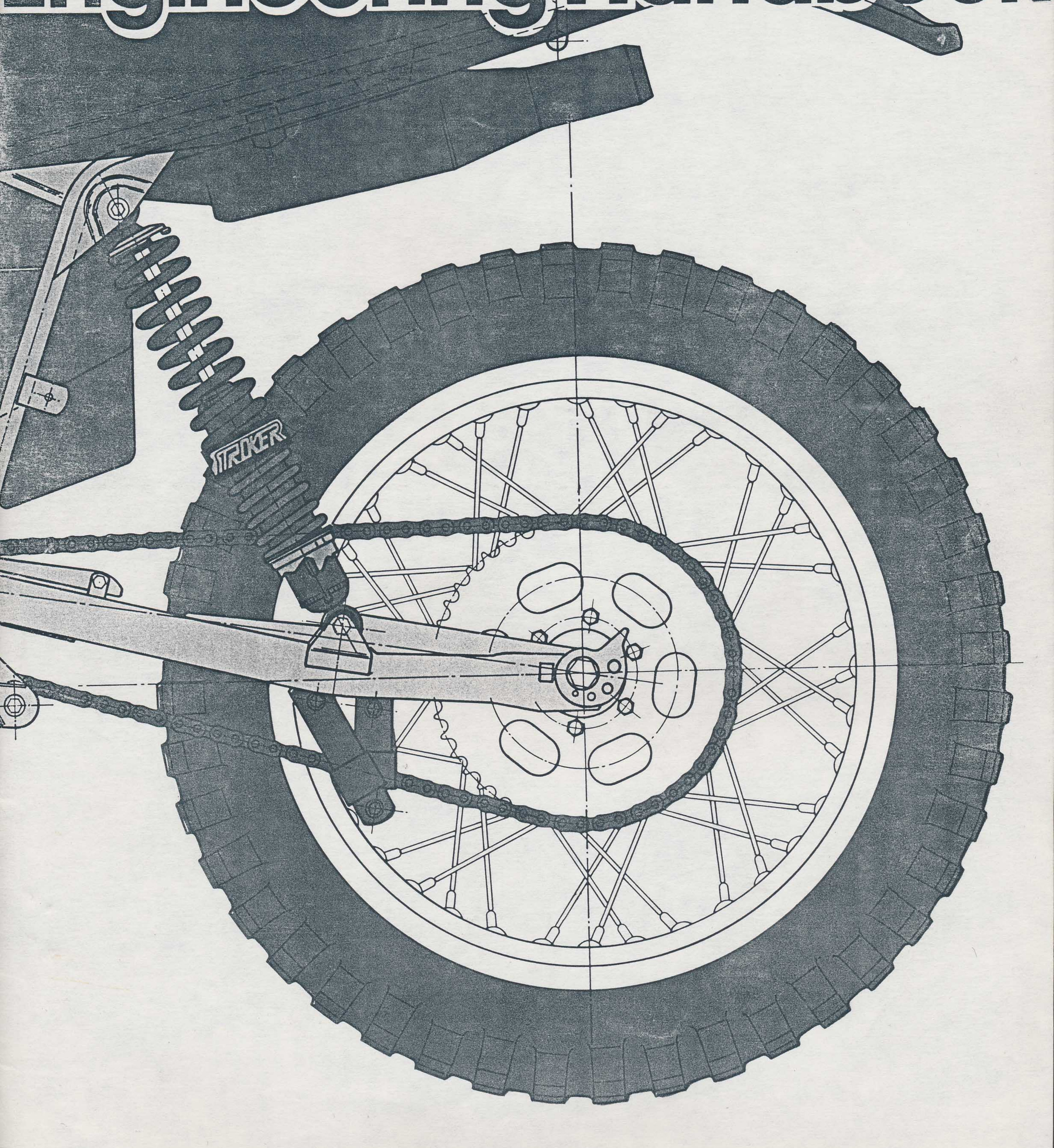
S&W

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.

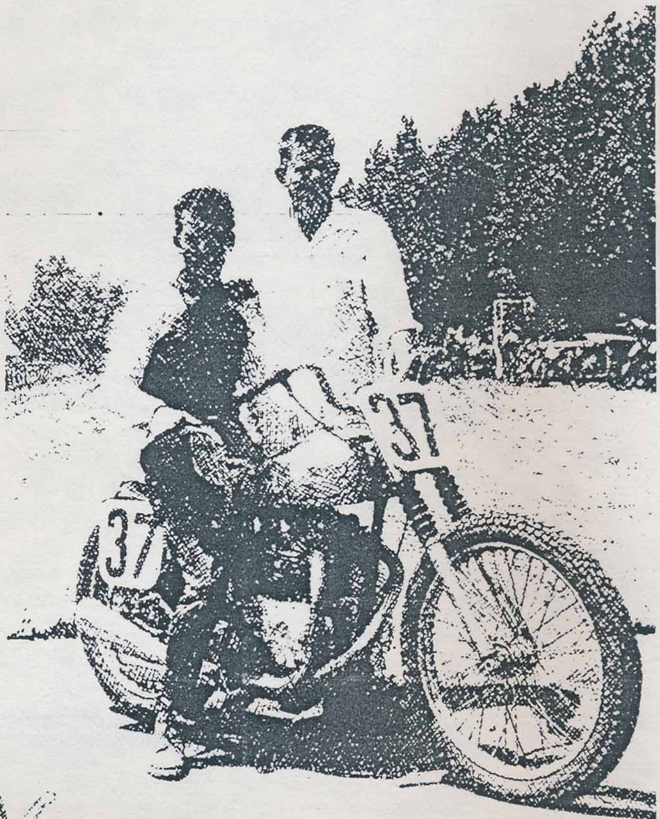
In 1974 Mr. Tim Witham of S&W Engineered Products asked Bruce to look at the booming sport of motorcycling as a practical avenue to apply the sophisticated technological fallout from Indianapolis and high performance automobiles. The next three years Bruce worked at S&W as Head Designer for Development 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.

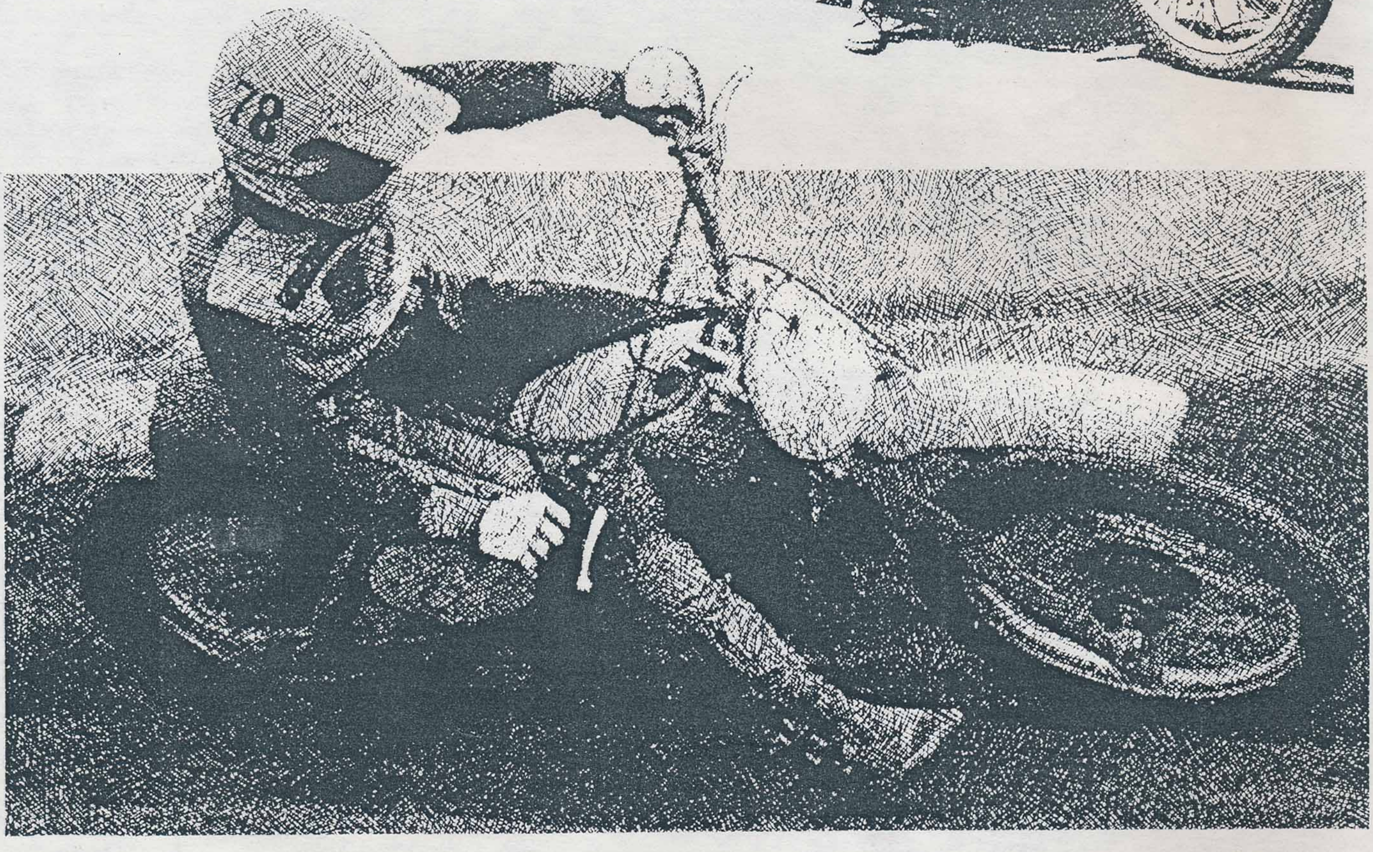
Introduction

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.

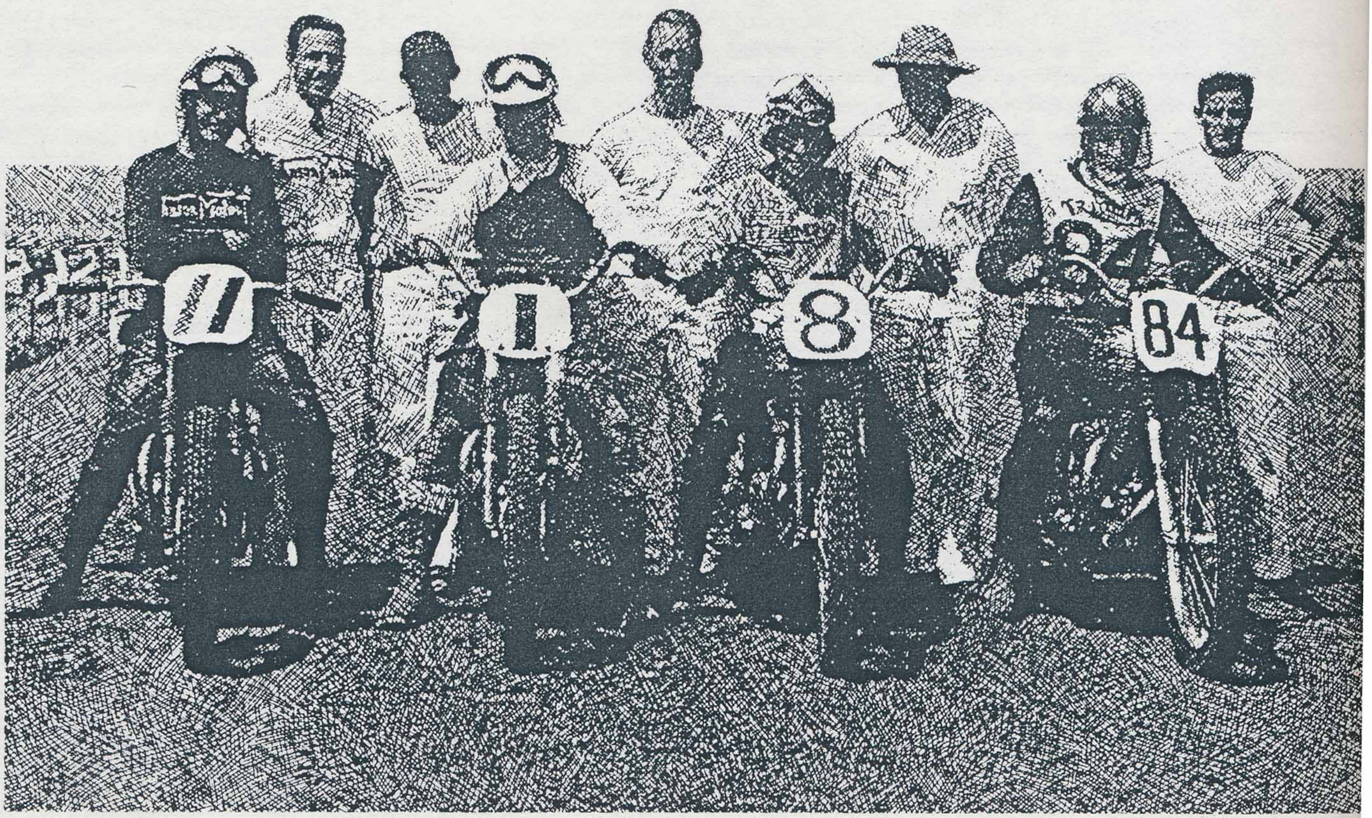
The first chapter will be very general in nature and will present some basic suspension theory and describe the various elements and components that comprise a successful suspension system. Beyond the basics, we will present specific discussions of the various elements that contribute to the feel and 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.

