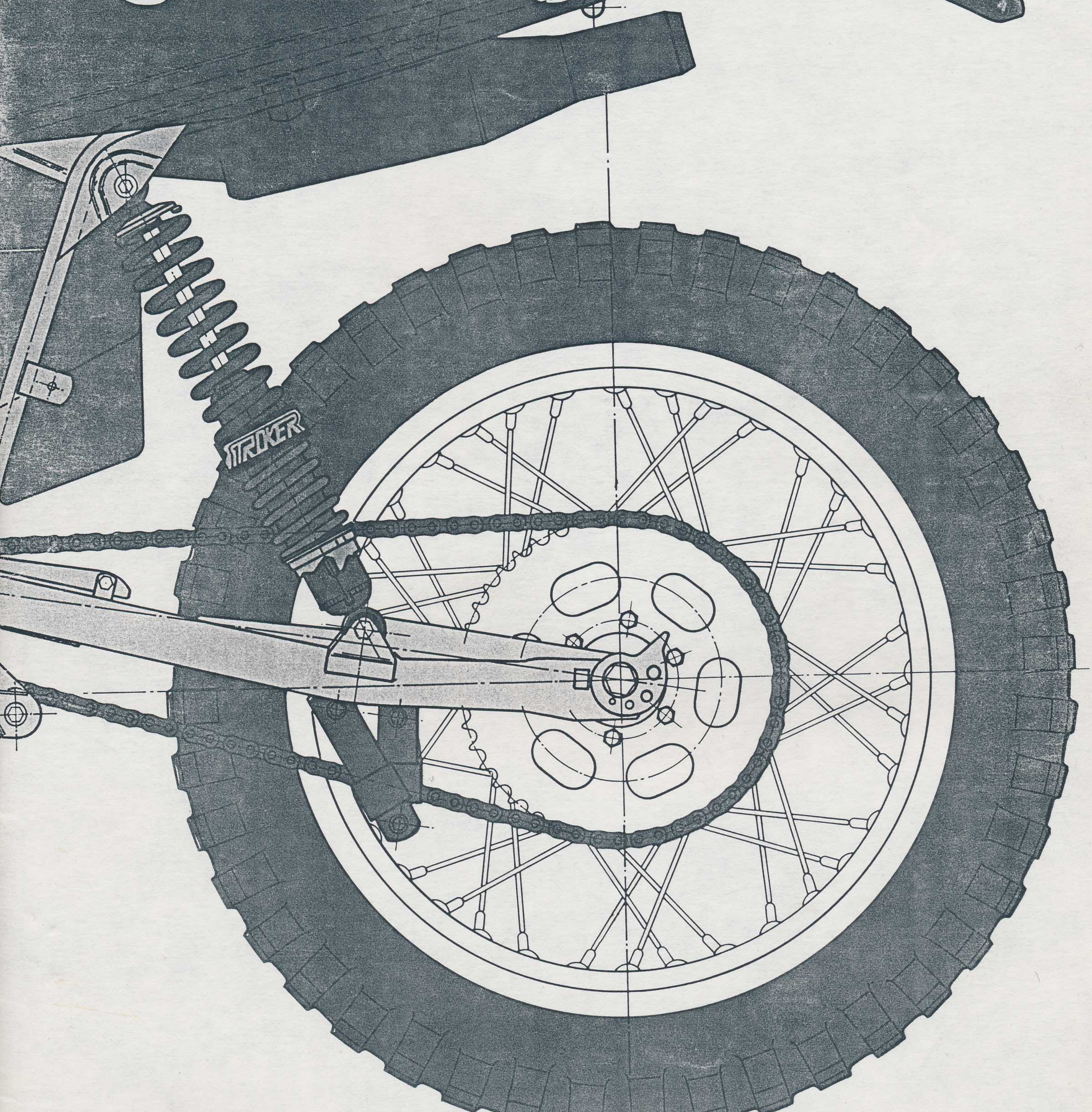
S&W

Suspension Engineering Handbook



about S&W

S&W Engineered Products are well established as one of the leading suppliers of motorcycle suspension systems.

Tim Witham — the "W" of S&W — is the originator of the company and both Tim and his associates have been responsible for bringing a whole range of practical and successful products to the market place.

Tim combined several ingredients for success, notably his long years of close contact with tuning both motorcycles and Indianapolis racing cars, and his intuitive approach to engineering improvements.

By 1972, S&W's active growth had begun, and the company's reputation, particularly for quality springs, was widely recognized. At that time, the same clear message came from racers and touring riders alike — provide us with better suspension and a wide range of spring rates and damping controls.

Using all-American built components as a basis, S&W set to work on a research program to develop a series of dampers and springs to suit everything on two wheels from the lightest off-road machine to the heavy touring bike.

There soon followed another innovation, Air Shocks for road touring machines, and these units generate as much enthusiasm today as when first introduced. Now accounting for a sizeable proportion of S&W sales, the air shock system combines the ride comfort of a low-rate suspension with load-carrying capacity.

The production of valve springs remains an important part of the company's output, and springs suitable for use with high-lift camshafts are available for the popular engines.

Now actively establishing world-wide sales, S&W's product line continues to grow to meet the needs of today's performance-conscious motorcyclists.

about the author

For his first venture into writing Bruce Burness brings to focus a wealth of practical experience gained from a 23 year involvement in motor racing.

In the early sixties Bruce turned his teen-age preoccupation with cars and speed into a profession by making a full time commitment to Carroll Shelby and Ford Motor Company for their assault on international road racing.

Subsequent to leaving the Shelby Organization, Bruce designed, constructed and maintained a road racing car that won for George Follmer the title of "United States Road Racing Champion" for 1965. For his contribution Bruce was awarded the "Mechanic of the Year" award by the Sports Car Club of America.

The next ten years he saw many varied forays into racing through motorcycles and sports cars, Indianapolis and Formula Cars, that brought a gradual transition away from the fielding of racing efforts and more in the direction of the design and fabrication of racing vehicles and equipment.

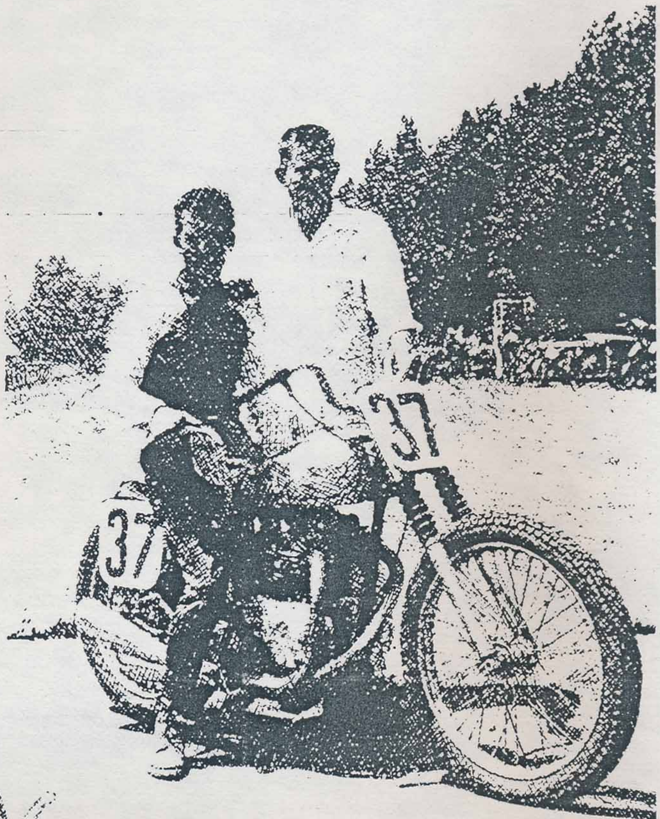
In 1974 Mr. Tim Witham of S&W Engineered Products asked Bruce to look at the booming sport of motorcycling as a practical avenue to apply the sophisticated technological fallout from Indianapolis and high performance automobiles. The next three years Bruce worked at S&W as Head Designer for Development of Suspension Components and New Concepts. Bruce Burness is now doing freelance design and consulting and continuing to develop his knowledge of high performance suspension components which are so vital to the modern motorcycle.

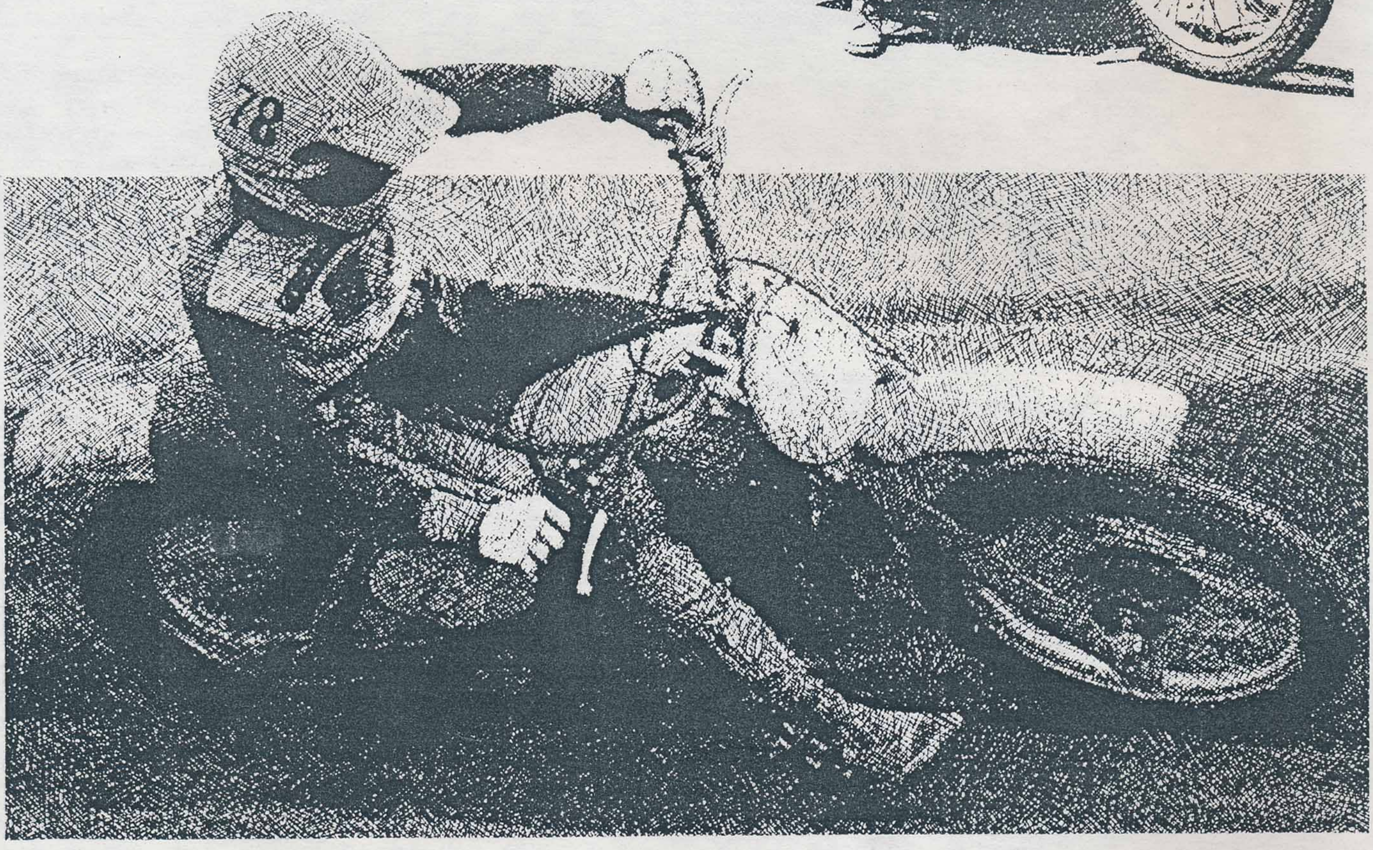
Introduction

The last few years have seen an incredible evolution in motorcycle suspension systems, creating an atmosphere in which this month's hot tip is next month's obsolete junk. We feel that the frantic evolution in suspension has finally stabilized, and that the time is ripe to offer a general review of motorcycle suspension and the current state of the art.

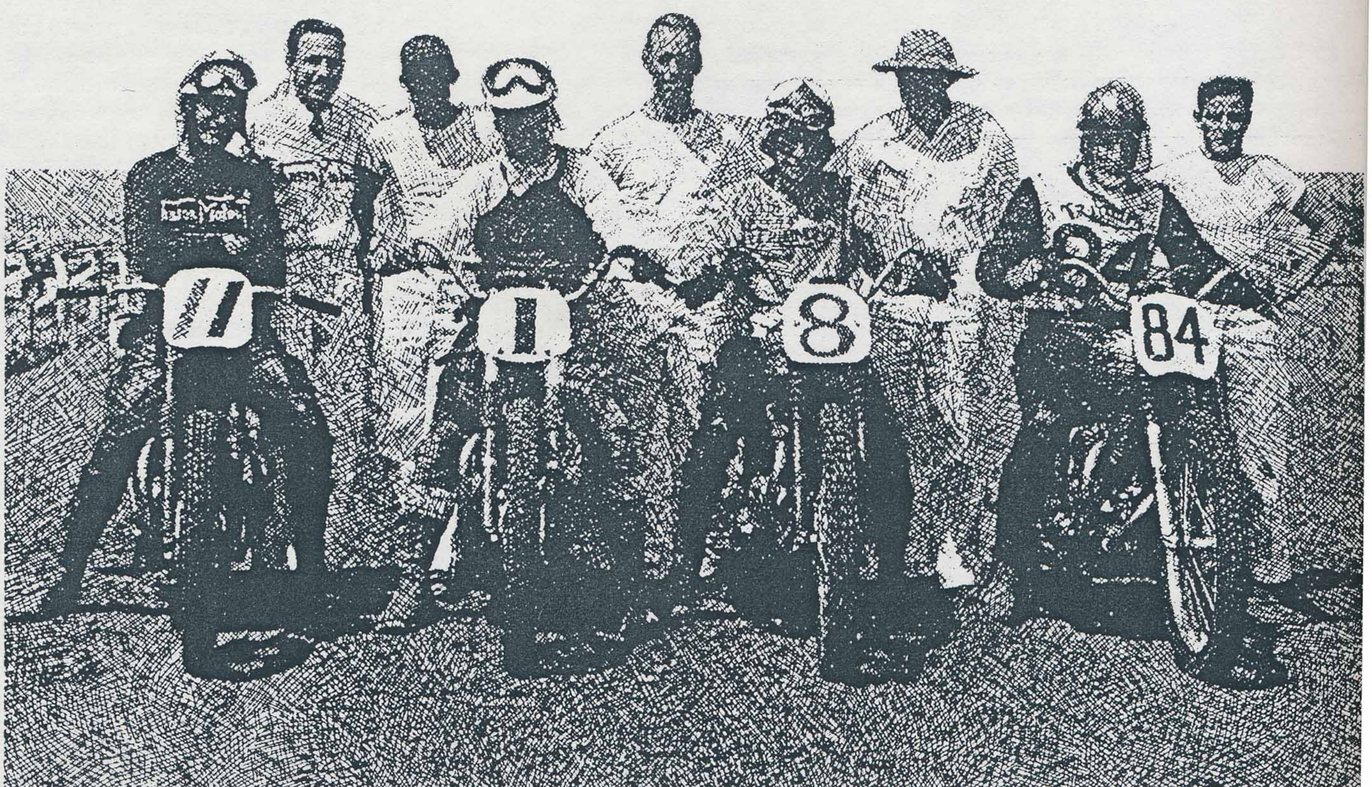
The first chapter will be very general in nature and will present some basic suspension theory and describe the various elements and components that comprise a successful suspension system. Beyond the basics, we will present specific discussions of the various elements that contribute to the feel and performance of your motorcycle. These later chapters will relate directly to your motorcycle and should aid in sorting out your suspension problems.

