damper/wheel rocker ratios. □