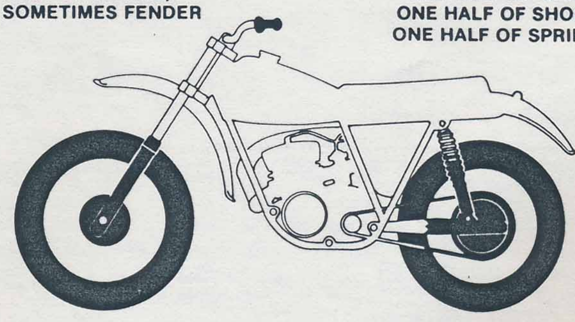
The designer's job is to find the happy medium for the specific application. The key word here is **specific.** As motorcycles become more sophisticated and specialized, a part designed for one type will not be suitable for another. Also enter in the particular bias of each designer (they all prefer 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.

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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

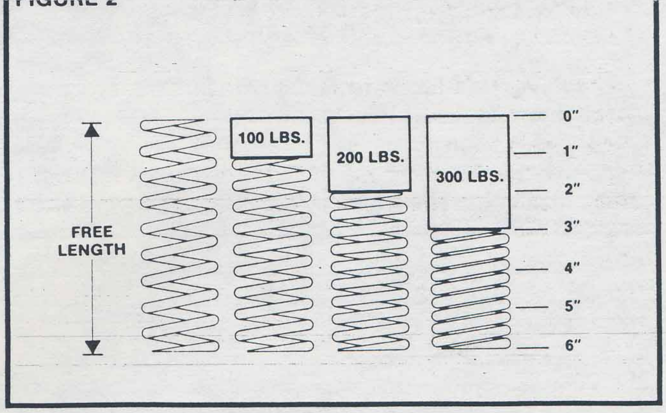
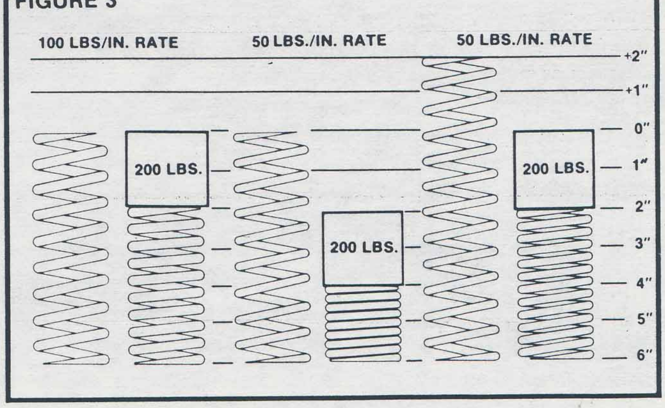


FIGURE 3



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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.

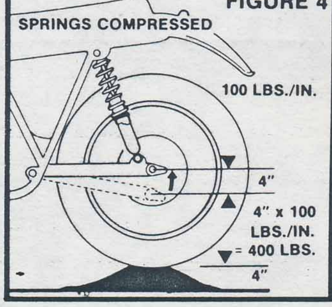


FIGURE 4

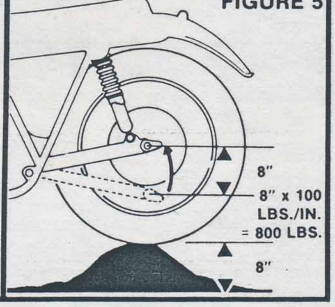


FIGURE 5

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.

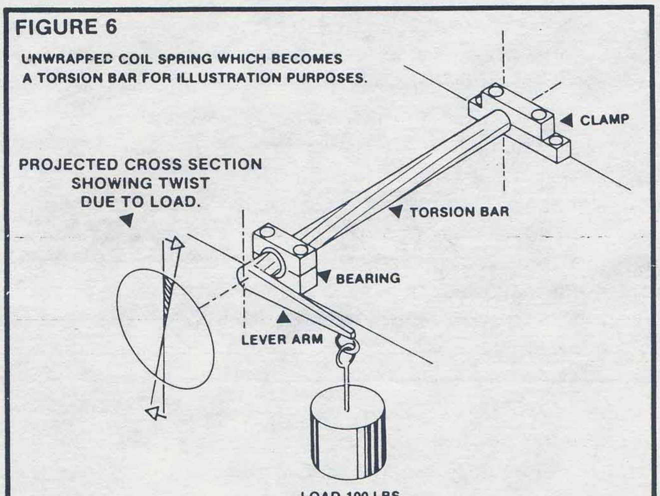
Many times the deflection caused by the load is referred to as **preload. Mechanical Preload** is the amount the spring is compressed when it is installed in the suspension. **Static Preload** is the mechanical preload plus the additional amount the spring compresses when it is supporting the chassis 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



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-

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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.**

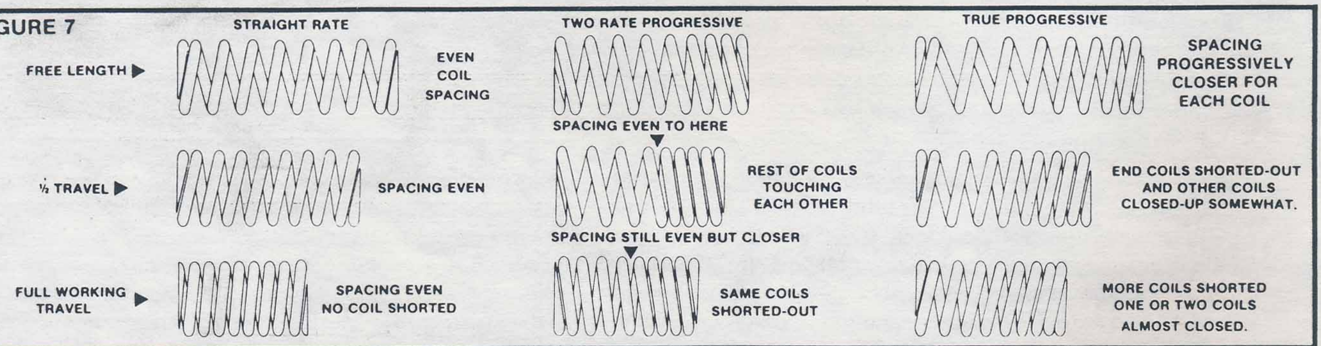
Overstress is a condition where you are making too little wire material to do too much work. The molecules actually slip in relation to each other. They don't return to their original position when the load is removed. A sign of this condition is when a spring takes a permanent set or sags-out by not 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



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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.

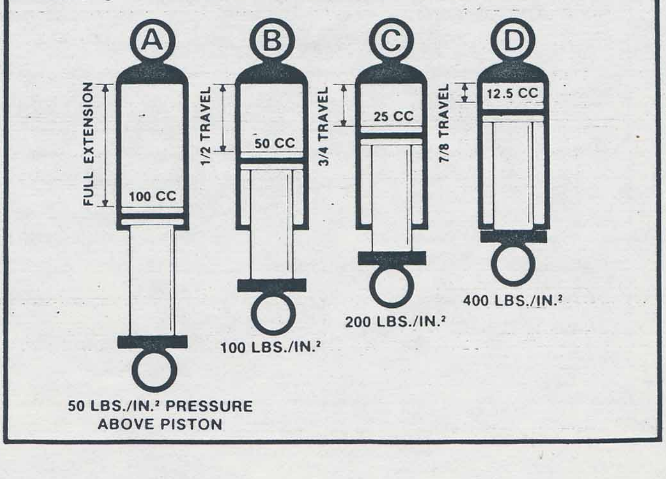
First we have the problem of making sure this air-spring device doesn't have any leaks. This requires careful design and that all the parts involved are kept in perfect condition. In some designs this also requires an investment in tanks, regulators, and gauges in order to monitor the pressure.

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



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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

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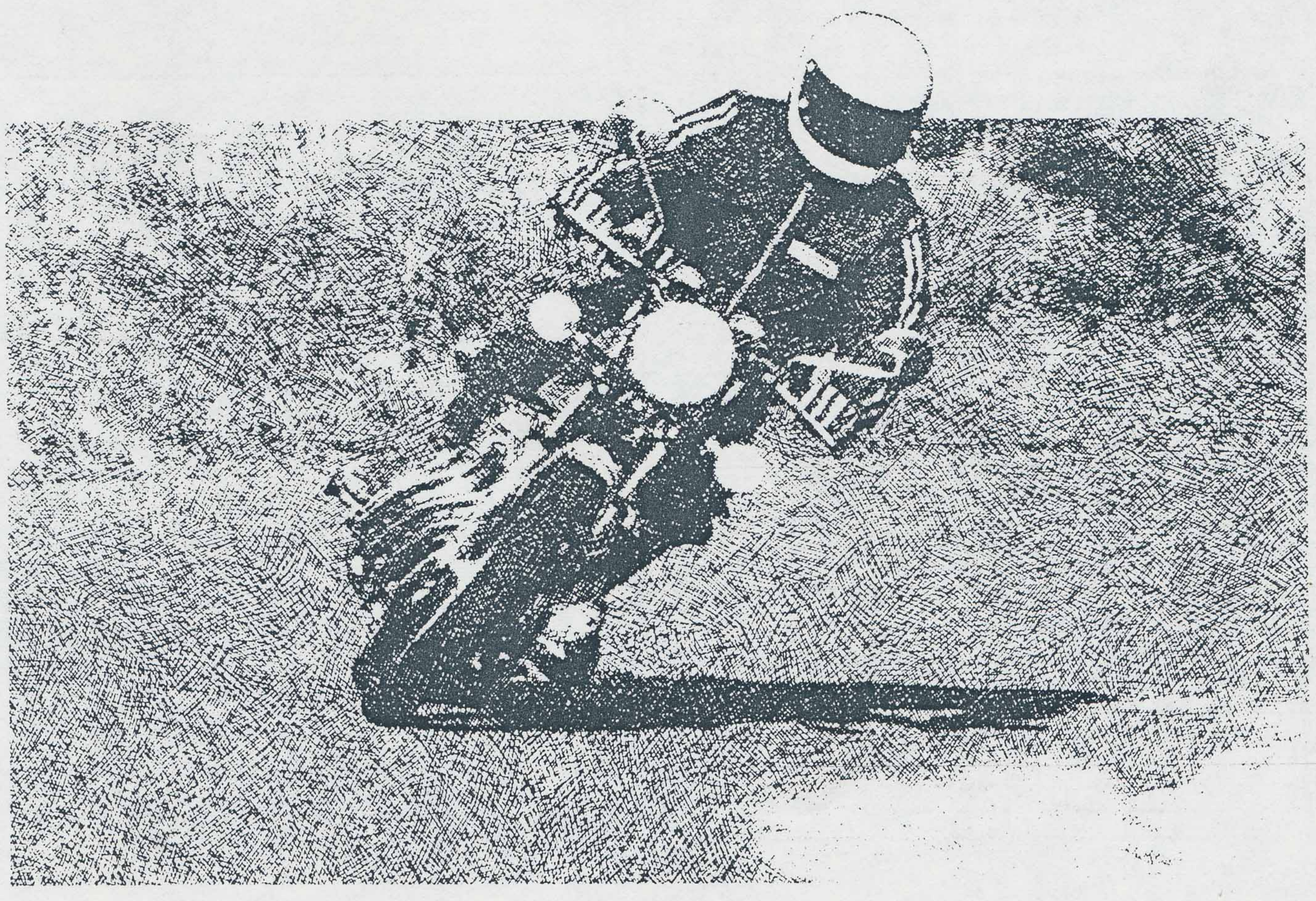
Let's see how we might apply all this theory directly to your motorcycle, and particularly to your front forks. Suppose while test riding you conclude the forks are too stiff and don't travel the full amount, so you lower the starting pressure. Suppose this change doesn't help the feel very much and 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.