This handbook on suspension will be presented in layman's terminology and should be easily understood. To better present the material in this manner, some literary license will be exercised, and you trained engineers out there may find some dialogue not precise enough for you. Try to be tolerant of the gap between the professional and the person in the street (or dirt, if you wish).





Chapter...



Chapter... 1 Suspension Theory

If one were to put down the criteria for the perfect suspension system, the primary objective would be for the vehicle in question to be able to negotiate any and all road irregularities while the passengers remain completely isolated from any feeling of swaying, bumps, or vibration. Of course, this is the "ideal" suspension. Still, it is important to keep this criterion in mind when improvements are contemplated for your own motorcycle.

Design engineers approach the suspension problem well aware that the best they can hope for is to *minimize* the adverse sensations imparted to the rider. The reason these sensations can only be minimized and not removed entirely is that there is always a compromise point where correcting one characteristic brings on another that is just as annoying. An example of this type of compromise decision can be found in most motorcycle front forks. A typical condition to overcome is the tar strips found every few feet between each concrete slab on freeways. Because of their repetitious nature they can be very uncomfortable. When designing a fork spring you would like to have a rate soft enough to comply with those miserable tar strips. However, when that is achieved, you have excessive nose dive the first time you apply the front brake. This can bring on a myriad of stability problems. Invariably, the compromise is acceptable nose dive and not so acceptable compliance with the tar strips.

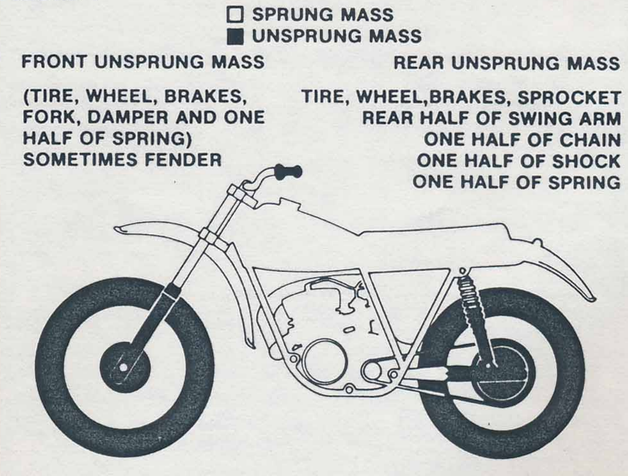
The designer's job is to find the happy medium for the specific application. The key word here is **specific.** As motorcycles become more sophisticated and specialized, a part designed for one type will not be suitable for another. Also enter in the particular bias of each designer (they all prefer a slightly different feel), enter in the speed potential of each individual rider (which alters the suspension requirements), and you begin to understand the dilemma that the manufacturer faces to fill your needs. What you get with the purchase of a motorcycle is a very involved compromise, not perfect for anybody.

The information presented here will hopefully help you assess your own particular set of needs and priorities to make the right selection of S & W components so your motorcycle is optimized for you.

SPRUNG AND UNSPRUNG MASS

To begin, let's take a quick look at the various elements comprising a suspension system. The two most elementary components are the *sprung mass* (or sprung weight) and the *unsprung mass.* The unsprung mass is perhaps more accurately referred to as unsprung *masses,* as it can be broken down into front and rear unsprung masses. Refer to **figure 1** to get a better idea of the parts of a motorcycle that belong to each group.

FIGURE 1



If you ride a vehicle across some road undulations the tires will have to move upwards the height of each bump in order for it to travel forward. Now if you don't want the sprung mass (or chassis) to notice those bumps, the system will have to allow the tires to move independently of the chassis. To do this, a system of links is incorporated to allow the wheels to swing (the rear swing arm) or slide (front forks), freely in one plane, but still remain connected to the chassis to perform other necessary functions like accelerating, braking and steering.

The original requirement for freedom of movement suggested that the wheel be allowed to move upwards or vertically to accommodate road bumps. Unfortunately, gravity also works in a vertical plane and is working on both the sprung and unsprung masses. The unsprung mass is in contact with the ground already, but the sprung mass is not, and without support it too wants to come in intimate contact with the ground.

To support the sprung mass we introduce springs into the system between the two masses, as they will deform and allow wheel movement, but at the same time will *transmit a force out of one end equal to the force put into the other,* which will support the chassis.

Indirectly, we have just made the statement that the springs put forces into and out of the two masses. Let's take a look at some of the properties of the sprung mass to see how it responds or reacts to these outside forces. One of the laws of physics states that a body moving in space will continue to move in the same direction and at the same speed until acted upon by outside forces. Visualize the sprung mass as moving freely in space. Visualize the springs as pushing and pulling underneath each end. The ability of the sprung mass to continue on a relatively undisturbed course is influenced by (1) its total mass; (2) the distribution of weight within that mass; and (3) the speed of that mass.

The amount of total mass is the primary influence. The greater the mass, the greater the force necessary to change its course. If you keep the speed and forces constant and increase the mass, the change in course will be less, i.e. road irregularities will affect the vehicle less, resulting in a better ride. Conversely, if you reduce the mass, the ride will become worse.

chapter ...1

Now we all know that light motorcycles are more maneuverable, accelerate and brake quicker, and are generally more pleasurable. The statement above suggests light motorcycles have a ride potential that is inferior. Fortunately (for reasons we won't go into here), a more specific gauge of ride potential is the ratio between the sprung mass and unsprung mass. If you can reduce the amount of sprung mass (taking weight off the chassis is relatively easy), and somehow reduce the unsprung mass by the same percentage (taking weight off the wheels is not so easy), you can have your light, nimble motorcycle and not sacrifice the boulevard ride.

Incidentally, this ratio between sprung/unsprung masses is considerably worse for motorcycles than for the average automobile. This partially explains the generally inferior ride potential of motorcycles. Also, because of the difficulty in removing a significant amount of weight from the wheels, the ratio becomes worse yet on racing motorcycles.

The last part of the sprung mass equation to consider is the speed at which it is traveling. While a mass is stationary, the force necessary to move it equals the mass. As you begin to move the mass, it acquires another value known as *momentum*. Momentum is the product of the mass and the velocity at which it is traveling. Translated, this means that the faster you go, the more momentum you have. The more momentum, the higher the force required to alter your course. As you increase your speed, little obstacles that were noticeable at slow speeds begin to disappear. However, increased momentum also raises the requirement of the springs to keep the suspension from bottoming out over larger bumps. Speed difference has been found to be the biggest variable in suspension requirements for various riders. This is an important point to keep in mind when considering your own suspension requirements. An honest appraisal of your own style and aggressiveness will shed a lot of light on the amount of suspension stiffness suited to you. Thus, this is the reason for new long travel suspensions which have the ability to soak-up high load resulting from large bumps.

A good general rule is to have the *softest suspension possible with rare bottoming-out.* Too soft will cause excessive bottoming and will be the limiting factor to speed over rough terrain (and dangerous too); too hard a suspension will generally feel uncomfortable under all conditions and will undermine the confidence and pleasure of the rider.

SPRINGS

Before we examine the nature of the forces being put into the sprung mass through the springs, we need to know more about spring characteristics.

The primary and most relevant trait of all springs is a thing called *rate.* Rate is the indicator of stiffness or softness of any given spring. Specifically, *rate is the amount of force (or weight) necessary to deflect (or compress) a spring a given distance.* It is generally expressed as lb./in. (pounds per inch). For example, look at the springs in **figure 2.** The spring in all cases is the same, but with varying forces (or loads) placed on top of it. With the 100-pound load the spring deflects one inch. With a 200-pound load it deflects two inches. With a 300-pound load it deflects three inches and will continue to deflect another inch for every increase in load of 100 pounds until coil bound. This spring can be said to have a rate of 100 lb./in.

In **figure 3** we show three different springs with and without load: One of 100 lb./in. rate, one of 50 lb./in. rate that is the same free length as the 100-lb./in. spring, and one of 50 lb./in. rate with a free length longer than the other two. The load placed on them is the same for all. Notice that the spring with the 50-lb./in. rate and the same free length is deflected twice as much as the 100-lb./in. spring until coil bind. Notice that the longer 50-lb./in. spring ended up the same length as the 100-lb./in. spring when loaded. The message is: Springs with lower rates will deflect more for the same load, but it is possible for two different rate springs to be the same length when loaded if the lower rate spring is longer to begin with.

Referring to **figure 3** once again, imagine lifting the three springs, with the load, off of their resting place. The force required to do so, in all three cases, would be the same even though the springs are all different. This goes back to our earlier statement about force into one end equalling force out of the other end. The concept behind all this is that spring rate and spring load are quite separate and both must be considered when selecting replacement components for your system. (Moving the spring cam up on your shocks does not make the springs stiffer, it just adds preload). An in-depth discussion of spring types and how rate or rates is determined will come in Chapter 2. Just fundamentals for now.

FIGURE 2

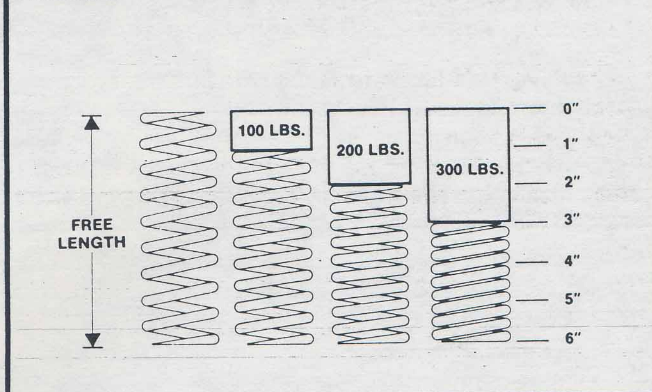
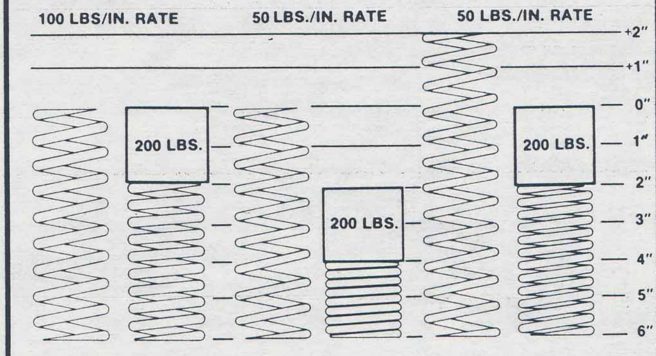


FIGURE 3



chapter ...1

SUSPENSION FORCES

When a motorcycle negotiates a bump a lot of energy is created upon impact. What we ask of our suspension system is to dissipate or spread out that energy to a level that is acceptable to the rider. The amount of this energy is proportionate to the momentum of the vehicle and the size of the bump. (Remember: Momentum is mass times velocity.

In **figures 4** and **5** we have installed our 100-lb./in. springs directly over the axles of our elementary motorcycle. For the sake of simplicity let's assume that the sprung mass (at this time) is so large that it won't change attitude regardless of the forces acting on it. In figure **4** the size of the bumps is four inches. To go over those bumps the wheels will have to move four inches. If the sprung mass doesn't move this will compress the spring four inches. Multiply four inches by its rate of 100 lb./in. and you have a force of 400 pounds.

In **figure 5** the bumps are twice as large and the force generated is twice as much lending credence to the notion that the energy created is proportional to the bump encountered. **Figure 5** assumes the elementary motorcycle has at least eight inches of wheel travel. If it had less than eight inches, the suspension would bottom out and the sprung mass would move upwards (regardless of its size) at a speed directly related to the shape of the remaining part of the bump.

FIGURE 4

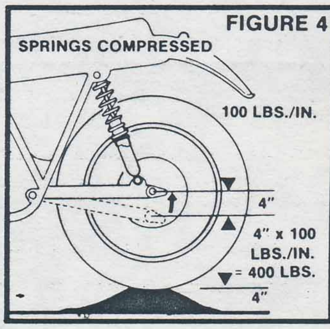
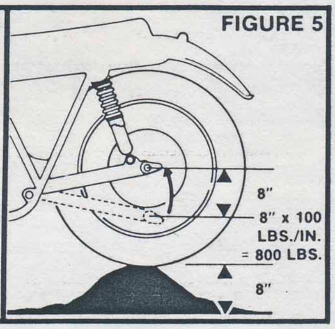


FIGURE 5



The assumption that the sprung mass is infinitely large and impervious to forces is, of course, erroneous. In fact, the chassis reacts to some degree to all changes in spring force. The amount it reacts is governed by its own momentum. In **figure 4** the wheel moved four inches, creating a force of 400 pounds. If we change the springs to a 50-lb./in. rate, the same bump will create a force of only 200 pounds (4 in. x 50 lb./in. = 200 lbs.). The rider will appreciate this smaller force, but where did half the original force vanish? In the 100-lb./in. case the sprung mass experienced much higher spring forces right away and reacted more by moving upwards. As it moved upwards the spring didn't have to compress as far to clear the bump. The full 400 pounds was never generated, but the **rider absorbed some energy.** In the 50-lb./in. case the change in spring forces was more gradual, so for the same bump the chassis moved less and the spring compressed more. It seems softer springs affect the chassis less but need more travel to do the same job. Conversely, if you increase the travel you can use a softer spring rate. It is safe to say that if all other things are equal, the motorcycle with the most wheel travel will be able to either give a better ride or go faster over rough terrain.

DAMPING

When last seen, our elementary motorcycle was sitting at the top of a bump with its springs compressed. One thing we have learned is that springs will give back the same energy put into them. Now if we don't want that energy to affect the chassis, we should devise a method of using up that energy. Enter the SHOCK ABSORBER.

Shock absorbers in this day and age are almost always hydraulic devices. They burn up energy from suspension movement by pumping oil through a system of valves and orifices to create resistance. This resistance to movement controls the springs and converts the stored energy by heating the shock oil as it works. The heat is then disposed of into the surrounding air.

We have learned that the amount of force generated by a spring is proportional to how FAR you deflect it. A shock absorber is quite another animal. Its forces are related to how FAST you move it. In fact, as you pass oil through a fixed orifice, its resistance will increase as the square of the velocity. The faster you move it the more resistance is created. The key to a successful shock absorber (or damper) is to match its resistance to the forces generated by the spring. Earlier we saw our elementary motorcycle traveling over the same bump with two different spring setups. Be aware that the different springs would require different shock absorbers. (A shock suitable for the heavy spring would overpower the lighter spring and limit suspension movement).

Because suspensions travel at an infinite variety of speeds and can be set up with any spring rate, a good selection of shock absorbers is needed. **(See the shock absorber range available from S & W).** Most sophisticated shocks have a system that allows a separate adjustment within the valves for each speed range at which the shock operates. These are sometimes called VALVING STAGES. This is because they progress from low speed, to medium speed, to high speed. Our S & W shocks have multi-stage valving. Sometimes, even more stages are utilized. With these stages, and various valve types, the shock engineer can juggle the damping curve for each application.

The two basic modes of a shock are **COMPRESSION** and **EXTENSION.** When we discussed burning up stored spring energy, we were describing the EXTENSION (or return or rebound) cycle. During the COMPRESSION cycle, real advantage can be taken of a shock's ability to respond only to speed. The compromise of soft springs bottoming too often can be helped with the addition of compression force. The trick is to have the compression damping come into play at higher speeds so that none of the advantages of soft springs are sacrificed to low speeds. Be careful of too much high speed damping as this will cause suspension lock-up over sudden bumps. In practice, the tailoring of damping for various speed ranges is much more involved, and subtle changes at all speed ranges are important.

The games to be played with shock damping could fill many volumes. In a later chapter we will get into some detail as to the damping possibilities and how to arrive at them hydraulically.

Chapter ...2 Springs

This chapter is going to deal with the idiosyncracies of various types of springs. Because motorcycles lean heavily to only two types of springs, coils and air, we will concentrate most of our attention there.

Coil and air springs each have individual and unique characteristics, therefore we will divide this chapter into two main parts and deal with coil springs first and air springs later in the text.

One trait of all springs is that it is impossible to predict their performance without direct field testing or elaborate test equipment. Most of the time it is impossible to make these evaluations until after you have spent your hard earned money. It is the intention of this chapter to provide you with enough basic knowledge of springs to insure you purchase what you and your motorcycle need and not just what your dealer has left on his shelf. In our first chapter we covered the basics of springs. We will now quickly review the essential concepts and add new additional terminology relevant to springs.

The term we will refer to most often is **spring-rate** or **rate.** Spring-rate is a measure of a spring's stiffness or ability to withstand various loads. Specifically spring-rate is the amount of load or force required to collapse or compress a spring one inch. In this country it is expressed as pounds per inch (lbs./in.), (in the metric system it is Kilograms per centimeter (Kg./cm.). If you apply a force of 300 lbs. to a spring and it compresses 3 inches, it is said to have a 100 lbs./in. rate.

$$\frac{\textbf{300 lb. load}}{\textbf{3 in. deflection}} = \textbf{100 lbs./in. rate}$$

It is very important not to confuse spring-rate with **spring load.** Load is the amount of force or pounds applied to the spring (for a motorcycle this is the weight of the chassis and rider multiplied by the leverage of the spring- shock mountings.)

If you know the load and you know the spring-rate you can predict the amount of **spring deflection.** For example if you have a 300 lb. load and you apply it to a 100 lb./in. spring, you will get 3 inches of spring deflection.

$$\frac{\textbf{300 lb. load}}{\textbf{100 lbs./in. rate}} = \textbf{3 in. deflection}$$

The spring rate determines how a spring reacts or deflects to changes in load.

Free length is the length of a spring before any load is applied or when it is standing free of its intended installation.

If you know the amount of spring deflection, due to the combination of load and rate, you can determine the **loaded length** by subtracting the deflection from the free length.

Many times the deflection caused by the load is referred to as **preload. Mechanical Preload** is the amount the spring is compressed when it is installed in the suspension. **Static Preload** is the mechanical preload plus the additional amount the spring compresses when it is supporting the chassis while at rest. Sometimes the mechanical preload is greater than the static preload. When this condition exists the suspension does not move until a bump large enough to overcome the mechanical preload is encountered. This is a condition to be avoided at all costs.

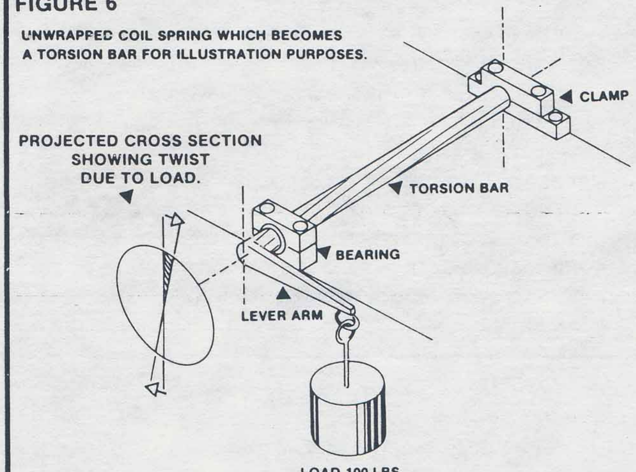
Now let's look into why one spring is stiffer than another. When a spring engineer is asked to design a spring with a particular spring-rate, he has three variables at his disposal. The first is the number of **active coils** he can fit into the requirements. (Active coils are the coils that do not touch each other and generally there are two less active coils than total coils. One dead coil is provided for each end to square up the ends so that the spring can be mounted between the adjustment cam and the top retainer clip, giving a good flat seat). Secondly, he can select various sizes of **spring wire** to cause the rate to go up or down. The last variable is the **mean diameter** of the spring. The mean diameter is measured from the center of the spring wire on one side of the spring to the center of the wire on the other side. (It is simpler to measure the inside diameter and add to it one thickness of the wire size.) The selected combination of these three elements determines the spring-rate of each spring. The difficult part for you and the cause of much confusion about spring stiffness is that two of the elements work inversely in relation to spring-rate. By that we mean that if you use more of one of those elements you get less spring-rate. Conversely if you use less you get more spring-rate. Understanding this concept is vital to comprehension of the rest of this chapter.

The two elements that are inversely proportional are the number of coils and the mean diameter. The size of the spring wire is directly proportional to spring-rate. (The bigger the wire, the stiffer the spring.)

For illustration purposes, imagine a coil spring unwrapped and straightened into a rod or bar. Now clamp one end solidly and support the other end in a bearing. Add a lever arm to the end in the bearing and we have a torsion bar **(figure 6).** If you pull on the end of the lever arm with a load the bar will twist in torsion at a specific spring-rate. The bar's resistance to twisting is generated by the interaction of the molecules working in shear against each other.

FIGURE 6

UNWRAPPED COIL SPRING WHICH BECOMES A TORSION BAR FOR ILLUSTRATION PURPOSES.



If you glance at **figure 6** again, you will see a cross section of the bar projected off the lever end of the bar. The pie shaped piece represents the amount of twist (or relative motion between molecules) from the clamped end to the free end due to the load. Notice the pie section is much wider at the surface than near the center. This indicates the mole-

chapter ...2

cules at the surface must resist movement much more than those at the center. This is the reason the spring-rate will increase if you increase the bar or wire size. Incidentally the stiffness increases to the fourth power of diameter. Also notice the center of the bar contributes very little to the stiffness which accounts for tubular bars being nearly as stiff as solid bars. We throw this in as an idea for saving weight.

Just like our original coil spring, bar size is not the only factor in determining the stiffness. The length of the bar is the other variable. Although it may be easy to comprehend that a larger diameter bar will be stiffer, it may not be so easy to accept that shortening the bar will also increase the stiffness.

Think again of the molecules tugging on each other creating a resistance to twist. A longer bar will have more molecules in it than a shorter bar. Each of these individual molecules is going to equally feel the total of all of the load or stress being fed into the bar. Each molecule will deform a specific amount when stress is applied. The amount of twist is the **sum** of the deformation of all its molecules. The bar with more molecules, or longer bar, will twist more, or conversely, a bar with less molecules, or shorter bar, will twist less with equal load.

Why are we preoccupied with torsion bars when our motorcycles come equipped with coil-springs? Keep in mind our torsion bar is just a straightened coil spring in disguise, and a coil spring sees exactly the same twisting action in its wire. The length of the bar has a direct relationship on two of the spring rate variables, associated with coil springs.

If you take that bar and coil it up, it will make just so many coils of a certain mean diameter before you run out of bar. If you want more coils or larger diameter coils you are going to need to start with a longer bar. That will make the coil spring softer. (If you add coils or make larger coils, the spring-rate will be lower. If you remove coils or shrink the diameter, the spring-rate will be higher).

Unfortunately the engineers' problems are not over once he has arrived at a combination that will produce the desired spring-rate. Generally the diameter of the spring is governed by whatever the spring goes over or into and cannot be varied much. One of his options is gone before he starts. The free length and travel are also likely to be predetermined. That means his combination of wire size and number of coils must not take up too much length when the spring is fully compressed (or coil-bound). (**Coil-bind** is when the coils are touching each other and are said to be **shorted out** and/or **dead coils.**)

If an engineer discovers his combination has too many coils to yield the required amount of travel before coil-bind he can remove some coils, which will raise the rate and then reduce the wire size which will bring the rate back down again. This may solve the travel problem, but brings on another dilemma known as **OVERSTRESS.**

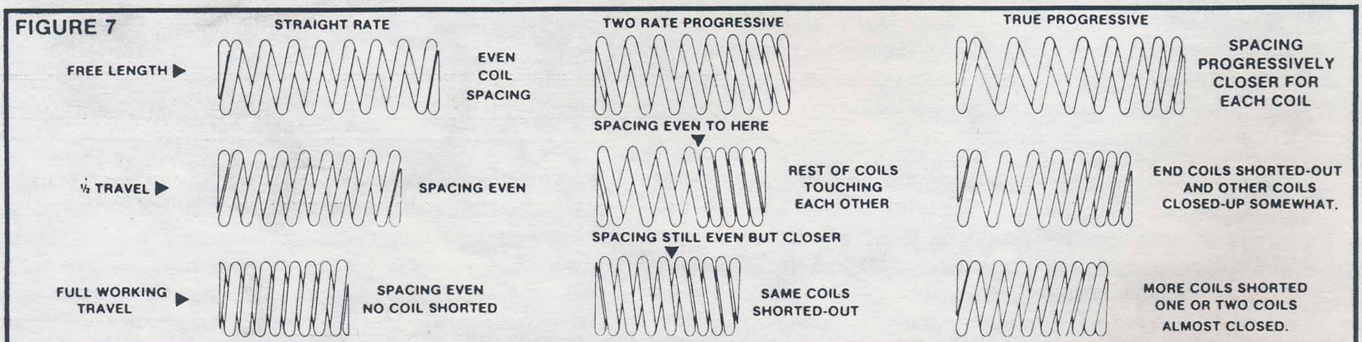
Overstress is a condition where you are making too little wire material to do too much work. The molecules actually slip in relation to each other. They don't return to their original position when the load is removed. A sign of this condition is when a spring takes a permanent set or sags-out by not recovering to its original free length after some use. The solution to this crisis is to change the quality of the wire. There are several different qualities of wire available beginning with **music wire,** progressing through various **oil tempered,** and ending with **chrome silicone** or **chrome vanadium.** The compromise decision is always one of cost. If a spring doesn't require a wire with a high stress capability, it is purely academic to use expensive, exotic wire. Most motorcycle springs are made from oil tempered wire with chrome silicon being used occasionally for the latest long travel suspensions.

The message of all this dialogue is that it is possible for a spring with few coils and small high quality spring wire to be every bit as stiff as a spring that is more massive. Don't assume because a spring **looks** light that it is not stiff.

So far we have been talking about springs that have just one rate. Today's motorcycles seem to respond to suspensions that have a soft rate for general conditions and a dramatic increase in rate to deal with the occasional larger bump. One way to accommodate this is to design a **progressive rate spring.** If you think back to our torsion bar, recall that the rate goes up if you shorten the bar. To make a progressive spring we must accomplish this shortening in stages as the spring is being compressed. The easiest way to do that is to take away active coils one at a time by having them short out on each other. This leaves fewer and fewer remaining active coils causing the rate to go up. **Figure 7** illustrates how the coils can be spaced in order to give various kinds of rate curves. The drawback to progressive springs is that it is difficult to get enough travel with all those dead coils stacked in the spring length.

Another approach is to stack two or three different springs together to get maximum utilization of the spring wire. This system provides more travel and, even more significant, a wider spread between initial and final rate. Remember when you stack springs together the effect is that of one spring with more coils. More coils mean a lower rate. That rate will be lower than either of the individual springs. If you add a third spring, the rate will be softer yet.

FIGURE 7



chapter ...2

The final rate will be the rate of one of the two springs, but not necessarily the rate of the stiffer of the two. The final rate will be the same as the spring that does not coil bind or short out. A combination of travel and rate determine which spring will still be active after the other has shorted out.