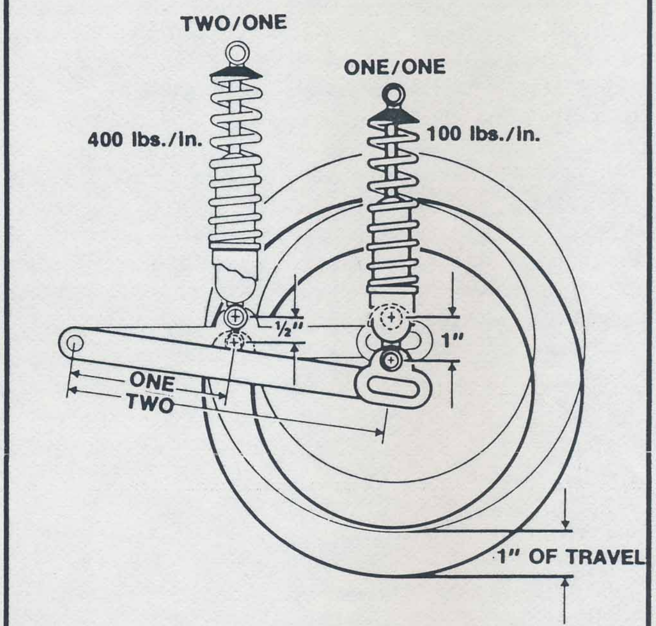
By now it has become pretty well accepted that longer wheel travel will get you over rough terrain more quickly. Historically the first attempts at getting increased wheel travel were accomplished by moving the shock mounting position farther up the swing arm away from the rear axle. That wasn't exactly 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.

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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.** **a = 7 in.**
B = 13.5 in. **b = 9.5 in.**

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

Leverage ratio = $\dfrac{\text{wheel travel}}{\text{shock travel}} = \dfrac{8}{4} = 2$

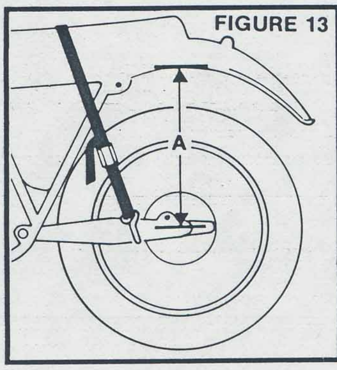

FIGURE 13

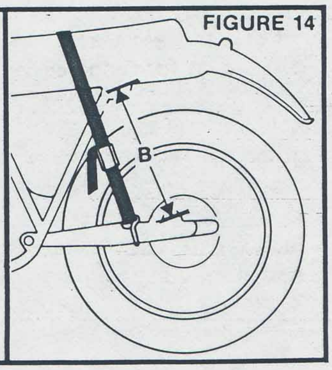

FIGURE 14

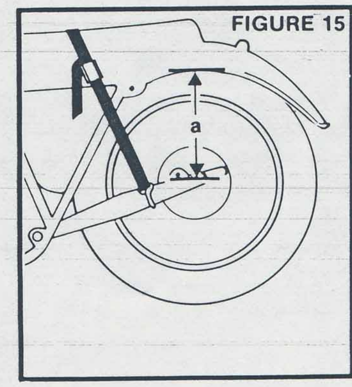

FIGURE 15

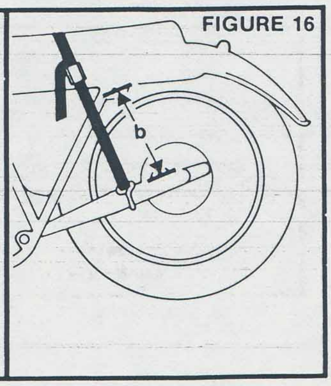

FIGURE 16

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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:

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

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

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

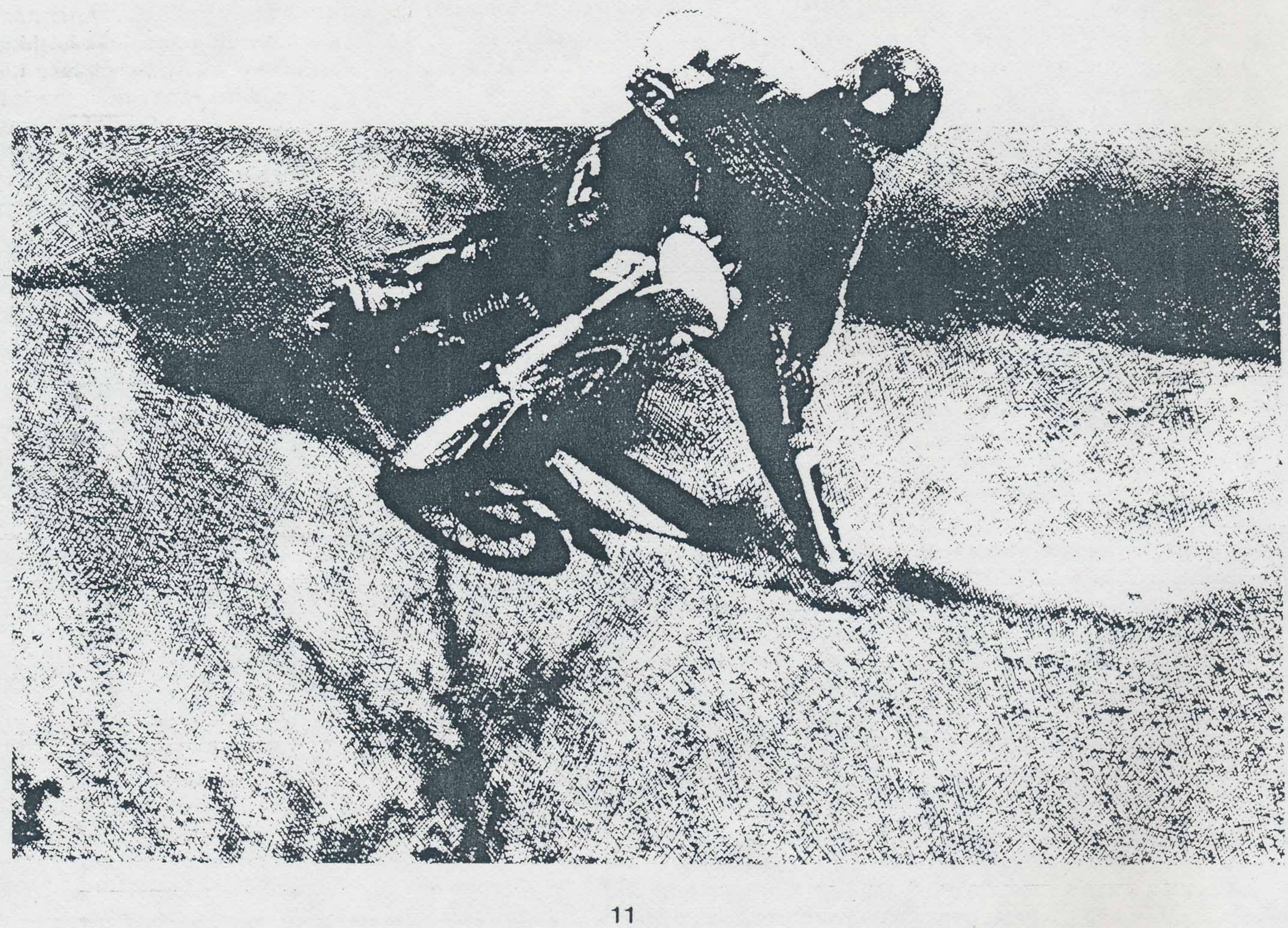
$$\frac{4 \text{ X } 65 \text{ lb./in.}}{2} = \frac{260 \text{ lb./in.}}{2} = 130 \text{ lb./in.}$$

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:

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

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.

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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.

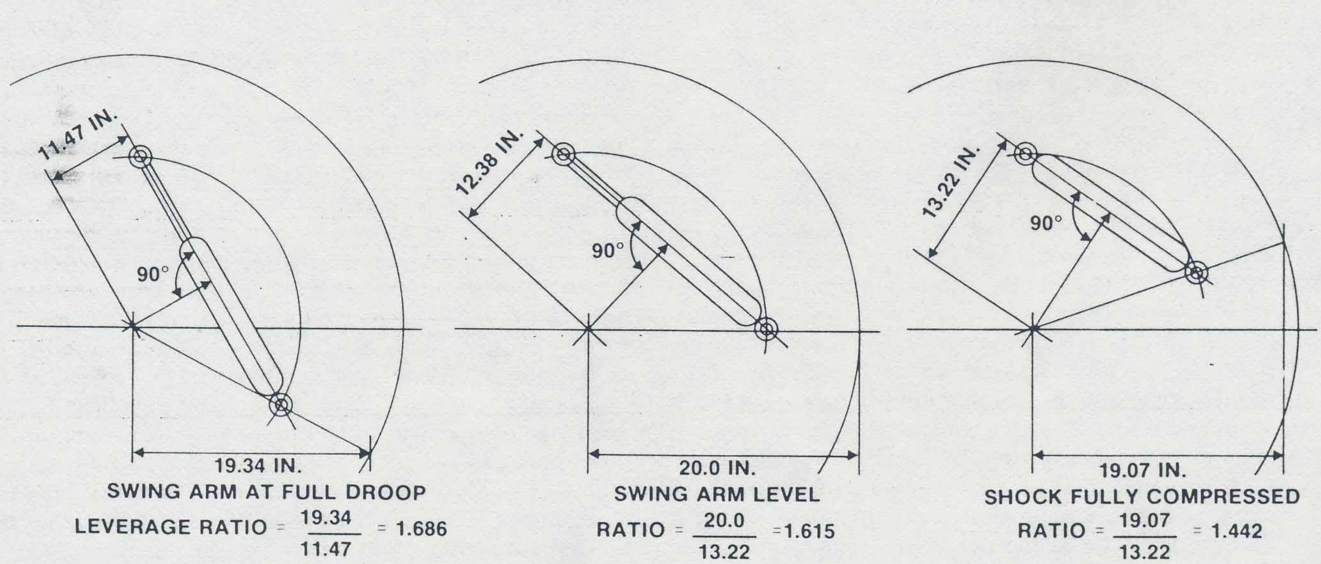
The question of anti-squat is covered in a future chapter, but for the moment be aware of how excessive anti-squat is determined. In **figure 18** you will see wheel travel divided into two parts: that above the horizontal swing-arm position (bump travel), and that below horizontal (droop travel). It 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

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

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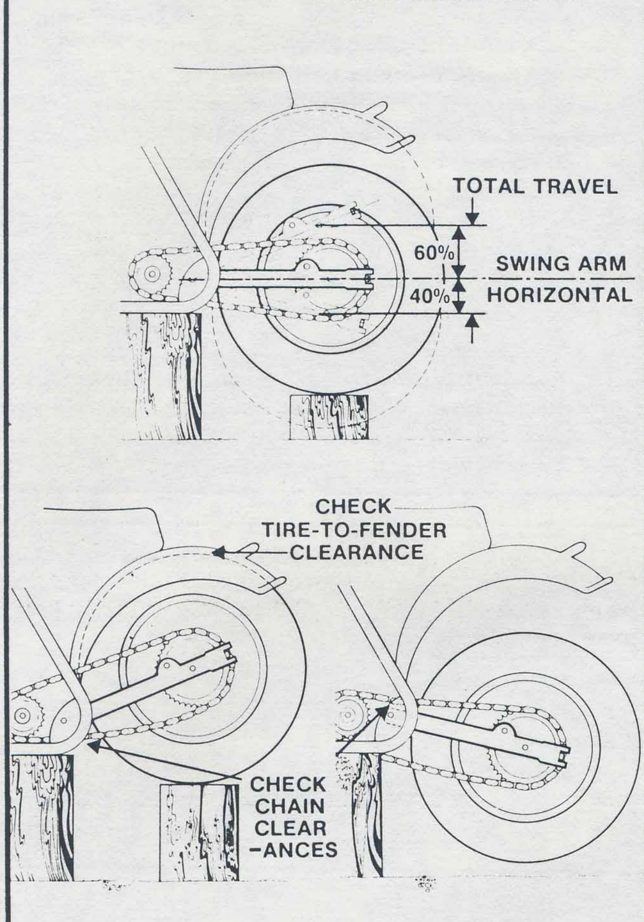
FIGURE 18 LIMITS TO REAR WHEEL TRAVEL



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.