Here is a simple formula to determine what the initial rate will be if you stack springs together.

$$\text{Initial rate of combination} = \frac{\text{rate of primary spring X rate of secondary spring}}{\text{rate of primary spring} + \text{rate of secondary spring}}$$

If you have three springs solve for two of the springs first and plug the answer and the third spring into the formula and solve again.

We have dwelled on this lowering of rate for stacked springs to help clear up a misconception about fork booster springs. A booster spring does not make a spring stiffer, only longer, in fact it makes the spring softer. However if an extra spring is installed inside or outside another spring (in parallel instead of in series as found in valve spring sets), the rate does go up and is the sum of the two rates.

The amount of desirable progression has a lot to do with the type of shock mounting geometry incorporated. If your motorcycle has geometry that causes a diminishing rate you will need more progression than if you have a rising rate geometry. (Suspension Chapter 4).

Here are some general guidelines to help you select the proper spring progression. Suppose the optimum straight rate spring for your motorcycle is 100 lbs./in. For a progressive spring we would start with about an 80 lb./in. initial rate and end up with about a 160 lbs./in. rate after about 2/3 of the travel. That is a rate spread of 100% which is about right for most shock geometries. Laydown or diminishing mountings will need a spread greater than 100% and upright or maximum rise geometries will need less percentage of rate spread.

The damping characteristics also influence the amount of rate spread. A lot of compression damping will minimize the need for rate spread and also allow you to start with lower initial rates. Remember to always use an initial rate lower than the optimum straight rate. It is possible to measure spring-rates yourself. If you want to know if a spring is just harder or softer than your existing spring, you can count the coils and measure the spring wire to make a quick estimate. If you want to know the actual rate, this can be done with a bathroom scale and a drillpress. This operation can be dangerous so we suggest extreme care. **S & W DOES NOT RECOMMEND THAT THIS BE DONE BY THE INDIVIDUAL. BUT IF IT IS ATTEMPTED, EXTREME CAUTION SHOULD BE TAKEN AND ALL SAFETY PROCEDURES FOLLOWED TO THE LETTER. ALWAYS USE EYE PROTECTION AND OTHER PROTECTIVE CLOTHING.**

Place the bathroom scale on the drillpress table and brace the top of the scale so the load from the spring will be evenly distributed. Now make some kind of pushing device to fit in or over the drillpress chuck. Try to make the pushing device act as a guide to keep the spring from flying out. Now compress the spring a little to make sure the end coils are truly shorted out. Take your first reading off the scale. Compress the spring exactly one inch and take another scale reading. If your scale will handle it, compress the spring another inch to get an average or an indication of progression. If you subtract the initial reading from the second and the second from the third, you will have two spring-rates. Average the two for the actual rate. Be very careful to keep everything in line when performing this operation otherwise the spring will be likely to fly out of the drillpress.

That's enough about coil springs. Let's get onto air-springs. Air-springs have the unique quality of being naturally progressive. In practice, the initial rate can start lower and the final rate can end up considerably higher than is possible with conventional springs. Additionally the individual owner can perform easy adjustments to create an infinite number of spring-rate curves.

It sounds as if air-springs are the complete answer. But just like everything else in this world, you can't get something for nothing. The problems of air springs are many and only the individual who is willing to tolerate the annoyances will be able to enjoy the advantages.

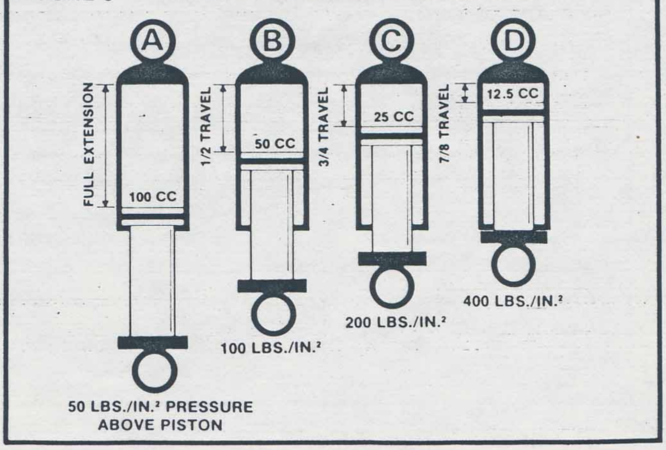
First we have the problem of making sure this air-spring device doesn't have any leaks. This requires careful design and that all the parts involved are kept in perfect condition. In some designs this also requires an investment in tanks, regulators, and gauges in order to monitor the pressure.

Air-springs are also sensitive to temperature. The addition of heat causes the air to expand, which causes more pressure which causes the spring-rate to change. Change in spring-rate due to temperature change is the achilles heel of most air-spring designs.

Another problem is the increase in seal friction caused by the internal air pressure pressing the seal against its sliding surface, such as in the case of air forks and piston type air shock absorbers. (Incidentally, this friction is not present in the S & W Air Adjustable Shock where the "rolling sleeve" type of air bag is almost frictionless.) This extra seal friction acts as excess damping and many times completely negates the benefits of the low initial spring-rate. In spite of all these and many more problems, air-springs still offer possibilities not available by any other method.

First let's see why air-springs are naturally progressive. The answer lies in one of the laws of physics. "Boyle's Law" states that if you compress a gas by reducing its volume, the pressure will rise inversely proportional to the change in volume. Simply stated, if you cut the volume in half, the pressure will double.

FIGURE 8 TYPICAL PISTON TYPE AIR SHOCK ABSORBER



chapter ...2

In **Figure 8,** we have applied Boyle's Law to a typical piston type of air shock configuration. The illustration shows an air chamber whose volume can be reduced by a piston moving up from the bottom. **Figure 8-A** shows the device fully extended. **Figure 8-B** shows the volume cut in half and the pressure doubled. **Figure 8-C,** the volume is cut in half another time and again the pressure is doubled (or four times the original pressure). Notice we had to move the piston only half as far this time to double the pressure.

Figure 8-D the volume is reduced by half once again, and this time it was only necessary to move the piston one quarter of the amount of change in travel **(Figure A & B)**. As you can see, the pressure quickly escalates near the end of travel. This pressure exerts a force on the piston to drive it back out of the cylinder and that is the spring resistance you feel. The amount of force you feel is the multiplication of the pressure times the area of the piston.

Our illustration shows a cylinder with a very high compression ratio. (Compression ratio is the original or starting volume divided by the volume remaining at the end of travel.) **Figure 9** charts the spring-rate curves of several different compression ratios. In all cases, the amount of stroke or change to the volume is the same (80 c.c.). The starting volumes do vary and of course the final volume changes accordingly. This kind of volume change can be accomplished with an air-fork or air shock by the simple addition or subtraction of oil fill.

Notice the final spring-rates are effected the most by compression ratio changes. Study the chart carefully to appreciate how sensitive air-springs are to volume changes. Especially notice the dramatic final rate increases associated with the higher compression ratios.

The extreme rise in pressure can sometimes be more than is desirable. Some of the current air-spring designs now incorporate some method to allow the volume in the cylinder to stretch or increase after a certain pressure is reached. This is accomplished by a secondary floating piston or a diaphragm with adjustable pressure behind it. The theory is to create a gentler increase in spring-rate near the end of travel when higher starting pressures are used and in fact this second adjustment gives an infinite number of combinations.

Another approach, common in front forks today, is to augment air-springs with a soft straight-rate coil spring. This is done for two reasons. First, less pressure is required to keep the motorcycle at an acceptable ride level. (If you start with less pressure, the final rates will not be so severe.) Secondly, the problem of seal friction is reduced and the forks regain sensitivity over small bumps.

The starting pressure is another tuneable adjustment, but it effects the spring-rate in quite a different manner than the compressed ratio. **Figure 10** charts the effect of several different starting pressures. The change in volume is again 80 c.c. and the starting volumes remain constant.

Notice the changes to starting pressure effect the initial rates more than was the case with changes to the compression ratio. Notice also that they effect the final rates much less than compression ratio changes. By juggling starting pressure and starting volume you can tailor a spring-rate curve that exactly matches the demands of your motorcycle.

FIGURE 9

DIFFERENT PRESSURE CHANGE CURVES (OR RATE) DUE TO VARIOUS COMPRESSION RATIOS

STARTING VOLUME	TRAVEL	VOLUME	PRESSURE LBS./IN.²	PRESSURE CHANGE LBS./IN.²
150 C.C.	EXTENDED	150 C.C.	10.00	—
	1/4	130 C.C.	11.54	1.54
	1/2	110 C.C.	13.64	2.10
	3/4	90 C.C.	16.67	3.03
	FULL BUMP	70 C.C.	16.67	3.03
	FULL BUMP	70 C.C.	21.43	4.76
100 C.C.	EXTENDED	100 C.C.	10.00	—
	1/4	80 C.C.	12.50	2.50
	1/2	60 C.C.	16.67	4.17
	3/4	40 C.C.	25.00	8.33
	FULL BUMP	20 C.C.	50.00	25.00
90 C.C.	EXTENDED	90 C.C.	10.00	—
	1/4	70 C.C.	12.86	2.86
	1/2	50 C.C.	18.00	5.14
	3/4	30 C.C.	30.00	12.00
	FULL BUMP	10 C.C.	90.00	60.00
85 C.C.	EXTENDED	85 C.C.	10.00	—
	1/4	65 C.C.	13.08	3.08
	1/2	45 C.C.	18.89	5.81
	3/4	25 C.C.	34.00	15.11
	FULL BUMP	5 C.C.	170.00	136.00

FIGURE 10

DIFFERENT PRESSURE CHANGE CURVES DUE TO VARIOUS STARTING PRESSURES

STARTING PRESSURE	TRAVEL	VOLUME	PRESSURE LBS./IN.²	PRESSURE CHANGE LBS./IN.²
5 LBS./IN.²	EXTENDED	100 C.C.	5.00	—
	1/4	80 C.C.	6.25	1.25
	1/2	60 C.C.	8.33	2.08
	3/4	40 C.C.	12.50	4.17
	FULL BUMP	20 C.C.	25.00	12.50
10 LBS./IN.²	EXTENDED	100 C.C.	10.00	—
	1/3	80 C.C.	12.50	2.50
	1/2	60 C.C.	16.67	4.17
	3/4	40 C.C.	25.00	8.33
	FULL BUMP	20 C.C.	50.00	25.00
15 LBS./IN.²	EXTENDED	100 C.C.	15.00	—
	1/4	80 C.C.	18.75	3.75
	1/2	60 C.C.	25.00	6.25
	3/4	40 C.C.	37.50	12.50
	FULL BUMP	20 C.C.	75.00	37.50
20 LBS./IN.²	EXTENDED	100 C.C.	20.00	—
	1/4	80 C.C.	25.00	5.00
	1/2	60 C.C.	33.33	8.33
	3/4	40 C.C.	50.00	16.67
	FULL BUMP	20 C.C.	100.00	50.00

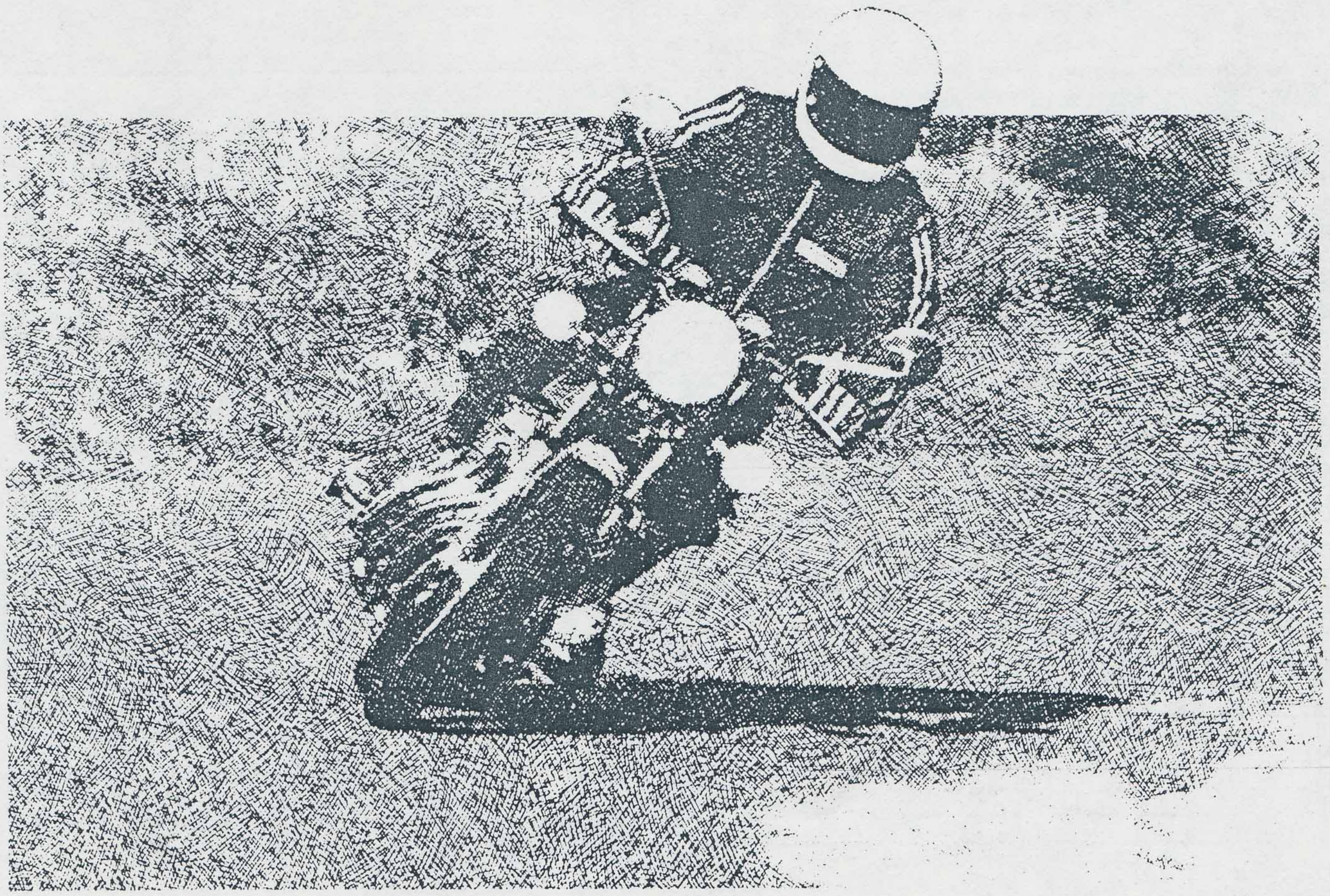
chapter ...2

Let's see how we might apply all this theory directly to your motorcycle, and particularly to your front forks. Suppose while test riding you conclude the forks are too stiff and don't travel the full amount, so you lower the starting pressure. Suppose this change doesn't help the feel very much and the fork travel is still incomplete. This is a good indication the problem is actually too much oil fill. Drain out a little oil and make another pass over your favorite motocross course or road. A possible result could now be a general improvement and complete travel but also a loss of damping control over small bumps (chatter). This indicates the forks need the extra oil to keep the damping parts submerged at all positions of travel. The solution now is to add oil to restore the oil level to its original height. Then provide more air space above the oil by adding a reservoir or extending the fork caps. With the oil level optimized you can once again survey various starting pressures. You may also have to play with the volume of the new reservoir in order to get the compression ratio just right.

A side effect to these new spring-rate curves may be that the hydraulic damping may have to be adjusted to match. Many times a new spring-rate curve will be sabotaged by incorrect damping. Don't disregard any "pressure/fill" combination until you are confident the damping has first been optimized. This can normally be accomplished by changing the oil viscosity.

Another possible condition is that no matter how high you make the starting pressure, the motorcycle forks will bottom-out. This indicates too low a compression ratio. Add some extra oil to correct this condition. If you find that you can't get a combination that works freely over little bumps without bottoming you might consider a light helper spring to augment the air-spring. Be sure to use slightly lower starting pressures in conjunction with a helper spring.

When performing these tests keep in mind the various combinations of spring-rate curves overlap and interact in an intricate manner. Any single adjustment may require one or two adjustments of other elements in order to completely evaluate your results. The perfect combination may be elusive but very satisfying once it is achieved.



Chapter... 3

Wheel Rates

This chapter will get right into specifics by covering the most controversial and talked about area of motorcycle suspension: the rear suspension on off-road bikes.

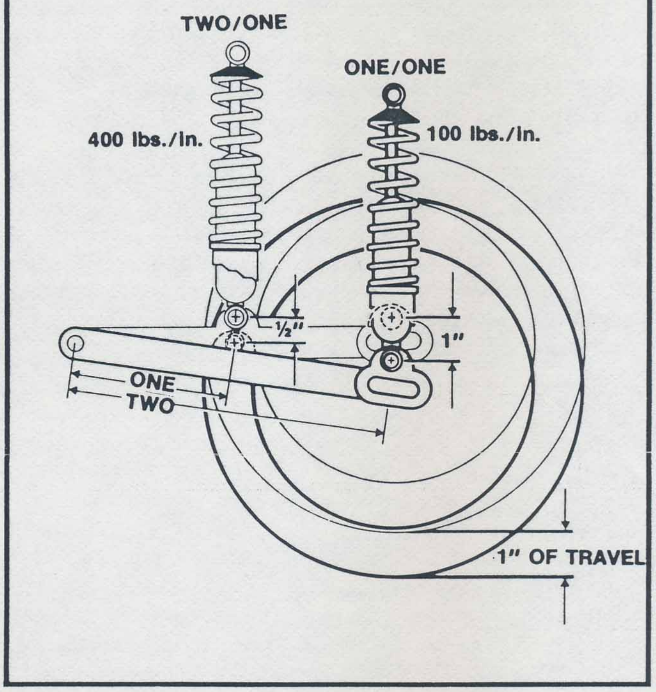
By now it has become pretty well accepted that longer wheel travel will get you over rough terrain more quickly. Historically the first attempts at getting increased wheel travel were accomplished by moving the shock mounting position farther up the swing arm away from the rear axle. That wasn't exactly the ideal way to do things, but the available shocks were limited to 12 to 13 inches of overall length, and about three inches of travel. If you wanted six inches of travel at the rear wheel you had to find a way to double the amount of travel available in the shock absorber. Getting that amount of travel from a shock with only three inches of travel of its own meant that you had to create a mechanical advantage of two to one. That advantage is called the *leverage ratio.*

Moving the mounting position up the swing arm solved the problem of increasing wheel travel, but many more problems were created. Swing arms and frames bent and broke under the stress of forces thay were not designed to accept. Spring selection also left many people baffled. It seemed logical that moving the shock mounting position halfway up the swing arm would require a spring twice as stiff, but a lot of experimentation proved that it was necessary to have a spring four times as stiff. Why? Take a look at **figure 11**. Illustrated is a swing arm with two different shock mounting positions. The rear mount will give a leverage ratio of 1:1, the forward mount will give a ratio of 2:1.

Let's say we want this suspension to resist with 100 pounds of force for every inch we raise the rear wheel. Using a 100-lb./in. spring in the rear mounting position will achieve that. But what spring rate in the front position will give the same resistance?

We know that the leverage ratio is 2:1, so it seems logical to use a 200-lb./in. spring. But let's move the wheel up one inch and look again. Because of the 2:1 leverage ratio we for sure need 200 pounds of force at the forward shock mounting to give 100 pounds at the wheel. If you measure the distance the shock mounting moves upward when you move the wheel one inch, you will find that it is only ½ inch. If you compress our "logical" 200-lb./in. spring ½ inch you only get half of its spring rate, which works out to 100 pounds. That is only half of the amount we know we need to make 100 pounds at the wheel. In order to get 100 pounds at the wheel, we will have to double our "logical" spring rate with a 400-lb./in. spring. Now if you square our original leverage ratio of 2:1 (multiply it times itself), you get a ratio of 4:1, which, coincidentally, is the same as for our two alternate springs (400/100).

FIGURE 11



No matter what the leverage ratio you must square it before trying to compute springs to give a particular action. The chart in **figure 12** will make it clear that as you use higher and higher leverage ratios this "squaring" procedure becomes more and more important. Also, the accuracy when determining the exact leverage ratio is increasingly critical.

Look at the column for the 1.1:1 ratio. When it is squared the result is not much different from the original number. But, the ratio of 1.9 squared is almost twice as much. Small changes in shock mounting, when already dealing with high leverage ratios, will require much bigger adjustments in spring rate.

FIGURE 12

SPRING REQUIREMENTS FOR VARIOUS LEVERAGE RATIOS

	100%										200%		
LEVERAGE RATIO	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7	1.8	1.9	2.0	2.1	2.2
(LEVERAGE RATIO)^2	1.0	1.21	1.44	1.69	1.96	2.25	2.56	2.89	3.24	3.61	4.0	4.41	4.84
SPRING RATE REQUIRED TO MAINTAIN A CONSTANT 100 lb./in. WHEEL RATE	100	121	144	169	196	225	256	289	324	361	400	441	484
	100%										400%		

Note that for an increase of 100% in Leverage Ratio the spring requirement goes up 400% and for low ratios the squaring effect is not very pronounced.

Leverage ratios in this range are not uncommon with today's Motocrossers.

chapter ...3

WHEEL RATE

Now that we know how to select the right spring for the job, how do we know what degree of "stiffness" or "softness" we want throughout the system? Suspension engineers use a term called *wheel rate* to allow riders to put a number on the type of feel they like in rear suspension. In the example we used we said we wanted a 100-pound force to move the wheel one inch, or a wheel rate of 100 lb./in. Wheel rate is the amount of force needed to move the wheel straight up one inch. No matter what type of suspension system is used, the wheel rate measurement allows you to know in advance what the bike will feel like. It is as though each rider carries a wheel rate figure tattooed to his brain. The closer a bike is to that rate, the better he likes it.

The outer limits of wheel rate are a low of 50 pounds and a high of 100 pounds. Actual rider preferences fit into a much narrower range of from 60 to 80 lb./in. There are several factors that can modify wheel rate preference.

A 125 racer will need less wheel rate than an open class bike and a 250 will fall somewhere in between. Big, rolling, sandy whoop-de-doos will require more wheel rate than hard washboard surfaces. Rider weight has a great deal of influence, too. If you stand up instead of sitting in the saddle you can lower the wheel rate, but if you lock your knees while standing the wheel rate will have to go up again. If you ride far forward on the gas tank the wheel rate can go down, and the converse is also true. The aggressiveness of your riding will likewise influence wheel rate. Still another factor is the amount of compression damping in the shock absorber. This is impossible for the individual to determine precisely, but some shocks are known to have more or less compression damping. A lot of compression damping will assist the spring and allow you to lower the wheel rate.

If you use these factors as guidelines and do some testing on your own motorcycle, you will arrive at a perfect wheel rate for your riding. If you already have a motorcycle with a perfect spring combination, work backwards with the formulas shown later on in this chapter to come up with your personal wheel rate. Then in the future, if you get a new motorcycle or change geometry, you can apply your personal wheel rate to select springs and be very close to optimum.

A few years ago, in private testing, S & W had the opportunity to have a National Motocross Champion test two bikes on the same day. The bikes were identical with the exception of the rear suspension; one had conventional shocks moved up to a forward mount, the other had longer shocks in a more laid-down position. After a long day of experimenting, we got both bikes to his liking. Even though the two suspension geometries were quite different and used different springs, the preferred wheel rate on both setups was 62 lb./in.

Working with another National Motocross Champ who was in the process of switching from a monoshocker to a conventional motocrosser, we found that what he preferred in the conventional rear suspension was within two pounds of the monoshocker in wheel rate. This time it was in the 80 lb./in. region, a rate that suited his riding style better than a softer one.

COMPUTING WHEEL RATE

Now that we know what wheel rate is, how do we find out what rate a given motorcycle has? The first thing we will have to know is the leverage ratio. We have already given a very simplified example of leverage ratio. In the real world shocks are positioned at a lot of weird angles, and the best method is to measure directly on the bike. To do so, put the bike on a steady centerstand. Remove both rear shock absorbers. Support the rear wheel so that it hangs in about the same position as when the shocks are installed **(figure 13.)**Make a mark on the rear fender directly over the axle as a reference point. Measure the distance between the center of the axle and the mark on the fender. This distance will be referred to as *A*. Without moving anything measure the distance between the upper and lower shock mounts. This will be known as *B*. Now compress the suspension until the tire hits the fender (refer to **figure15**). You can use a tie-down strap draped over the seat to hold the suspension up. Now make the same two measurements with the suspension compressed. We will call this dimension *a* and *b*. Now subtract *a* from *A* and *b* from *B*. Divide the result of *b* from *B* into the result of *a* from *A* and you get your leverage ratio.

EXAMPLE

Measurements from motorcycle: **A = 15 in. B = 13.5 in. a = 7 in. b = 9.5 in.**

$$\textbf{Wheel travel} = \begin{array}{r} A \\ -\,a \\ \hline \end{array} \text{ or } \begin{array}{r} 15 \\ -\,7 \\ \hline 8 \end{array} \qquad \textbf{Shock travel} = \begin{array}{r} B \\ -\,b \\ \hline \end{array} \text{ or } \begin{array}{r} 13.5 \\ -\,9.5 \\ \hline 4 \end{array}$$

$$\textbf{Leverage ratio} = \frac{\textbf{wheel travel}}{\textbf{shock travel}} = \frac{8}{4} = 2$$

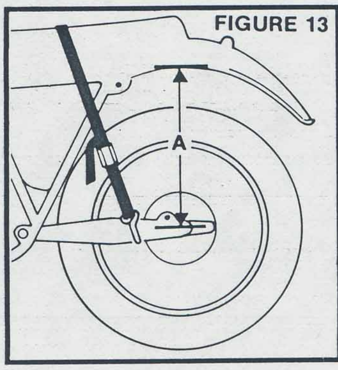


FIGURE 13

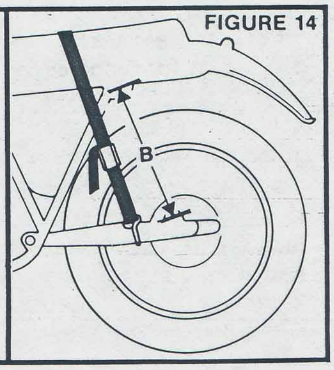


FIGURE 14

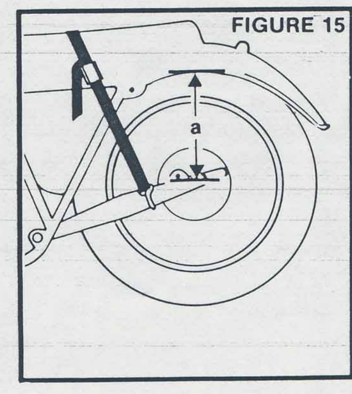


FIGURE 15

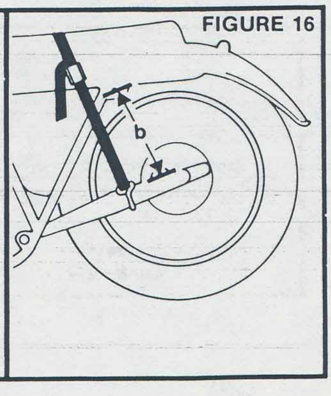


FIGURE 16

chapter ...3

This may seem like a roundabout method, but if the procedures are performed with care, it is the only method that takes all angles and shock lengths and swing arm arcs into account.

With the leverage ratio and your estimated wheel rate you now have enough data to compute spring rates using the following formula:

$$\textbf{Spring rate} = \frac{\textbf{(leverage ratio)}^2 \textbf{ x estimated wheel rate}}{\textbf{number of springs in suspension system}}$$

EXAMPLE

Collected data:
leverage ratio = 2
estimated wheel rate = 65 lb./in.
number of springs = 2 (monoshocks have only one spring)

$$\textbf{Spring rate} = \frac{(2)^2 \text{ X 65 lb./in.}}{2} = \frac{\text{2x2 X 65 lb./in.}}{2} =$$

$$\frac{\text{4 X 65 lb./in.}}{2} = \frac{\text{260 lb./in.}}{2} = \text{130 lb./in.}$$

Thus, 130 lb./in. is the correct spring rate to give a wheel rate of 65 lb./in. If, when field testing, you find that your best wheel rate is far out of the range suggested earlier (60 lb./in. to 80 lb./in.), you probably have other parts of your suspension set up incorrectly. **Wheel rate is fundamental** and should not be deviated from very much. Adjust other parts of the suspension to suit the correct wheel rate. We will analyze all of the other aspects of rear suspension in future chapters.