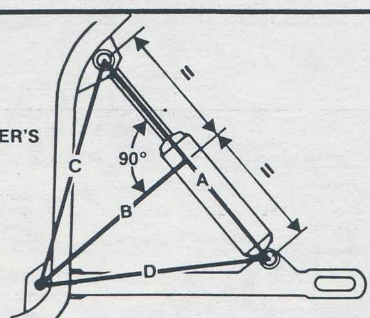
B. From that halfway point, project a line 90° from line A. The calculated length of your shock lever arm determines the length of this second line, line B. Mark line B with a short dash to indicate this length.

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

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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

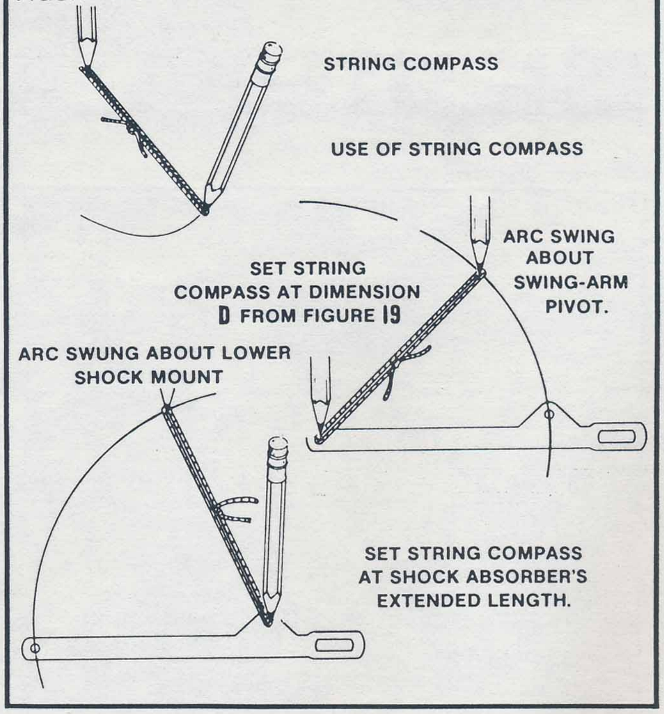
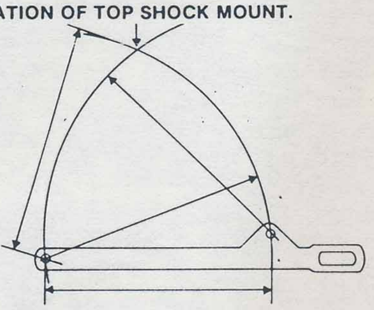
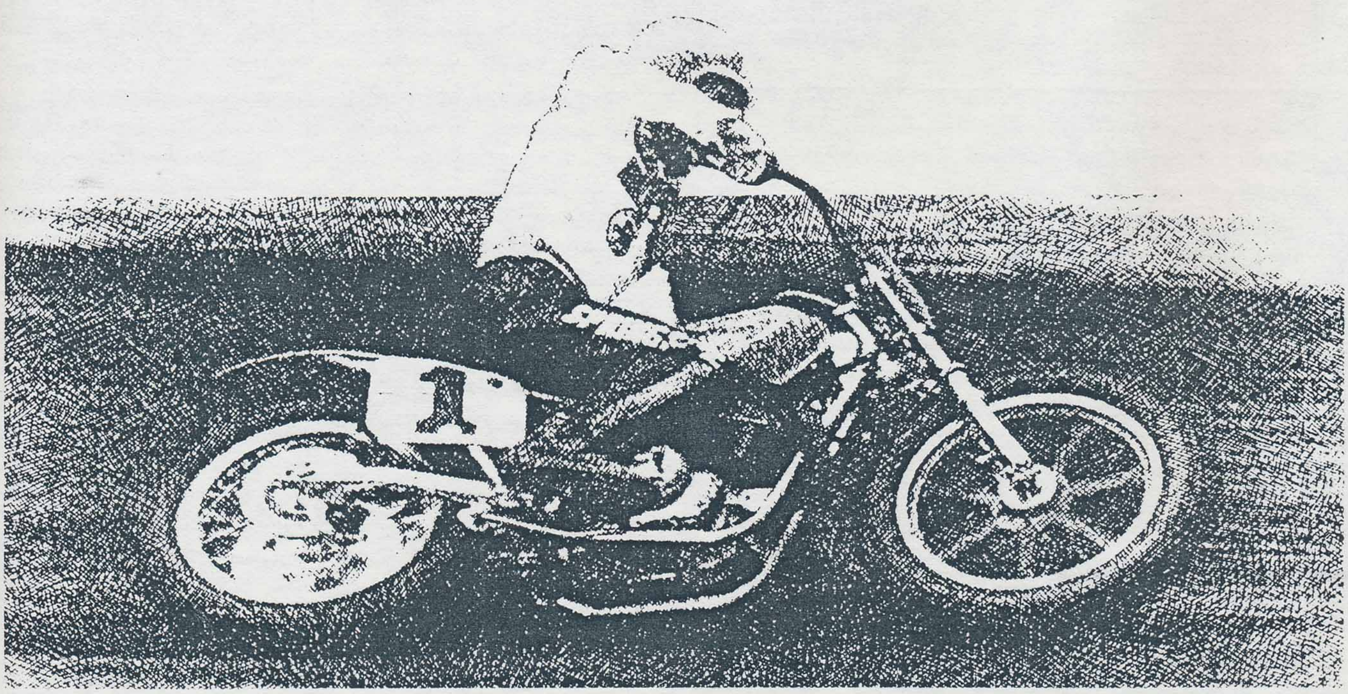


FIGURE 19 B





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.

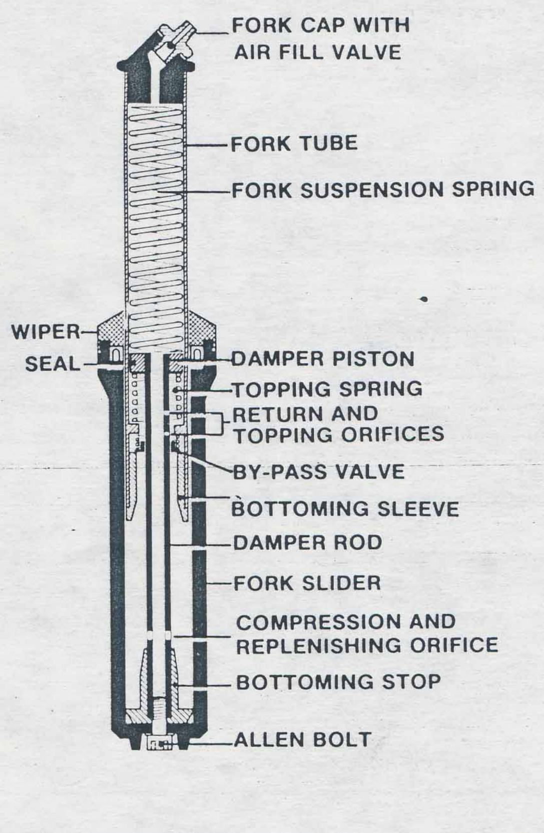
PISTON SPEED	SPEED SQUARED	DAMPING FORCE
2 in./sec.	4 in./sec.	20 lbs.
4 in./sec.	16 in./sec.	80 lbs.
8 in./sec.	64 in./sec.	320 lbs.

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



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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

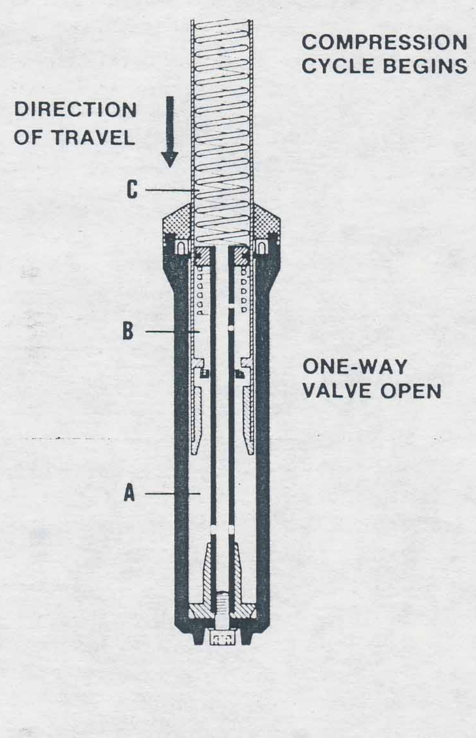

FIGURE 22

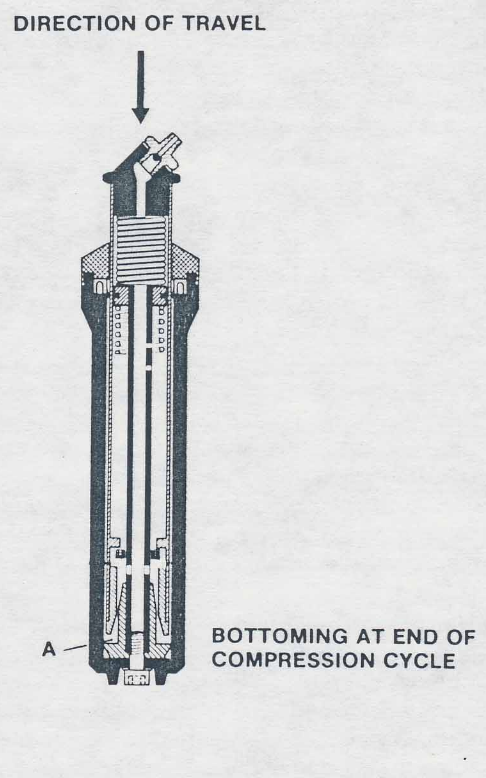

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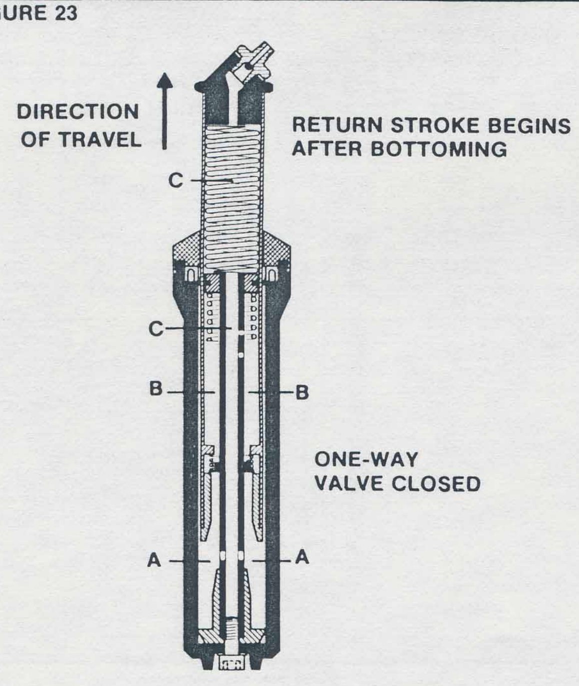
Figure 23 diagrams the fork on the rebound (or return) stroke just after it has cleared the bottoming stop. The return stroke is initiated by the force of the spring pushing on the top of the damper piston. When this happens the pressure differential across the one-way bypass valve reverses and it closes 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

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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.