If you want to solve for wheel rate rather than spring rate, use the following formula:

$$\textbf{Wheel rate} = \frac{\textbf{spring rate X number of springs in system}}{\textbf{(leverage ratio)}^2}$$

In summary, we have learned that spring rate is outdated as a measure of rear suspension performance. The advent of suspensions that multiply movement have changed this concept. The real figure you are interested in is the wheel rate. This is a more realistic concept in that it measures the actual force at the axle. By computing the leverate ratio of your machine, and then converting it to wheel rate, you have a basis for fine-tuning your suspension.

Chapter... 4 Frame Geometry

It's time to talk about some of the subtleties that make the difference between an adequate rear suspension and a superior one. This chapter will also be a do-it-yourself manual for more enterprising racers looking for a little extra edge in performance.

The last chapter dealt with shock-mounting leverage ratios and calculations of wheel and spring rates. Determining the (leverage) ratio between the vertical wheel movement and the movement of the shock absorber is essential even if you plan to forego major structural changes and only adjust shocks and springs.

RISING AND DIMINISHING RATES

An average leverage ratio taken for the complete wheel travel does not tell the whole story. In fact, in most cases, the leverage ratio will change a little bit with every inch you move the wheel. Some suspensions become stiffer the more you compress them *(rising rate);* others begin to get softer *(diminishing rate).*

In researching geometry for motocross and off-road use, you will find many conflicting opinions as to the superiority of a rising or a diminishing rate. Obviously the motorcycle manufacturers don't agree on what is optimum. We lean heavily in favor of rising rate geometry. But before getting into our reasons, let's take a look at the physical differences in various motorcycles that determine a rising or a diminishing rate.

Studying the rear shock mounts of today's motocrossers will reveal an array of angles for leaning the shocks forward. These angles determine whether a suspension system has a rising or a diminishing rate. As a rule of thumb, if a shock is leaned forward about 20 degrees from vertical it will have a rising rate. If the shock is laid down farther the rate will begin to stay constant throughout its travel; and if it is laid down still farther, a diminishing rate will result. Oddly enough, if you begin making the shock more vertical than 20 degrees, the rates change and deteriorate in exactly the same way they do when laying the shock down.

If you make changes to the shock angle by moving just one end of the shock, you not only change the rise characteristic, but also the basic leverage ratio. If that is done, the difference in rise characteristic will be overshadowed by the new leverage ratio and evaluation will be misguided. It **is** possible to retain a basic leverage ratio and to alter the rise characteristic independently.

We are sure that many of you have heard or read that it is beneficial to have a shock pointed at some magical angle, such as "more towards the center of gravity," or "at the steering head," or "in the direction the motorcycle is traveling," or "directly into the rider's body." We say baloney to all of those statements. It may be true that all of the motorcycles that received such treatment were improvements over their predecessors, but not for those reasons. The only thing the motorcycle knows and feels is how the rear tire interacts with the ground, and that is partially determined by the rate at which the wheel can move in relation to the chassis. We believe that you can ignore the direction your suspension unit is mounted, **provided** the leverage ratio is correct and you have designed in maximum amount of available rise in rate.

IN FAVOR OF RISING RATE

Why are we so in favor of rising rates? A quick look at the springs on most professional motocrossers gives the first clue. Many are of the double spring type. Some motorcycles with heavily laid-down shocks (or diminishing rates) use double springs. A double spring gives a very soft rate in the beginning and a significant increase in rate toward the end of travel *(rate progression).* This benefits the rider because the suspension can move easily over small and medium bumps, but rise up in rate enough not to bottom over larger bumps. The **S & W "FE" STROKER DUAL SPRING** is a good example of this type.

If double springs work so well why bother with geometry that changes the leverage ratio? There are two reasons. First, it is physically difficult to manufacture springs with enough rate spread to compensate for a diminishing rate geometry. Second, it is much more difficult to produce shock damping that is matched with a low spring rate at one end and a high rate at the other.

Rising rate geometry doesn't completely eliminate the need for dual springs. At best the most change in leverage ratio that can be expected is about 20 percent. That may not sound like a lot, but if you have a 20 percent diminishing rate to begin with, a 20 percent rise in rate will help the shock problem by 40 percent. Also recall from the last chapter that leverage ratio must be squared in order to predict the rate at the wheel. If you square two ratios that are 20 percent different, you end up with a 44 percent difference at the wheel. Note that a leverage ratio that diminishes (numerically) will give an increasing wheel rate.

Proof that 20 percent improvement from the geometry is significant was dramatically demonstrated to us in a shock test conducted last year. The test bikes were a 450 Maico AW and a 400 KTM. Both motorcycles have the same wheel travel and use the same free-length shock absorber, suggesting they have the same average leverage ratios. However, the KTM shock absorbers are leaned forward considerably more than the Maico's, giving the KTM a diminishing rate. The test results were completely predictable. The spring combination that gave a nice soft ride yet did not bottom out on the Maico, was too stiff over small bumps and bottomed out on the KTM. The performance of still other confirms in our minds the need to take advantage of rising rate geometry. Late model Can-Am motocrossers suffer the same symptoms as the KTM. They are too easily bottomed and not all that smooth over the small stuff. Pre-1976 Husqvarna GPs are another example. In fact, on '76 models, Husky saw fit to move the shocks to a more upright attitude in order to rectify the problem. Kawasaki did the same on its works team bikes. Honda is still fighting the bottom/softness compromise. The shocks on the Suzuki RM-B are more upright than those on A models. Finally, the factory KTMs of Moissiev and Kavinov have shock mountings that are much more upright than those on the KTMs seen in the U.S. in 1977.

chapter ...4

To learn how the rising rate phenomenon occurs, study **figure 17.** The three drawings show the same suspension in three different positions of travel. First note that the leverage ratio is computed by dividing the effective shock lever into the effective swing-arm lever. Note also these distances are measured at 90° to the reaction point. The set of drawings shows how the effective lengths change during travel. The numbers are real and represent a typical amount of rise in rate even though the drawings are not to scale. We remind you that a numerically reducing leverage ratio gives a rising rate.

DESIGNING GEOMETRY

Now that we all agree that rising rate geometry is a desirable characteristic, let's begin designing a perfect geometry for your existing motorcycle. The highest priority is obtaining maximum wheel travel without sacrificing anything else. First you must make a survey of your own motorcycle to determine existing limitations.

Begin the survey by placing your motorcycle on a box or stand and removing the shock absorbers. Now you can swing the rear suspension up and down manually and get a better look at the problems. Can you safely let the swing arm hang down farther without getting too much angle in it? Will the chain clear the top of the swing arm? If you let the swing arm droop farther it will raise the back of the motorcycle and the seat. Is that tolerable for your size and type of riding? If the back of the motorcycle is raised, steering angle (rake) will be reduced. Can you raise the front end enough to compensate? All of these things and more must be considered before you commence cutting and hacking. Maybe you don't have to droop out the swing arm. Maybe the fender and seat can be reconfigured to allow the wheel to travel upwards a greater amount. At any rate, you must now check the chain and its clearance under the swing arm.

Your first objective is to end up with a minimum of 10 inches of wheel travel. Most team bikes are now in the 11 to 12-inch range. Next, you will want to keep the swing-arm droop angle at a minimum. If you measure down to the floor from both the swing-arm pivot and rear axle, there should be about a four-inch difference between the two measurements. Some motorcycles have more droop than that, but too much swing-arm angle will cause excessive *anti-squat.* Anti-squat is a force created when power is applied that tends to raise the rear of the motorcycle, or, conversely, push the tire into the ground. This effect in small amounts helps traction, but when excessive, anti-squat generates unwanted wheelspin and pogoing. The swing-arm angle is one of the components that contributes to the amount of anti-squat generated.

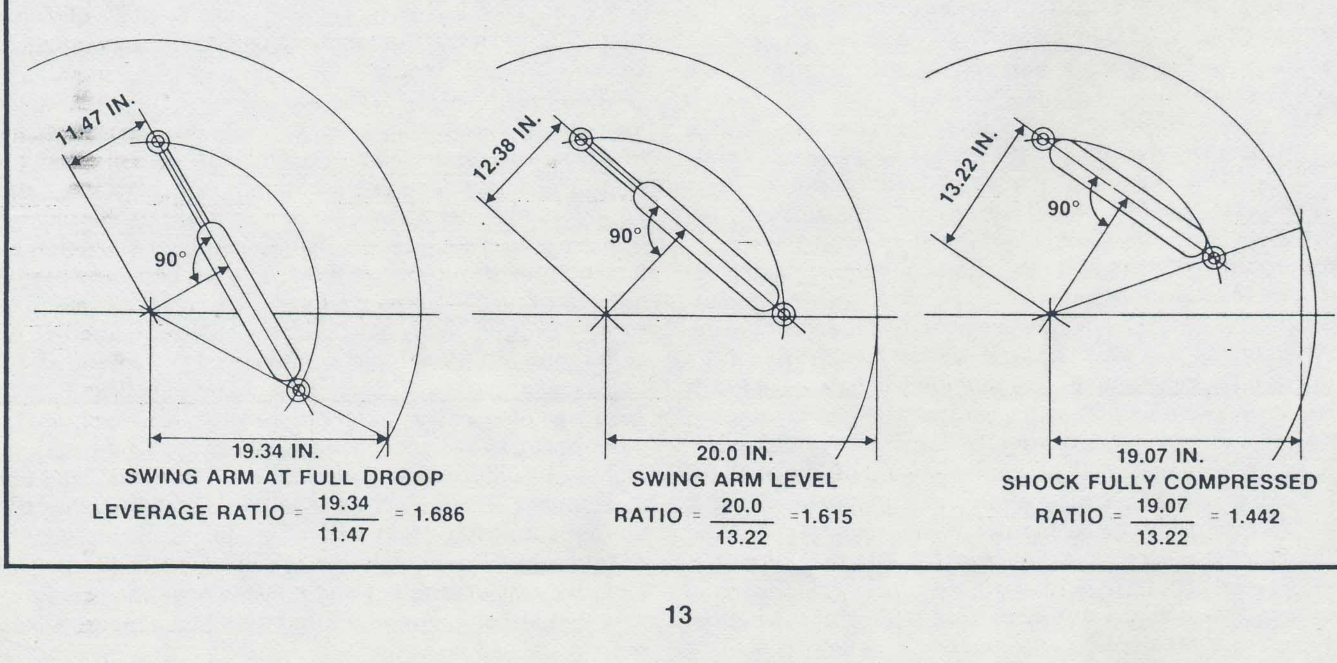
The question of anti-squat is covered in a future chapter, but for the moment be aware of how excessive anti-squat is determined. In **figure 18** you will see wheel travel divided into two parts: that above the horizontal swing-arm position (bump travel), and that below horizontal (droop travel). It is good practice to keep droop travel to about 35 to 40 percent of total travel, and bump travel to about 60 to 65 percent of the total. Keep these figures in mind but don't be too much of a fanatic about them. Use your own judgement.

Another thing to consider at this time is whether swing-arm length needs to be altered. Maybe the general handling is too responsive and a longer swing arm is in order. If it seems right for improved steering and weight transfer do it now as it will also help deliver more wheel travel without excessive swing-arm angle.

FIGURE 17

MECHANICAL LEVERAGE AND HOW IT CHANGES DURING WHEEL TRAVEL

$$\text{LEVERAGE} = \frac{\text{SWING ARM}}{\text{SHOCK ARM}}$$



SWING ARM AT FULL DROOP

$$\text{LEVERAGE RATIO} = \frac{19.34}{11.47} = 1.686$$

SWING ARM LEVEL

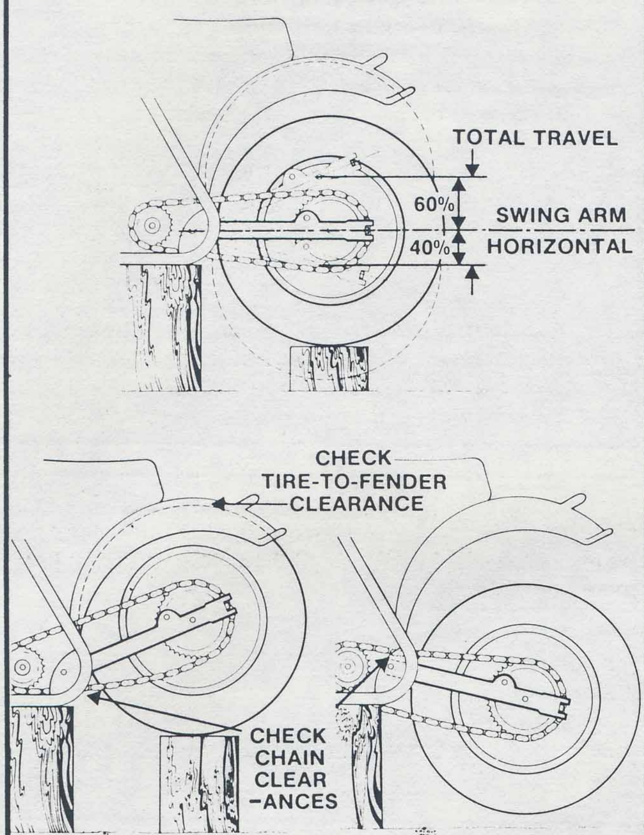
$$\text{RATIO} = \frac{20.0}{13.22} = 1.615$$

SHOCK FULLY COMPRESSED

$$\text{RATIO} = \frac{19.07}{13.22} = 1.442$$

chapter ...4

FIGURE 18 LIMITS TO REAR WHEEL TRAVEL



FULL BUMP POSITION

FULL DROOP POSITION

CHOOSING THE CORRECT SHOCK

The next step is to check the S & W Specification List for possible units.

The first thing to consider is travel. Try to choose a shock with a lot of travel such as the L, LL and FE series. The pro blem here is that we can't build in a lot of travel without increasing the overall extended length of the shock. In fact, the best a manufacturer can do is increase the shock travel only half the amount we add to the extended length. Maybe the shock travel you want comes with such a long extended length that there is no way to fit the unit to your motorcycle.