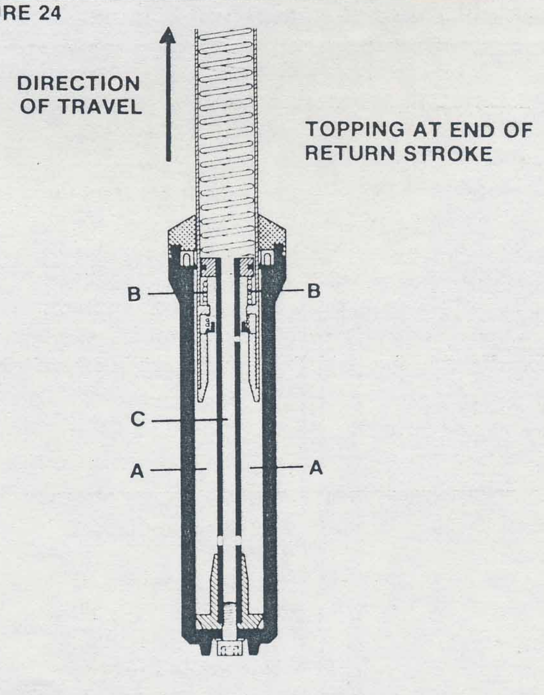
Mechanically that is about all you can do except choose forks that have better bearing arrangements near the top, such as those with bronze or Teflon bushings. One last thing to watch out for is binding due to the forks being over extended for the amount of overlap of the slider and fork tube.

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



Chapter ...6 Shock Absorbers

In this chapter we are going to take an in-depth look at the inner workings of shock absorbers. We will look at the differences of the basic shock types. We will follow the hydraulic oil on its many journeys inside the shock absorber. We will see how to arrange the shock valvings to give various types of damping forces. We will explain those funny little oval Shock Dyno Charts you may have seen in the road test sections of several motorcycle publications.

Before we go inside the shock, let's discuss why we need damping in our suspension system and how a shock absorber supplies damping.

In the earlier chapters of this handbook we explained the need for springs in our suspension system and how a myriad of different spring forces occur while the suspension is operating. We also learned a spring will give back the same energy put into it. If we don't want that energy to have an effect on the chassis, we must devise a method of consuming that energy. The damper (or shock) takes the energy stored in the spring and dissipates it gradually by pumping oil through tiny orifices and valves. These orifices create resistance to the flow of oil and that resistance becomes the damping force. The resistance through the valves converts the spring energy into heat and eventually the heat is distributed out of the shock into the surrounding air.

The key to a successful shock absorber is to match its resistance or control force to the force generated by the spring. **Be aware that different spring rates will require different shock absorbers.** A shock suitable for a heavy spring will overpower a light spring and suspension movement will be limited. Conversely a shock suitable for a light spring will not be able to handle a heavy spring and suspension movement will be excessive.

In actual practice damping also occurs as the spring is being compressed, and not just after it. Most shocks assist the spring by resisting movement during the compression part of the bump sequence. This has a two-fold advantage. First it keeps the suspension from overrunning or floating higher than the road obstacle, minimizing suspension travel. Secondly, compression damping has the effect of adding to the spring rate so a softer spring can be used. This means the shock will have less spring force to deal with on the return stroke.

There are many philosophies among shock manufacturers as to the proper amount and proportions of damping. Some believe no compression damping combined with stiff springs and a lot of rebound damping is the answer. (Compression or bump damping is the force generated while the shock is going together or being compressed. Rebound or return damping is the force generated while the shock is being pulled open or returning to its original state.) Others favor a substantial amount of compression damping with very soft springs and very little return damping.

After exhaustive testing we have come to prefer a damping ratio somewhere in between those two extremes. Both of the above philosophies work excellently in special types of conditions, but both have problems performing the full range of suspension requirements. The "no compression" concept has the problem of stiff suspension even at very low speeds. It has the same suspension rate at all speeds. This scheme is also prone to "pumping down" of the suspension, and the stiff springs put a heavy burden on the shock absorber during the rebound stroke. The "high compression" concept has limitations of an entirely different nature. Since damping force generally varies with the speed of the damper movement, if you use a lot of compression force for normal suspension speeds, you will likely get an excessive amount of damping force at high suspension speeds. This can cause suspension "lock-up" which can lead to a trip over the handlebars. Another problem associated with the combination is not enough spring rate to deal with situations where you encounter sustained "G" forces. These are found on bankings and long corners and at the bottom of large undulating "whoops" or "sand rollers." The effect on the motorcycle is one of slowly settling into the suspension until it either bottoms the suspension or grounds out on the underside. Needless to say, there is also no suspension left to deal with any further bumps.

The amount and proportions of damping control unfortunately don't provide a complete picture of a shock's damping characteristics. We have learned that the amount of force generated by a spring is proportional to how far it is deflected. A shock absorber is quite another animal. Its forces are related to how fast it is moved. In fact **as you pass oil through a fixed orifice its resistance will increase as the square of the velocity.** Luckily a "fixed orifice" describes a primary or fundamental shock absorber. Today's shocks have an elaborate system of valves that allow us to deviate from this "force squaring with speed" phenomena. Generally the shock internals are arranged to provide individual adjustments for each of the three speed ranges or stages (low, medium, and high speed). Additionally compression and rebound will each have their own set of valve stages bringing the total of stages to six. Some shocks have an even finer adjustment capability with more than six stages. S & W shocks are capable of eight stages in adjustment.

If you were to measure damping forces at various piston speeds with a Shock Dyno (more about Dynos later) and then plot those forces on a chart, you will see a **damping curve** that reveals how the shock reacts to changes in speed. **Figure 25** illustrates several variations of damping curves. The curves represent four distinct valving options. **Figure 26** illustrates those four different valving configurations. Valve type **A** correlates with damping curve **A,** Curve **B** correlates with **B** etc., etc. For simplicity we will illustrate the valve stages at the piston during the rebound stroke. The compression valving generally occurs at the base-valve and the manner in which the parts are arranged is very similar to rebound, but in miniature.

Curve **A** and valving **A** are the fundamental "fixed orifice" type. Notice the only parts creating resistance to oil flow are the holes passing through the piston. The size of these holes can be varied to change the resistance but in all cases there is little resistance until substantial piston speed is

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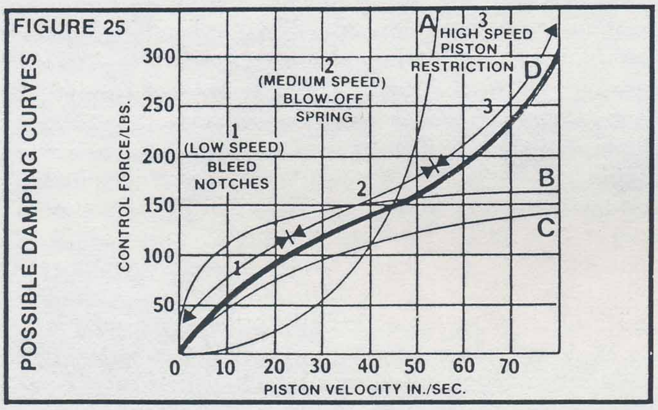
achieved. The curve demonstrates how the force builds up slowly but quickly escalates at the higher piston speeds. If you stroke this shock by hand it will feel as if there is no damping at all but in actual fact is has considerable damping at the speeds it will see when in use.

Suppose you decide valving **A** has not enough low speed damping and too much high speed damping. Valving **B** will produce damping exactly the opposite of valving **A.** In this instance we have arranged a blow-off valve to cover over the piston holes and backed it up with a spring. Now the oil cannot flow until the pressure is high enough to push open the blow-off valve. The point at which the blow-off opens is regulated by the load of the spring behind it. Once the valve is open the damping is again controlled by the size of the orifice through the piston. In valving **B** we have greatly enlarged the orifice to eliminate the excessive high speed damping. If you stroke this shock by hand it will be very difficult to move it, even though it really doesn't have as much damping as valving **A.**

Be careful about judging damping characteristics by how a shock feels when stroked by hand. The speed achieved by hand is lower than any speed the shock sees in service.

FIGURE 25



Now suppose valving **B** is too extreme and you would like damping somewhere in between valving **A** and **B.** Valving **C** is the same as valving **B** except we have added a **bleed notch** to the piston so some oil can flow before the blow-off valve is opened. Damping curve **C** shows that this change has brought the low speed damping right in between the first top options, but the damping does not rise up even as much as valving **B.**

In order to get the damping curve to continue higher than blow-off curve **B** we must arrange the valves like valving **D.** In this case the valve is exactly like valving **C** except we will use a stiffer spring to raise the blow-off pressure and reduce the size of the piston orifice to cause some further restriction at very high speed.

Valving **D** has all the necessary ingredients to gain complete control over the damping curve. Most shock manufacturers have a valve system very similar to that shown in valving **D,** sometimes the parts are arranged slightly differently like flexible spring disc shims in place of coil springs. Sometimes the entire system is a stack of spring disc shims like those found in the "De Carbon system." Whatever the system, there should be a provision to adjust each speed range independently. A typical shock system will have about ten different options for each of the adjustments, whether it is the bleeds, the blow-off valves, the spring discs, the blow-off spring, or the piston or base-pin restriction.

With that many options it is clear the damping curve can be tailored for just about any set of conditions. The pivotal difference between a good and bad shock is the amount of time and money and engineering each manufacturer has invested in learning the optimum valving combination for each application. S & W is proud of the fact that we have devoted time, money and talent to achieving the optimum application for today's motorcycles.

Now let's look at the paths the hydraulic oil takes during the compression and rebound strokes. If you read Chapter 5 of this handbook, the following text will sound familiar as the oil flow inside a shock is similar to the flow inside a front fork.

FIGURE 26

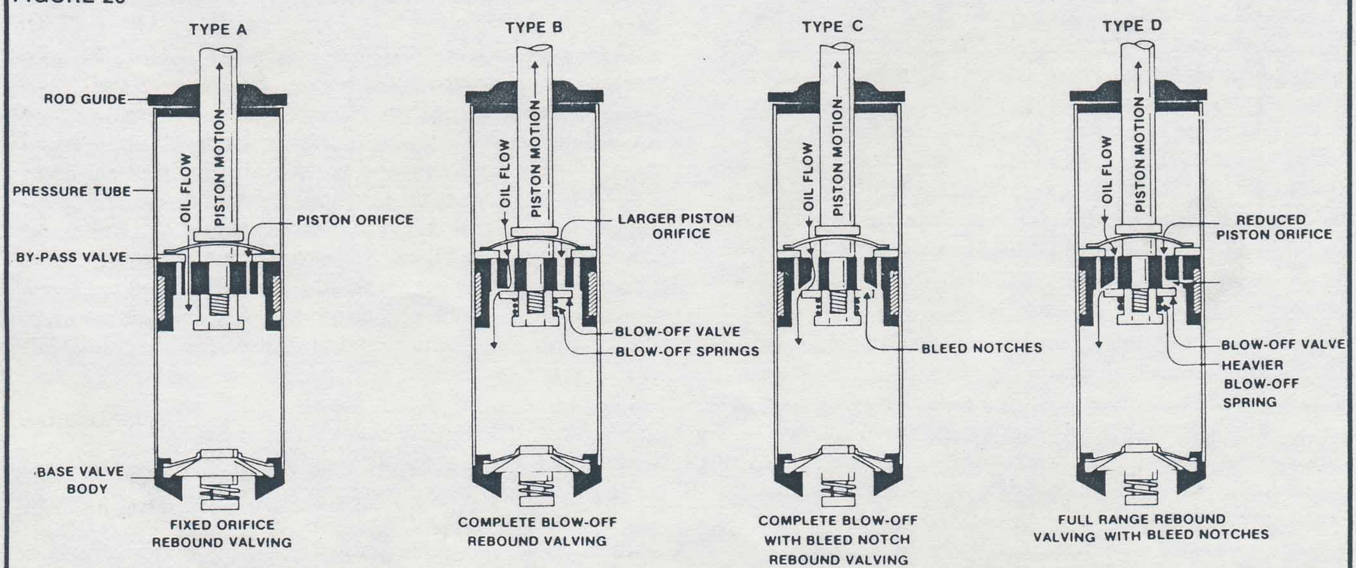


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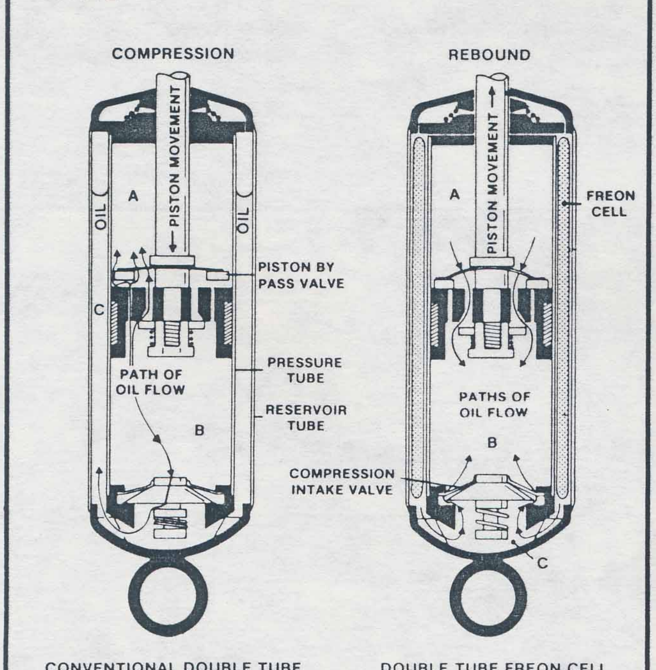
Figure 27 illustrates a conventional double tube type of shock in both compression and rebound cycles.

Compression: During the compression cycle the rod and piston move down in the pressure tube which results in a small pressure drop in the upper chamber (Chamber**A**). Simultaneously the volume of Chamber **B** is reduced causing high fluid pressure. The fluid, following the path of least resistance to correct the pressure imbalance, flows up through the piston's outer passages, unseating the piston by-pass valve, filling Chamber **A**. However, all the fluid originally in Chamber **B** cannot pass into Chamber **A** because the piston rod now displaces fluid. That volume of fluid displaced by the piston rod is forced down through the center of the compression valve and out into the reservoir (Chamber **C**). Compression control is the amount of force necessary to transfer the fluid from Chamber **B** to Chamber **A** and down through the three-stage compression valve at a given piston velocity. (Motorcycle shocks do almost all the compression control at the base valve.)

Rebound: As the piston and rod are moved closer to the top of the pressure tube, the volume of Chamber **A** is reduced and thus becomes the high pressure area. To correct the pressure imbalance, fluid flows down through the piston's three-stage rebound valve into Chamber **B.** However, the piston and rod have been withdrawn from Chamber **B,** greatly increasing its volume. Thus the volume of fluid from Chamber **A** is insufficient to fill Chamber **B.** The pressure in Chamber **B** thus falls below that in Chamber **C**, forcing the compression intake valve to unseat. Fluid then flows from Chamber **C** into Chamber **B** keeping the pressure tube full. Rebound control is the amount of force required to pass fluid through the three-stage piston valve system at a given velocity.

FIGURE 27



This system is perfectly adequate for normal road use and for most automobiles but for Motocross, where high piston speeds and a lot of leaping and jumping go on, a condition can occur that causes the pressure tube to not refill itself completely. If you look at Reservoir **C** in **Figure 27** you will notice an air space above the level of the oil. Because of the gyrations a motorcycle can experience it is possible to have the oil in the reservoir traveling up into the air space while the shock is traveling down and demanding more oil at the compression intake valve. (This oil weightlessness phenomena can be caused by high piston reversal speeds alone, and it is essential the motorcycle leave the ground for this to occur.)

If the oil is not there at the precise moment, the intake valve drags in air instead. Now we have a mixture of oil and air in the pressure tube, which, unlike normal hydraulic oil, can be compressed. There will be a loss of damping while this oil-air mixture is pumped through the varied control valves. Normally the problem is only momentary and the shock automatically purges the air out of the pressure tube. However, the demands of Motocross are severe and the problem is chronic. More and more air is mixed with the oil until the fluid becomes a foam. At this point the riders all complain of the shock "going away" or "fading" or "boinging." These are all great graphic terms to describe the symptoms, but "aeration" is the name of the failure.

To combat aeration a few manufacturers have provided a diaphragm in the reservoir that isolates the air space from the oil but still accommodates changes in the oil level due to rod displacement. In **Figure 27** you will notice the right hand shock has this device, known as a **Freon Cell,** submerged in its reservoir. The freon cell is a simple plastic bag that has freon gas trapped inside. When a freon cell is incorporated the reservoir is then completely filled with hydraulic oil and the only provision for oil expansion is to compress the freon inside the cell. Because the freon is inside the plastic bag it can never mix with the oil and the oil can never migrate away from the compression valve because the shock is completely full of oil.

The freon cell looks to be ridiculously simple but in fact it is a highly engineered component and must be applied correctly. First in importance is the volume of trapped freon must be in the correct proportion to the volume of oil displaced by the rod. (About 2½ to 3 times the volume of the rod.) Additionally the cell must be positioned in the reservoir so oil can always flow easily from top to bottom. The cell material must be nylon and the gas must be freon. The reason nylon is chosen is that most plastic materials are slightly porous. This allows the encapsulated gas to leak out. Nylon has the smallest pores of the readily available plastics. Freon is chosen because it has a molecule structure too large to pass through the nylon. Air has much smaller molecules so it can pass through the nylon. Because of this a freon cell will collect air from its surroundings and once the air is mixed with the freon it acquires a molecular structure too large to pass back through the nylon bag. This process works to your advantage inside the shock reservoir.

Figure 28 illustrates a **De Carbon** Type Shock Absorber. It derives its name from that of its inventor, Dr. De Carbon. As you can see, it is quite different from the double tube type shock. The primary differences are that it has no reservoir and it has no compression base valve. The lack of a

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base valve is taken care of by providing both compression and rebound valving on either side of the main piston. The lack of a reservoir is overcome by an ingenious arrangement. At the bottom of the shock Dr. De Carbon has designed a floating piston with high pressure nitrogen gas behind it. As the piston rod enters the shock it displaces some hydraulic oil. This causes the oil to push on the floating piston and move it slightly to compress the nitrogen gas. The floating piston accomplishes exactly the same job as the freon cell in the double tube shock but without the need for a separate reservoir and reservoir tube. It also operates at a much higher pressure than the freon cell which is both good and bad news.

The good news is the high pressure acting on the hydraulic oil virtually eliminates the possibility of aeration. The bad news is the high pressure creates a force that pushes the piston rod out of the shock. This force is proportional to the area of the cross section of the piston rod. To keep this force to an acceptable minimum, De Carbon Shocks generally have small diameter piston rods which sometimes are not strong enough for the rigors of Motocross. Additionally as the shock gets hot the gas pressure increases, which increases the force at the piston rod. Unfortunately, the extra force on the rod comes at a time when the shock damping control is diminishing from heat. This can cause a mis-match of damping to springing, making the suspension feel "boingy."

Another drawback of De Carbon Shocks is the available piston rod stroke is generally less than that for other types. This is because of the space occupied by the floating piston and gas chamber in the bottom of the shock. Sometimes this problem is sidestepped by removing the piston and gas chamber from the main tube and placing them into a separate external housing. This housing is then connected to the main housing by a short hose. This yields excellent piston rod stroke but brings on added expense and complication.

The De Carbon System has very precise valving components which respond instantly to changes of the direction of piston travel. This is one of the real advantages of this system. Another advantage is the aforementioned natural tendency to resist aeration of the oil. Still another advantage is derived from the fact the main piston can be larger in diameter than is possible with a double tube type. This is because the piston runs right against the outer housing. This arrangement also encourages heat to dissipate slightly faster. However, only in serious professional racing are these advantages worth the additional complication and expense.

Figure 29 illustrates the simplest of the basic shock types, the **Emulsion Type.** This shock is very much like the De Carbon Shock except it has no reservoir at all and does not separate the oil from the air. These shocks are designed to operate best when the oil and air (or gas) are mixed together in an emulsion or foam. The advantage to this type of shock are lots of piston rod stroke, few parts to go wrong, and they are very inexpensive to manufacture. The disadvantages center around the requirement that the oil be in an emulsified state before they will damp correctly. This suggests that until the oil is emulsified the damping will be excessive. This also suggests the damping will be inadequate if the oil is over-emulsified. Both of these conditions happen with this type of shock unit.

From our earlier discussion we learned that air in the oil makes damping unpredictable until all the air is pumped out. The pumping out process happens at the first part of the stroke which coincides with the low speed part of the damping curve. This makes it hard to achieve precise low speed damping with an emulsion type of shock absorber. This compromise in the damping curve explains why emulsion shock absorbers work well in the terrain they are designed for, but do not adapt to other riding conditions. They are seldom used as a universal, all-around shock absorber, but they are excellent in places like desert racing where the conditions are narrowed down to go straight and fast.

We have already stated it is impossible to tell very much about shock damping by pulling them in and out by hand. In order to move the shock fast enough to make all the valving work we need a machine called a shock Dynamometer. This machine cycles the shock up and down at numerous speeds and measures electronically the damping forces. These forces are then displayed on an oscilloscope so the operator can visually see the action of that particular shock.

FIGURE 28

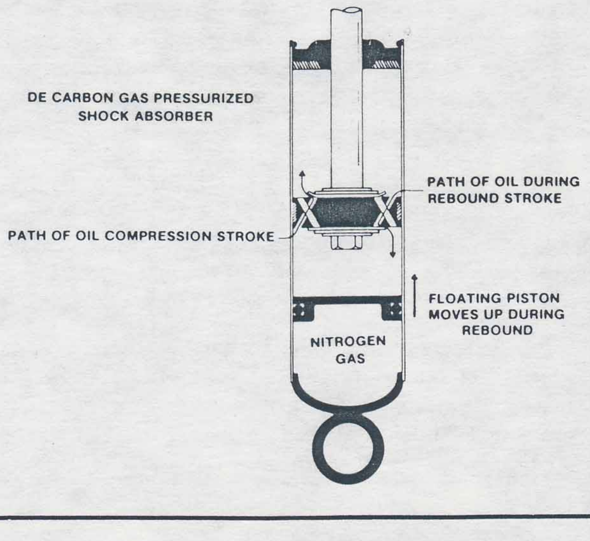

FIGURE 29

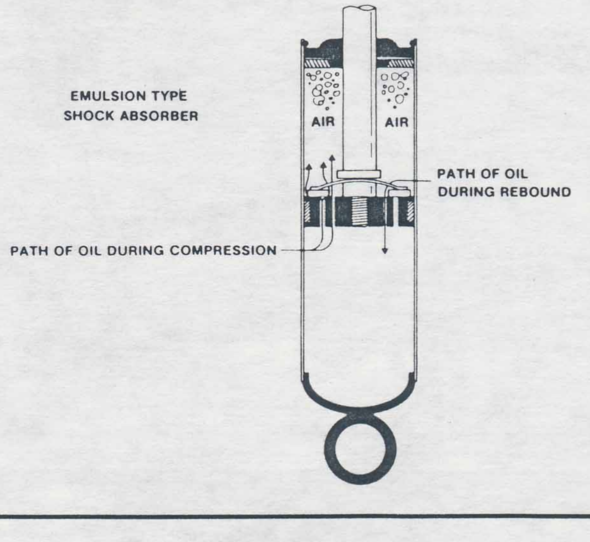

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Figure 30 illustrates a basic shock Dynamometer. The main parts start with the motor driven flywheel-crankshaft, which is attached to one end of the connecting rod. The other end of the connecting rod is hooked to the sliding ram. This arrangement converts the rotary motion of the crankshaft into the linear motion needed at the ram to stroke the shock in and out. To the upper end of the ram we connect the lower end of the shock absorber. The other end of the shock is connected to a **load cell** that is located at the top of the dyno, directly over the center of the ram.

Except for the electronics, that is all there is to a shock dyno, but before we get into the electronics we must fully understand the motion at the top of the ram. This motion is straight up and then a reversal to straight down. In order for the ram to reverse direction it must come to a momentary stop at each end of the stroke. The ram accelerates from a standstill at the bottom up to peak speed about half way up the stroke. (The exact point of peak speed is determined by the angle and length of the connecting rod) and then begins to slow down to come to a standstill at the top of the stroke. Then it begins to accelerate back down again going to peak speed about halfway down. The cycle is completed as the ram slows down to another complete stop at the bottom. Some dynos use a hydraulic drive mechanism that can vary the type of ram motion but for the most part this type of motion is universal in the industry. Familiarity with the way the ram speed varies according to its position in the stroke will help you comprehend the dyno charts.

The ram has an electro-mechanical device (a Potentiometer) hooked to it that sends a signal to the oscilloscope that follows the position of the ram. At the upper end of the machine we have the rest of the electronics in the form of a load cell. We hook the upper end of the shock to this load cell and it measures the amount of force transmitted by the piston rod as the shock is cycled. With the two electronic measurements we can now draw a chart on the oscilloscope screen.