If that is the case, use a shorter shock and increase the leverage ratio.

But if you use a high leverage ratio, make sure the shock has adequately high damping for the application.

When you think you have decided on a particular shock, divide its travel into the travel you feel you can build into the rear wheel. This will give you the leverage ratio you must use for mounting. Check again to see if the damping is correct for that ratio.

CALCULATING CRITICAL DIMENSIONS

Before you go any farther a little math is in order. You know your leverage ratio and you can measure the swing arm to get the length of one of the two levers. In order to determine the length of the shock lever arm you must divide the leverage ratio into the swing-arm length.

EXAMPLE

Planned wheel travel: 10 in.
Selected shock travel: 5.75 in.
Swing-arm length: 20 in.

$$\textbf{Leverage ratio} = \frac{\textbf{planned wheel travel 10 in.}}{\textbf{selected shock travel 5.75 in.}} = \textbf{1.74 in.}$$

$$\textbf{Shock lever arm} = \frac{\textbf{swing arm 20 in.}}{\textbf{leverage ratio 1.74 in.}} = \textbf{11.49 or } 11\tfrac{1}{2}\textbf{ in.}$$

Once you have these figures you can make a diagram on a large piece of paper to determine where to mount the shock on the swing arm. Refer to **figure 19** to better visualize this procedure. The hypothetical shock we selected with 5.75 inches of travel will probably have an extended length of 16.5 inches.

Figure 19 illustrates a triangle on its side, and this is the ideal geometry for maximum rate rise. To draw your own optimum rate rise geometry, follow these steps, working full size to scale.

A. Draw a straight line at about 35° to 50° from vertical, this is an average angle for the shock position. Mark the extended center-to-center length of your selected shock on this line and then bisect (divide by two) to determine the halfway point.

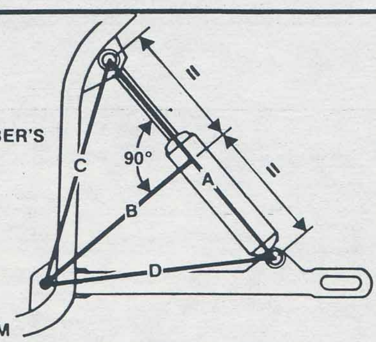
B. From that halfway point, project a line 90° from line A. The calculated length of your shock lever arm determines the length of this second line, line B. Mark line B with a short dash to indicate this length.

C. From this short dash on line B, draw another line to intersect line A at the top mark. This is line C.

D. Draw a similar line from the point where B and C connect to the bottom mark on line A.

What you should now have before you will be a triangle lying on its side. Intersection A and C is the top shock mount, intersection A and D is the bottom shock mount, intersection B, C and D is the swingarm pivot. The important thing to remember is that maximum rate rise will only occur when those two dimensions are equal, that is, the length of lines C and D.

FIGURE 19
HOW TO DETERMINE IDEAL POSITION OF SHOCK MOUNTS



A IS THE SHOCK ABSORBER'S EXTENDED LENGTH

B IS THE CALCULATED LENGTH OF THE SHOCK LEVER ARM

C IS THE DISTANCE FROM THE SWING-ARM PIVOT TO THE UPPER SHOCK MOUNT

D IS THE DISTANCE FROM THE SWING-ARM PIVOT TO THE LOWER SHOCK MOUNT

chapter ...4

The next step is to transfer these dimensions to the motorcycle in order to fabricate the mounts. To do this you need a device capable of drawing arcs with radii as long as those in your calculated dimensions. A string compass will be adequate for this purpose. To make a string compass tie the string into a loop and stretch it around two pencils. Retie the string until the distance between the pencils is the same as the length of the sides of your triangle. **Figure 19A** gives a clearer view.

Now if you hold one pencil stationary right over the center of the swing-arm pivot you can swing the other pencil in a circle and draw two short arcs: one in the general area for the upper shock mount and the other across the swing arm, where the lower mount will be. **Before drawing these arcs it might be useful to tape some cardboard over the two general areas on the motorcycle to help identify the lines. Figure 19 B illustrates the arcs to be made with the string compass.**

Next reset the string compass so that its radius is the same as the extended length of the shock absorber you have chosen. Take a close look at the arc drawn on the swing arm and select the spot you want to be the center of the lower shock mount. For the next arc this spot will be the stationary end of the string compass. Swing an arc from this point across the upper arc already drawn. The point at which the two arcs cross is the correct location for the upper shock mount. To verify that you have done this procedure correctly, take your paper triangle and hold it up to the motorcycle. The tips of the triangle should coincide exactly with the shock mount points and the swing-arm pivot. That is the complete process. All that is left to do is to refer back to Chapter 3 to compare the appropriate spring rate for your new geometry. And, of course, you must weld in some new shock brackets.

FIGURE 19 A

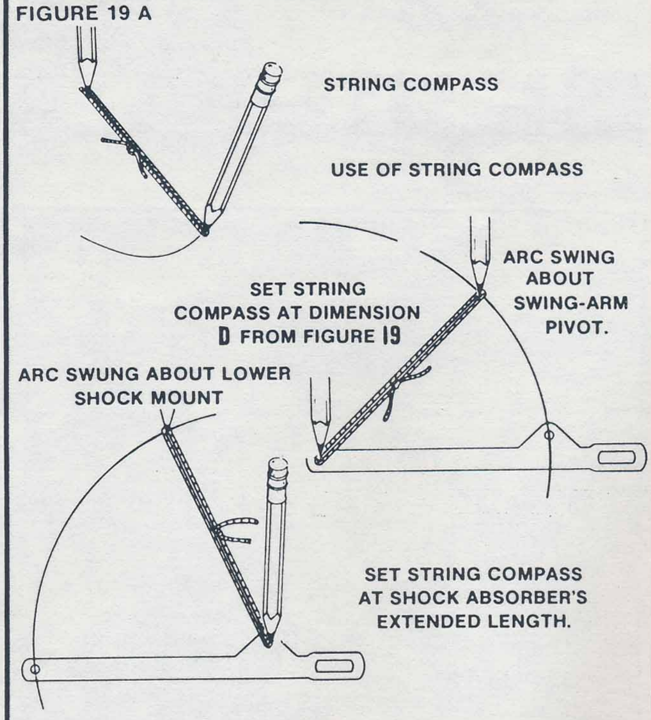
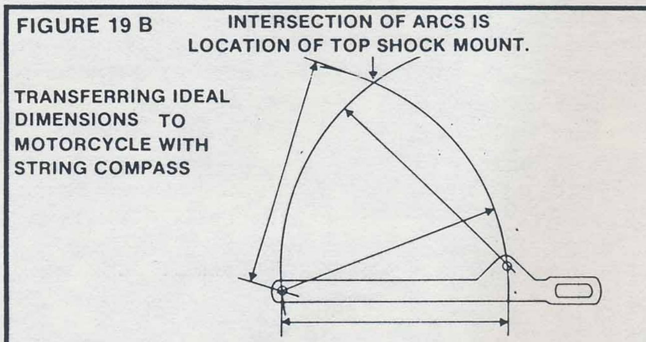
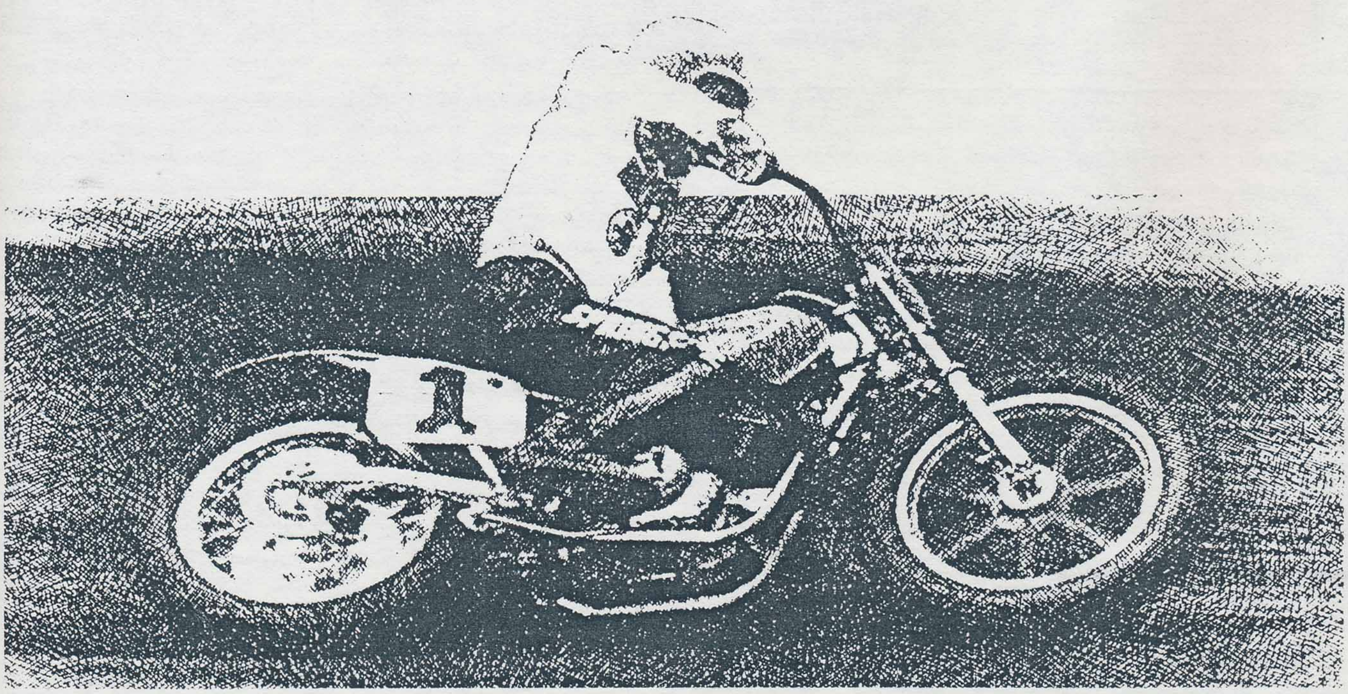


FIGURE 19 B





Chapter ...5 Front Forks

Once again we bring you another chapter in the continuing saga of motorcycle suspension. Our subject is the inner workings of front forks, which we selected for two reasons. First, the individual owner has easy and inexpensive access to the various adjustable parts of the front fork system. Secondly, the performance of the motorcycle is very responsive to meddling of this type. The chapter should give you the incentive to obtain all the performance potential of your forks and at the same time supply the caution necessary to avoid backward steps.

A fork's primary functions are springing, damping, and to serve as structural linkages connecting the handlebars to the front wheel. Modification to the structural parts of forks is out of the reach of the individual and subsequently out of the scope of this chapter. We will focus only on the springing and damping functions, beginning with damping. Springing is dealt with in Chapter 2. Study the illustrations carefully while reading this dialog. Refer to them as often as necessary and don't read on without being able to picture in your mind each movement of oil and the way it happened. Begin by studying **Figure 20** to familiarize yourself with the nomenclature of each part. Use **Figure 20** as a reference to follow the text.

Before we go any further we must clarify a few basic laws of hydraulics. The first fundamental concept is hydraulic fluid (or oil) is not compressible. It is important to remember this because the lack of compressibility requires the oil must move from compartment to compartment as the fork moves through its travel.

The next concept of importance is that oil will always follow the path of least resistance. If one hole exiting a chamber is larger than another, a much greater volume of oil will escape through the larger hole because it offers less resistance. A fork has many compartments with various exits. Some of the exits are not there by design, like leakage past the piston ring, but all the exits, planned or not, contribute to the amount of force generated when oil is pumped between compartments.

Another rule to keep in mind is hydraulic fluid will always migrate in a manner that will equalize the pressure throughout. If you move a piston in a hydraulic cylinder that has oil on both sides of the piston you will create a negative pressure on one side and a positive pressure on the other (or pressure differential). If the piston has passages to allow oil to migrate, the oil will travel from the high pressure side of the piston to the low pressure side until the pressures are equalized.

The last law of hydraulics that we must remember is when a fluid flows through an orifice (or hole), the pressure causing the flow varies as the square of the (piston rod) speed. When oil is passed through an orifice resistance is generated that results in a pressure build-up inside the chamber holding the source of the oil. This pressure is the damping force — the force you feel when you push a fork or shock in-and-out by hand. According to our last law this resistance increases drastically as the piston speed increases.

Look at the little chart below to get a better feel for the magnitude of damping change associated with squaring of the piston speed. The numbers are completely arbitrary but notice the jump in damping force for each step up of piston speed. The piston speeds are doubled for each step but the increase in damping force is much more severe.

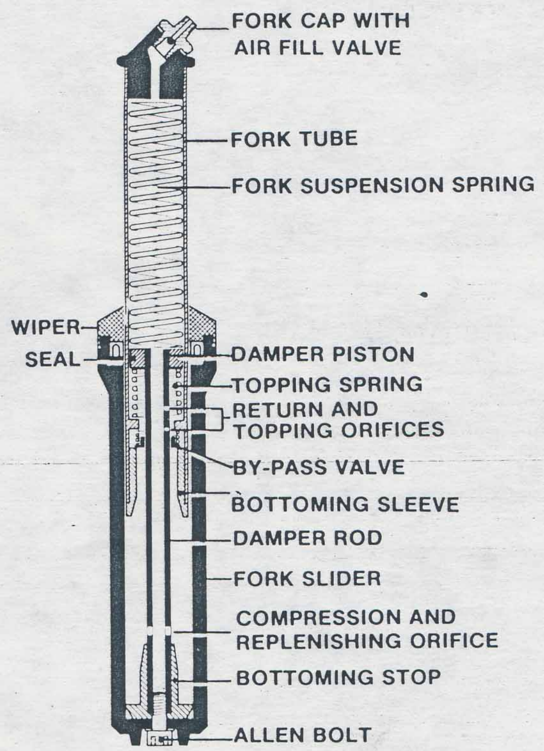
PISTON SPEED	SPEED SQUARED	DAMPING FORCE
2 in./sec.	4 in./sec.	20 lbs.
4 in./sec.	16 in./sec.	80 lbs.
8 in./sec.	64 in./sec.	320 lbs.

It is important to recognize this progression of force so as not to be misled by the feel of a damping unit when stroked by hand. In action the speeds and forces involved are much higher.