The oscilloscope draws these charts with a high intensity illuminated dot that moves around the screen according to the signals received from the dyno. If you run the dyno with the load cell disconnected the dot will move up and down on the screen exactly as the ram does. The dot will stay right on the vertical line marked zero in **Figure 31.** If you connect the load cell and push up on it with your hand the dot will move to the left. If you pull on it, the dot will move to the right. Pushing on the load cell represents compression damping and pulling on it represents rebound damping. Now if you install a shock in the machine and turn on both signals, the dot will draw a pattern that resembles an oval in shape. As the dot moves upwards the compression damping will move it to the left and because shock damping is sensitive to the speed of the piston, the dot will move farther and farther to the left as the speed increases. As the shock slows down near the top the damping diminishes and the dot will move back closer to the zero line. At the top the direction of travel reverses and as the shock moves downward the dot moves to the right denoting rebound damping.

The oscilloscope screen is phosphorous so the dot momentarily leaves a trace on the screen which can be photographed for a permanent record of the damping forces. **Figure 31** has three traces taken at various piston speeds. To construct a damping curve a multiple exposure picture would be taken showing many traces from the slowest to the fastest dyno speeds. Then the peak forces of each speed would be transposed to a piece of graph paper and then connected with a line that would reveal a curve of the damping characteristics.

That is the basic story of shock absorbers. The reality ot shock absorber design is the fine tuning inside these general boundaries.

FIGURE 30

COMPONENTS OF SHOCK ABSORBER DYNAMOMETER

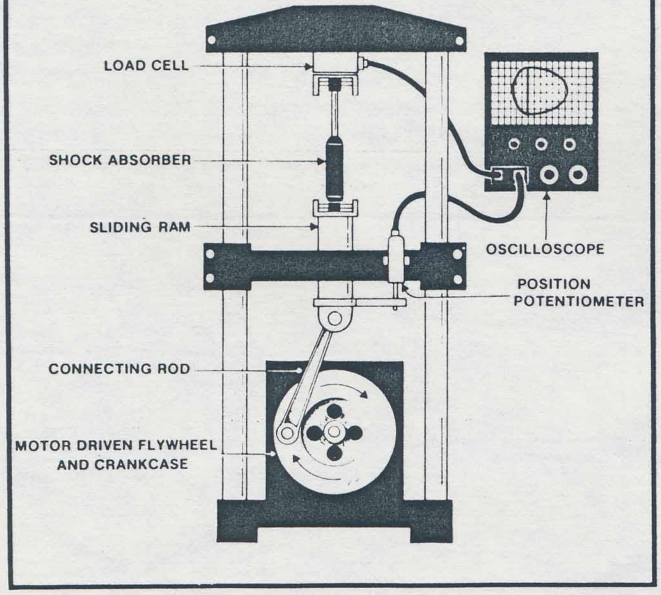
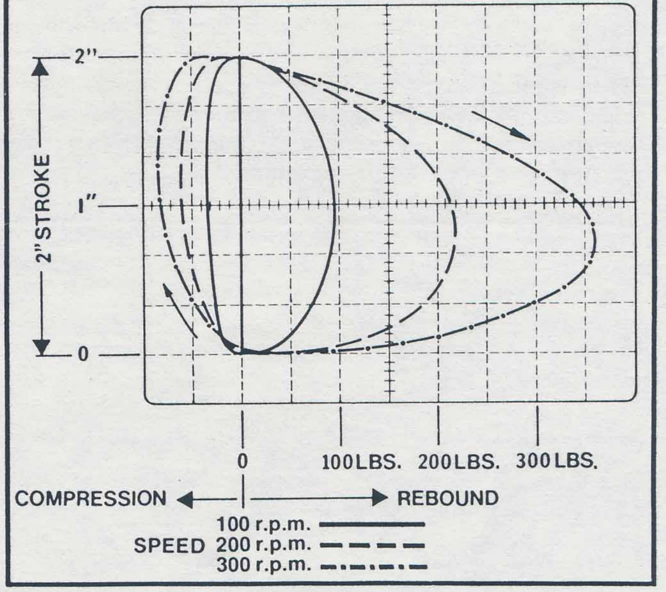


FIGURE 31



Chapter ...7 Stability and Steering

In the first six chapters of this book we explored in detail specific areas of motorcycle suspension and encouraged you to apply the concepts to your own motorcycle. Some of you may have begun experimenting and found the desired improvements rather elusive and sometimes accompanied by problems of a new nature.

This chapter will be more general, and concerned with those areas that are affected by changes to your suspension system. By understanding the interdependency of your suspension with steering geometry, weight distribution and transfer, and overall dimensions (wheelbase, height, center of gravity, etc.), it should be possible for you to better analyze your own test results. Many times a particular effect is attributed directly to a certain modification, when, in fact, the effect was caused by some accompanying change. Assumptions of this nature will cause each successive experiment to go farther and farther astray. Reality is that many factors are altered by even the simplest modifications to your suspension. Our hope is to expand your awareness to the point where you can assign the appropriate cause to any particular effect.

Geometry, weight and dimensions are fundamental in determining the handling character of each motorcycle. In fact, the choice of engine placement, steering angles and trail are the most significant differences between motorcycles on the market today. It is the careful juggling of these parts *in relation to each other* that makes the difference between a really fine handler and a motorcycle that is evil and scary. Keep in mind, however, that something as simple as moving the fork tubes up or down in the triple clamp will affect the balance of everything already mentioned. The key words above are "in relation to each other," because the final performance you feel is a combination of many contributing factors. Absolute values or rules cannot be postulated, however there are many guidelines or rules of thumb to use in your evaluations.

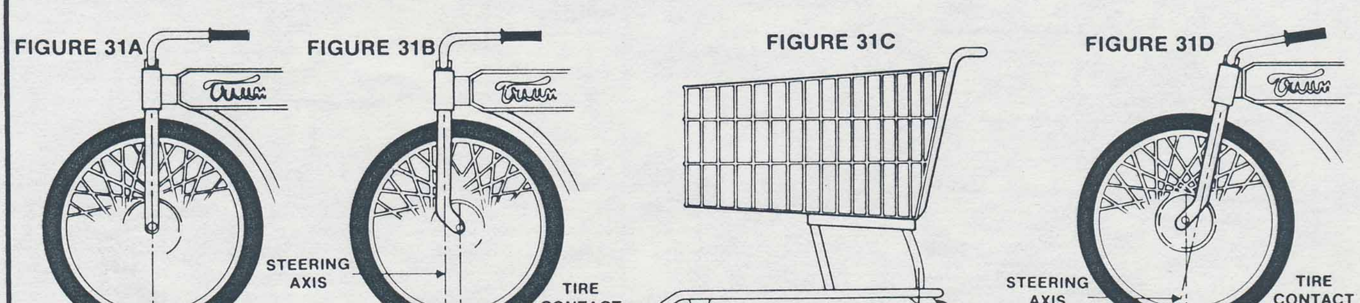
Let's begin by reviewing steering geometry. Primitive motorcycles had their steering-head angle or steering axis perpendicular to the ground and situated directly over the front axle **(figure 31A)**. Immediately the first test riders complained that the handlebars had to be held very firmly to keep the slightest bump from rotating the steering to full lock. In addition, there was no feedback or feel through the handlebars to indicate how much to move the handlebars in order to turn. Stability was also absent. In order to counteract these problems, a modification was made to move the point where the tire contacts ground to a position behind the point where the steering axis intersected the ground **(figure 31B)**. Now the bumps tended to push the contact patch straight back behind the steering axis and restore (or self-center) the steering. This geometry is exactly like that found on the rear wheels of shopping carts and is known as *trail* which creates castor effect **(figure 1C)**. Unfortunately, early motorcycles emulated our shopping carts in other respects and sometimes developed the same side-to-side wheel wobble seen everyday in your supermarket. To eliminate the effect, the steering axis was leaned back from vertical and the axle repositioned. This change retained the wheel's self-centering castor effect, but eliminated the wheel's tendency to swing past the center point of straight-line tracking as it self-centered, thus creating a wobble. This layout gave remarkably wobble-free straight-line stability and is basically the steering geometry in use today **(figure 31D)**.

(In the automotive world castor effect is achieved by leaning the kingpins back from vertical to create trail at the contact patch. The word "castor" is used to describe the number of degrees or the angle at which the kingpin is laid back. Motorcycles don't necessarily have a direct relationship between head angle and trail, therefore, "castor" as used by car people is misleading terminology. From now on we will use the term *trail* to describe the distance the tire contact patch "trails" behind the point at which the steering axis would intersect the ground.)

Head angle will be the term used to describe the angle the steering is leant back from a vertical.

You might assume that, because head angle and trail are such a panacea to stability, if some is good, more is better. Wrong. True, they do wonders for straight-line stability, but when you lean a motorcycle in a turn, the tire contact patch moves around off to the side of the tire, and trail then tends to continue turning the steering into the turn. Because that

EARLY STEERING GEOMETRY EVOLUTION



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is true, the combination of head angle and trail must be held to very critical limits and in correct proportion to each other.

Take a look at **figure 32A** and you will get a better idea of how steering geometry is diagramed. Trail is the distance the center of the contact patch lies behind the steering axis and is measured in inches. To find the trail, first project one line straight down from the front axle to the ground. This will reveal the position of the contact patch. Then project another line through the center of the steering axis to the ground. Trail is the distance from the contact patch to the point the steering axis intersects the ground.

There are a number of things that determine the amount of trail on a motorcycle. Put an alternate size front tire on the bike and the ground will cross the line drawn through the axle centerline at a different place. Essentially, a larger front tire makes the ground lower, increasing the trail. A considerably smaller front tire would create a ground plane that would result in *lead* which is the opposite of trail. Remember that a change in tire diameter will change both trail and head angle because the front of the motorcycle will be either lifted or lowered. A new tire could also have a different tread pattern or cross-section or both, further confusing you.

Figure 32B illustrates what happens to the trail if you change the head angle. Notice that the head angle was reduced simply by sliding the fork tubes up in the fork crowns. Notice also that the weight of the motorcycle has been lowered and that the wheel is now closer to the engine. Once again, many alterations result from one simple modification.

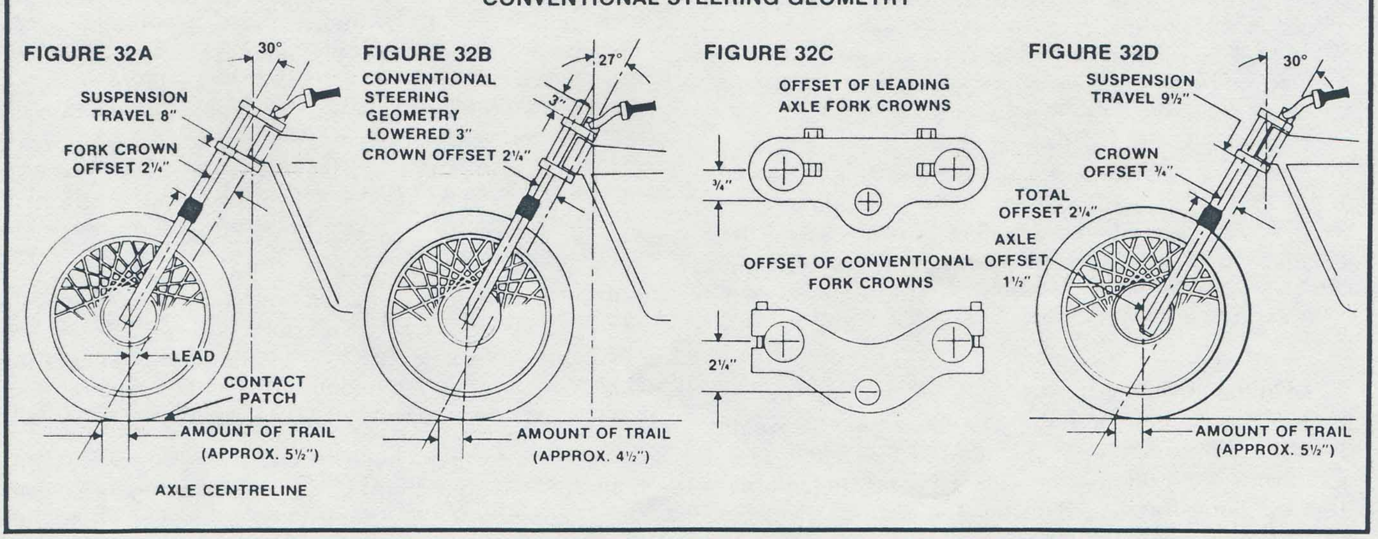
The primary components that determine trail are the fork crowns, or tripie clamps. By varying the offset of the fork crowns the trail can be adjusted without affecting too much else. **Figure 32C** illustrates how to measure fork-crown offset. One thing to take notice of in **figures 32A** and **32B** is that the amount of trail is inverse to the amount of crown offset. That is, more crown offset will produce less trail and conversely, less crown offset will produce more trail. This is often confused. Don't be misled by the offset of the leading axle crown. For this type of fork the offset is the sum of the crown offset plus the offset of the axle from the fork tube centerline. Almost always the total offset of a leading-axle configuration is identical to a standard fork's offset.