Two additional factors influence the amount of damping — the size of the orifice and the viscosity of the hydraulic oil. Obviously the larger the orifice the easier it will be for oil to pass through, causing lower damping rates. If you consider the damping escalation due to the squaring effect, it should be clear that a larger hole, which produces slightly reduced low-speed damping, will substantially reduce the damping at higher speeds. The damping is sensitive to minute changes in orifice size.

Damping is also directly proportional to oil viscosity. The thicker the oil the more resistance it has to being shoved through a small hole.

FIGURE 20 FRONT FORK NOMENCLATURE



chapter ...5

Now that we are armed with a fund of technical tidbits we can look at the oil flow inside the fork leg. **Figure 21** through **24** diagrams in simplified form the four different conditions encountered by a fork while in action. **Figure 21** shows the fork beginning the compression cycle.

To begin we must establish the boundaries of the various chambers the oil **can** occupy. Chamber A below the one-way bypass valve contains the oil that creates the compression damping and supplies the oil for the upper chamber. Chamber B above the one-way valve and below the piston is used for rebound damping. Chamber C is the space above the piston. This chamber is the reservoir and unlike the other two, it is not completely full of oil for it has an airspace above the oil. Its function is to keep the other two full of oil at all times and, hopefully, devoid of any air.

As the fork is compressed the oil in the bottom chamber (A) is squeezed by the slider moving up. Being in-compressible, pressure is generated which opens the one-way bypass valve and the oil is free to migrate up to Chamber B. When the oil arrives in Chamber B it discovers the space available is smaller than Chamber A due to the volume now occupied by the wall thickness of the incoming fork tube. Again being in-compressible the extra oil must go somewhere else. Other than leakages around the various parts the only other path for the oil is to go into the bottom hole in the damper rod and up through the hollow damper rod to the reservoir Chamber C which will accept the leftover oil. As the oil travels through the bottom hole resistance is generated which creates the compression damping. The compression damping is directly proportional to the volume of oil displaced by the volume of the incoming fork tube. We must add that some oil goes to the reservoir through the upper damper rod (or return) holes during the compression cycle. The amount of compression force is calculated by adding the area of all the holes together. Make a mental note here that the oil level in the reservoir is raised by all this additional oil. This will be important when we discuss fork springing.

Figure 22 shows the fork at the end of the compression stroke interacting with the hydraulic bottoming stop. You will notice this condition only occurs when the fork has utilized its full travel. About an inch and a half from the end of the stroke the bottoming stop engages the bottoming sleeve. This gradually closes off the passageway that leads to the one-way valve. The effect is that you get extra compression damping in a manner that provides more and more damping the nearer the fork is to the end of travel. In fact the damping is so high that metal to metal contact is virtually impossible, which is why forks of this type are said to have hydraulic bottoming control.

A side effect of this hydraulic stop is the return stroke has extra damping until the stop and sleeve are disengaged. It is just as difficult for oil to return to Chamber A as it was for it to be squeezed out. To minimize this effect most Japanese forks now provide an additional one-way valve on the bottoming stop to hasten in the refilling of Chamber A.

FIGURE 21

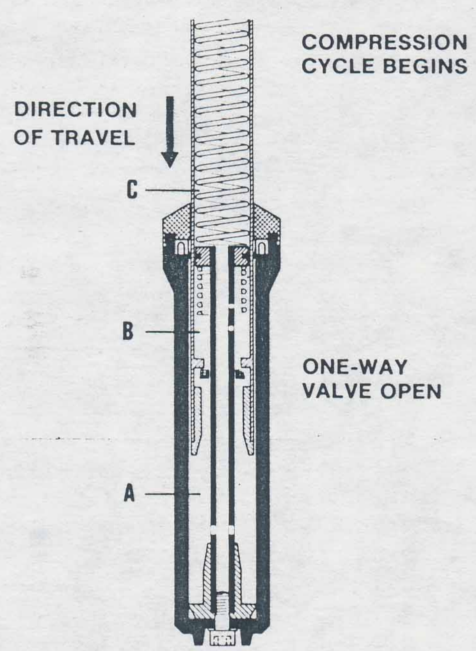
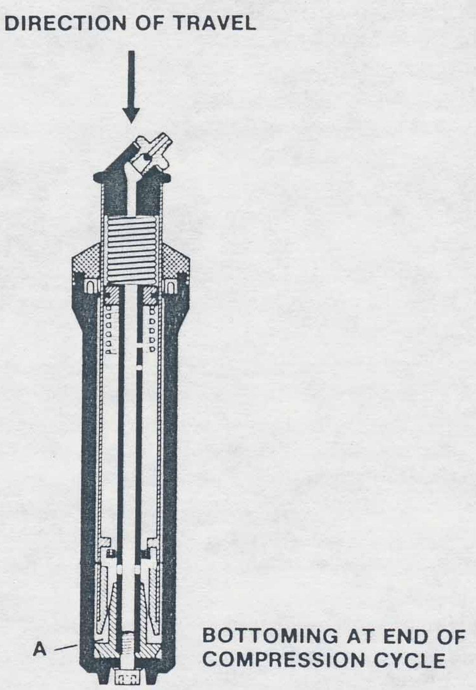


FIGURE 22



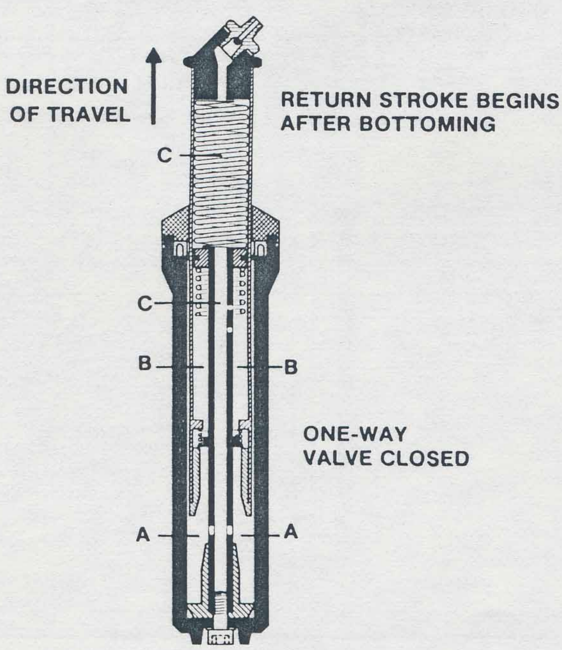
chapter ...5

Figure 23 diagrams the fork on the rebound (or return) stroke just after it has cleared the bottoming stop. The return stroke is initiated by the force of the spring pushing on the top of the damper piston. When this happens the pressure differential across the one-way bypass valve reverses and it closes instantly. This traps the oil in Chamber B. The spring continues to push on the piston, forcing the oil in Chamber B to leave via the upper damper rod holes. The resistance to flow through these holes creates the rebound damping.

We must remember the oil in Chamber B arrived there via Chamber A, and is on its way through the damper rod to the reservoir Chamber C. Additionally the oil that didn't fit in Chamber B went to Chamber C. This leaves Chamber A to draw in fresh oil back through the bottom damper hole to replenish itself. It must draw in all the oil, not just the amount that passed through during the compression stroke.

A problem can arise at this point because of the difficulty in completely refilling Chamber A. The pressure differential across the compression hole is never as much during the return stroke as during the compression stroke because of slower piston speeds and because the pressure in the reservoir is not as high or positive as that in Chamber A. If this problem is severe enough some air will be sucked into Chamber A. The air floats to the top of Chamber A to be pumped into Chamber B as soon as the fork goes into a compression stroke, where it then floats to the top of the Chamber B and is pumped out of the return damping holes on the next return stroke. This sequence causes a momentary loss of damping on both compression and rebound because the air passes through the holes with little resistance.

FIGURE 23



The working speed of the fork compounds this problem because once the oil returns to the reservoir Chamber C there is insufficient time for the air to float to the top of the oil and escape. The air is sucked back down to the compression hole and the cycle is repeated. Continuous repetition of this cycle quickly turns the oil into foam by mixing it with air. This is called aeration.

This phenomenon is aggravated by the use of heavy weight oil, so it is much better to use thin oil. If the damping is inadequate, smaller sized damper rod holes are a better cure than thicker oil.

Another way to combat foaming is to raise the oil level in the reservoir to create greater pressure head. This is a good solution but care must be taken not to excessively reduce the remaining air space in the reservoir Chamber C, which causes too much air spring effect. When this condition occurs the forks become very harsh, and if you inspect the marks left on the fork tube by the wiper boot you will likely discover the forks do not use all their available travel. Drain a little oil if you suspect this condition. If you immediately detect a loss of damping you will have to raise the oil level back up and consider increasing the air space by adding some kind of canister or chamber to the top of the forks.

Another alternative is to enlarge the compression hole to make it easier for the oil to return. The penalty here is the compression damping will be reduced which may or may not be an improvement. Our experience has been that most forks work better with a slight increase in the size of the compression hole. If you embark on a program of drilling out the compression holes, we suggest .010 inch increments. If you get the hole too big there won't be enough pressure in Chamber A to open the one-way valve and all the oil will go out of the compression hole — the path of least resistance. Valve opening pressures must be balanced with damping holes.

The best approach to the replenishing problem is to reroute the oil on its return from Chamber B by plugging up the rebound damper holes and relocating similar sized holes in the one-way valve or valve body. This creates the same rebound damping force but routes the oil directly back to Chamber A reducing the demands on the bottom compression hole for replenishment. On a shock-testing dynamometer this modification shows that piston speed may be approximately doubled before aeration occurs. To the rider this means faster speeds before fork damping fade sets in.

Figure 24 shows the fork in a fully extended (or topped-out) condition. Generally a combination ~~of~~ springs and hydraulic control is employed to help slow the fork down as it nears the end of travel. The purpose again is to avoid metal-to-metal contact and subsequent high shock loads to fork parts and the rider's arm. If you study **Figure 24** notice the top damping holes slide right through the one-way valve as the fork nears the top. This reduces the amount of remaining holes in Chamber B which raises the return damping in a progression of steps, and helps slow the fork before the topping spring becomes the final brake. Sometimes the diameter of the damper rod tapers out near the top to act as a brake. The idea is to reduce the clearance between the damper tube and the one-way bypass valve and progressively close that additional leakage. In fact some forks depend on that leakage to control the entire return damping, and S & W Fork Kits make use of this principle. When this is the case you will find few or no top holes

chapter ...5

and the damper rods will be contoured to provide different amounts of clearance for varying damping characteristics depending on the position of the fork. This system provides a lot of scope for tuning-in your suspension.

Now that you understand what goes on inside, you can study your own fork problems and make some adjustments for improvement.

The most common damping problem is a phenomenon called "pumping down" which means the fork doesn't return quickly enough to recover for the next bump. Specifically the return damping is overpowering the spring force. This can happen in various degrees; it can be so minor that it can only be detected as harshness near the top of the stroke, or it can cause the forks to stay compressed much of the time. If it is too severe the steering geometry will be affected, the ride will be even harsher, and the suspension won't have enough remaining travel to deal with more bumps. This is most pronounced in a series of "whoops."

This condition can be attacked in several ways. If you feel your forks are also too soft you can increase the spring-rate to help return the forks quicker. It is easy to be fooled here — sometimes forks that are pumped down seem too soft because they work (or stay) way down within the travel. You will have to use your own judgment. Another approach is to use thinner oil. If that doesn't work, drill out the upper damper holes about .005 inches each time or increase the clearance between the damper tube and the one-way bypass valve.

The opposite problem is not enough damping. This is detected by the fork bouncing back too quickly and the steering attitude never stabilizing while cornering. The first thing to look for is not enough oil. If that doesn't work try the opposite of the cures for "pumping down."

The above two problems involve return damping, but another set of problems surrounds the compression damping and spring combination.

If you conclude that your forks are just too stiff and you have eliminated the possibility of pumping down, check a few of these items. First be sure you don't have too much oil. If that doesn't help try a softer spring combination. Next drill out the compression damping hole. If the forks continue to be insensitive to your efforts it is possible you have extra damping due to an unwanted restriction. The most likely culprit is restriction past the one-way bypass valve. The cure is to machine more space around the valve by drilling extra holes in the body or cutaways in the valve itself, or use thinner valve body parts. If you suspect this condition try these modifications before any of the other steps. It can't hurt the damping.

Obviously if your forks are too soft and bottom-out all the time, the opposite cures can be applied. Sometimes as you increase compression damping you will discover that you have been overcompensating for this deficiency with extra spring rate. You may get a better combination with a softer spring (and/or less air pressure in the case of "air assisted forks").

Another area that may be affecting your damping is the sliding friction and drag associated with the seals, wiper boot and the bearing surface of the slider against the fork tube. If you have any problems in this area it will manifest itself as extra low speed damping on both compression and rebound. A good clue would be minor pumping down. First check alignment of all the parts. Check the fork tubes and sliders and the axle for straightness — a bent axle will cause the slider to tip out creating friction. Check the width of the wheel assembly to make sure the sliders are not pinched in or out when the axle clamp is tightened. Finally check the fork crowns to make sure the fork tubes are parallel from the front and the side.

Mechanically that is about all you can do except choose forks that have better bearing arrangements near the top, such as those with bronze or Teflon bushings. One last thing to watch out for is binding due to the forks being over extended for the amount of overlap of the slider and fork tube.

The seals are the next focus of attention. The main function of the seal is to keep the oil from leaking out, but if it does its job too well excessive drag results. Drag comes from inordinate seal and wiper tension or from wiping the fork tube dry. Your selection of seals is very important. The material and tension and the number of lips are all factors. There is no complete answer to seal selection but the bare minimum of sealing seems to be the best approach. Even slight leakage is acceptable because it helps smooth action.

We have seen well-used forks that have become highly polished and thereby taken on added seal drag. The explanation is they can then be easily wiped too clean and dry. The best forks are cross-hatch honed on the outside of the fork tube to retain a little oil. You might consider roughing up your tubes slightly with wet emery paper if you suspect this problem.

The wiper boot (or dust cover) is generally way too effective and if you want to reduce its tension this can be accomplished by carefully grinding away the top so the hole gets a little bigger. If everything is still too dry consider a ring of oil-soaked felt or foam air filter material sandwiched under the wiper boot.

FIGURE 24