There is a common misconception that leading-axle forks work better because the steering geometry is different. Let us digress a moment to explain why we believe they are superior. There are several reasons for the more precise feel of leading axle forks. Structurally, they have considerably less flex, which makes control through the handlebars more direct. If you have taken a hard fall recently you probably recall having to align the handlebars with the front wheel after you picked yourself up. This is done by standing over the front wheel and clamping the tire between your knees. By twisting the handlebars you can put everything back in line. If you take a look at the fork tubes where they enter the fork slider you will notice that the two parts rotate in opposition when you tug on the handlebars. With a leading axle this rotation cannot happen without the fork sliders also swinging in a little arc about the axle offset. The extra arc creates resistance to rotation, thereby achieving more integrity between the handlebars and wheel.

If you look at the leading axle in **figure 32D** you'll notice that even though the steering geometry is identical to that in **figure 32A** the mass of the fork assembly is much closer to the steering axis. To the rider this means that when he swings the handlebars quickly there will be less momentum of the fork/wheel assembly. This reads out as less effort to control the steering, or more control for the same effort.

Again in **figure 32D** notice that the bottom of the fork is well below the axle. The top of the slider does not project as high above the tire, which means that for the same travel the fork crowns, frame and bearings can be closer to the axle. This results in a couple of benefits. One is that the center of gravity of the motorcycle can be placed much lower, a benefit in turning. Also, because the flexing fore and aft of the fork tubes begins right under the lower fork crown, the closer that fork crown is to the axle, the shorter the lever arm to do the bending. The result: less flex. Lastly, in this day and age of extra-long fork travel, leading axles with their lowered sliders are necessary to keep the overall height to an acceptable level.

CONVENTIONAL STEERING GEOMETRY



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So much for leading axles; let's discuss the effects of head angle and trail. It is common practice on motorcycles today to use a specific head angle for each type of event, a pretty good indication that changes in head angles will make big differences in handling. The compromise always seems to center around the amount of ease in turning or straight-line stability desired. The range of head angles begins with speedway bikes at 14-16 degrees, where you can't operate in anything less than a full-lock slide, and progresses to 32-33 degrees on desert racers where the motorcycle must continue straight ahead regardless of the size of the obstacle it just hit. In between those extremes we have flat trackers at 25-27 degrees, road bikes at 26-29 degrees and motocrossers at 29-31 degrees.

Most racers know to within half a degree the correct head angle for their particular motorcycle. The general rule is that a lot of head angle will make the motorcycle go straight, but it will not be easy to initiate a turn or to slide. Conversely, a motorcycle with less head angle will shake its head at high speed and be prone to tank slappers in the whoops, but it will slide like crazy. If you experiment with your motorcycle by sliding the forks in the crowns or by raising or lowering the rear suspension with different shock lengths, you will begin to feel the best compromise for your type of riding. Put a bubble protractor on your fork tubes when the suspension is topped out and arrive at a number that suits your style. Raising or lowering the chassis at either end by one inch will generally change the angle by almost one degree.

Suppose that after much experimenting your motorcycle cannot be made to turn or go straight, or that by the time you have achieved one handling goal the other has deteriorated enough to be dangerous. That is a good indication that something else is not quite right. The culprit might be trail. If the motorcycle seems dead stable, but when leaned over the steering tends to continue turning, it is a good bet that it has too much trail. On the other hand, if the steering wanders excessively it will show up worst in deep sand. In sand, because the tire is down in the sand opening a groove, the tire contact patch moves forward on the tire and more trail is required to counteract this phenomenon. Another culprit might be weight distribution. This affects the interaction of the tire tread with the ground and determines the amount of *traction* at that point. As a general rule, the heavier the load on them, the more traction or bite motorcycle tires have. The balance of these loads front to rear will determine whether a motorcycle will push the front tire when cornering or oversteer with the rear wheel slightly hung out. Too little weight on the front tire will cause the push and too little weight on the rear will cause the oversteer.

The difficult part is determining whether the pushing condition is caused by too much trail or too little weight. Sometimes the answer can be found by trial and error. Insufficient weight on the front tire is a common fault these days with long-travel suspension. The addition of suspension travel is invariably accomplished by extending the front forks, which raises the chassis. When the wheel is allowed to extend down farther, it also moves away from the engine, making the wheelbase longer. This makes the static weight on the front tire lower. The increase in chassis height also allows more weight transfer to the rear when accelerating, causing even less weight on the front.

If you have a front-end pushing situation only when accelerating, it indicates that the center of gravity is too high, though not necessarily too far back. If the push is there whether the throttle is on, off, or neutral, it indicates that the center of gravity is too far back. The whole subject of weight transfer will be investigated in Chapter 8, but something to consider is that weight distribution can also be altered at the rear by use of a different length swing arm. Also be aware that changes to the rear suspension, like a different shock length, affect the head angle and trail in much the same manner as changes in the front height.

Up to this point we have primarily been talking about chassis angles and dimensions measured while the motorcycle is stationary or *static*. If you have handling problems that still can't be explained, perhaps those critical angles are quite different when your motorcycle is in motion or its *dynamic* condition.

One of the compromises brought about by long-travel suspension is that under braking and acceleration the chassis is free to *pitch* (or rock) fore and aft much farther than before. We have seen that raising or lowering the chassis at one end changes the head angle and that pitching is a raising of one end and a lowering of the other. What this means is that under braking, where straight-line stability is important, the chassis nose-dives and much of the head angle goes away. When this happens it is very difficult for the rider to hold the handlebars straight ahead.

Under acceleration coming out of a turn the chassis can rock backwards, causing the forks to extend and provide an excess of head angle. This makes it difficult to maintain your line accelerating out of a turn.

So far the improvement in ride over rough terrain has more than compensated for the sacrifices in the turns caused by long-travel suspension. However, if you are not careful with the details, the sacrifices become more than is tolerable. It is important that both front and rear suspensions ride at about the same average distance into the travel when in motion. Spring-rates that are too soft, especially at one end, will allow excess pitching. Too little shock damping will also allow the chassis to pitch farther than necessary and suddenly enough to be quite a surprise to the rider. It should also be noted that some pitching is desirable in order to start a turn; if your motorcycle doesn't want to turn without first having braked hard it is a good indication that your front suspension is too stiff or your static head angle excessive.

Another dynamic condition that might be a problem is *pumping down* and *pumping up* of the suspension. This can happen at either or both ends of the motorcycle. It is caused by an improper amount of shock damping for the spring rates being used.

Pumping down is caused by an excess of return (or rebound) damping in the shock absorber. When the suspension is compressed by a bump the rebound damping is supposed to let the suspension return to its original position very gently. However, if you have an excess of rebound

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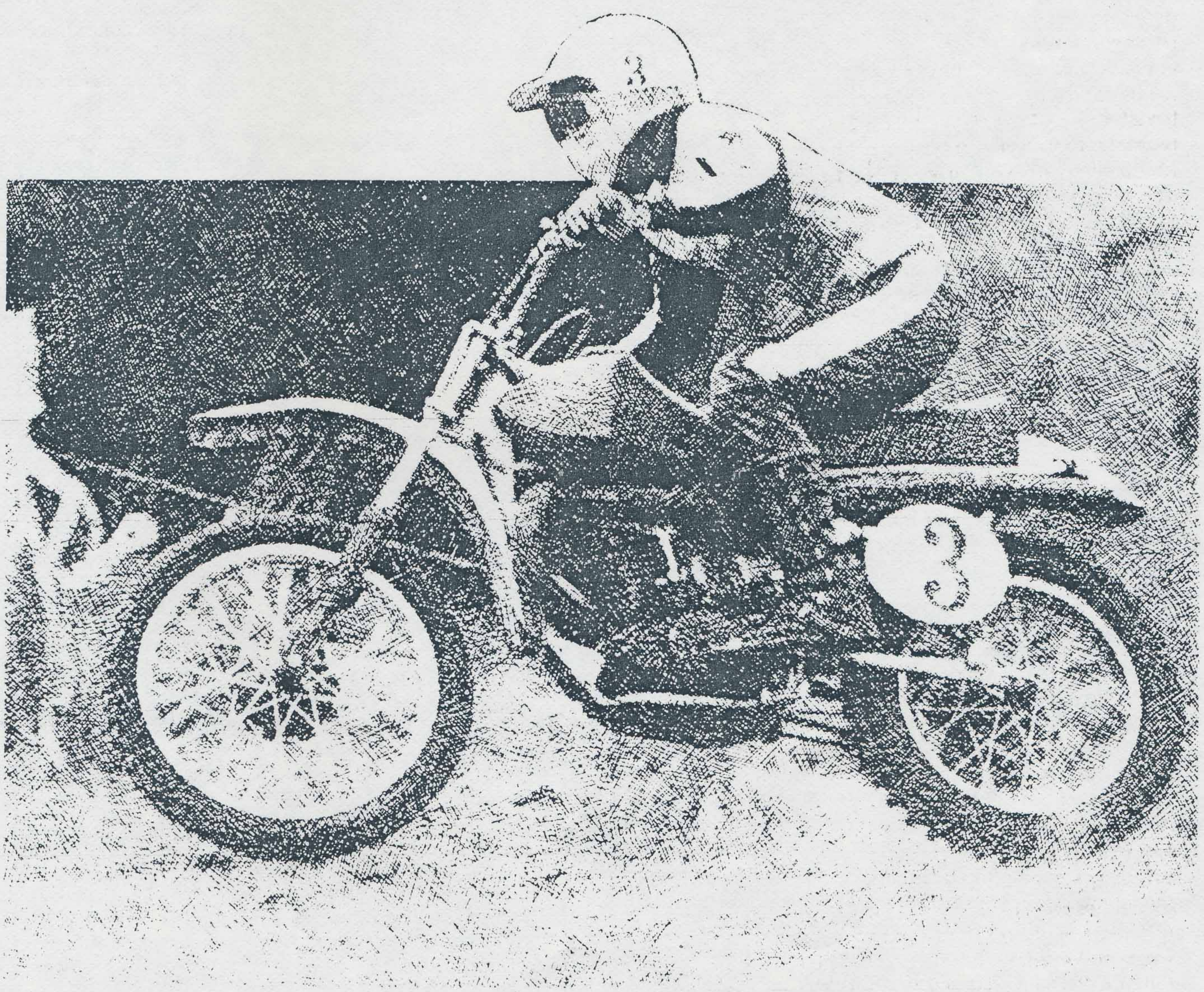
damping the suspension returns so slowly that you are likely to encounter another bump before the suspension has fully recovered. This becomes very apparent on a series of bumps or a washboard surface. The symptoms on washboards are a very harsh, chattery ride and a reduction in forward traction. This is because the suspension rides at an average (or dynamic) height a couple of inches pumped down. To the wheel this seems like a couple of inches of extra preload.

The symptoms on a series of larger bumps are much more severe. The scenario goes like this. As you traverse these bumps the suspension does not return quickly enough, so each successive bump is encountered with a progressively higher impact. The suspension is pumping down farther with each bump and there remains less and less suspension travel to soak up the bumps. Eventually the remaining travel will be inadequate and the suspension will bottom out and pitch that end of the motorcycle into the air. Unfortunately, this "always" occurs when there are still more bumps left to negotiate, which can be very stimulating to the rider.

Pumping up is the opposite of pumping down and is caused by an excess of compression damping accompanied by too little rebound damping. The suspension tends to ride in a raised up condition which is desirable for events like desert races. The softness of the ride is usually very good in a pumped-up condition except that if this condition happens in one end only it is likely that too much pitching of the chassis will occur, bringing on steering/stability problems. The same is also true of pumping down in one end.

This is why it is important to use information linking damping to spring rate, as found in the S & W catalog.

As you can see there are many things going on in your chassis at the same time. If you are getting test results that are bizarre, consider all the various gremlins presented here.



Chapter ...8 Weight Transfer & Anti-Squat

This is the final chapter in this manual. Based upon that fact, the subjects were chosen to stimulate your curiosity and, perhaps, establish a foundation for a whole new wave of experimentation to a long-neglected part of your suspension system.

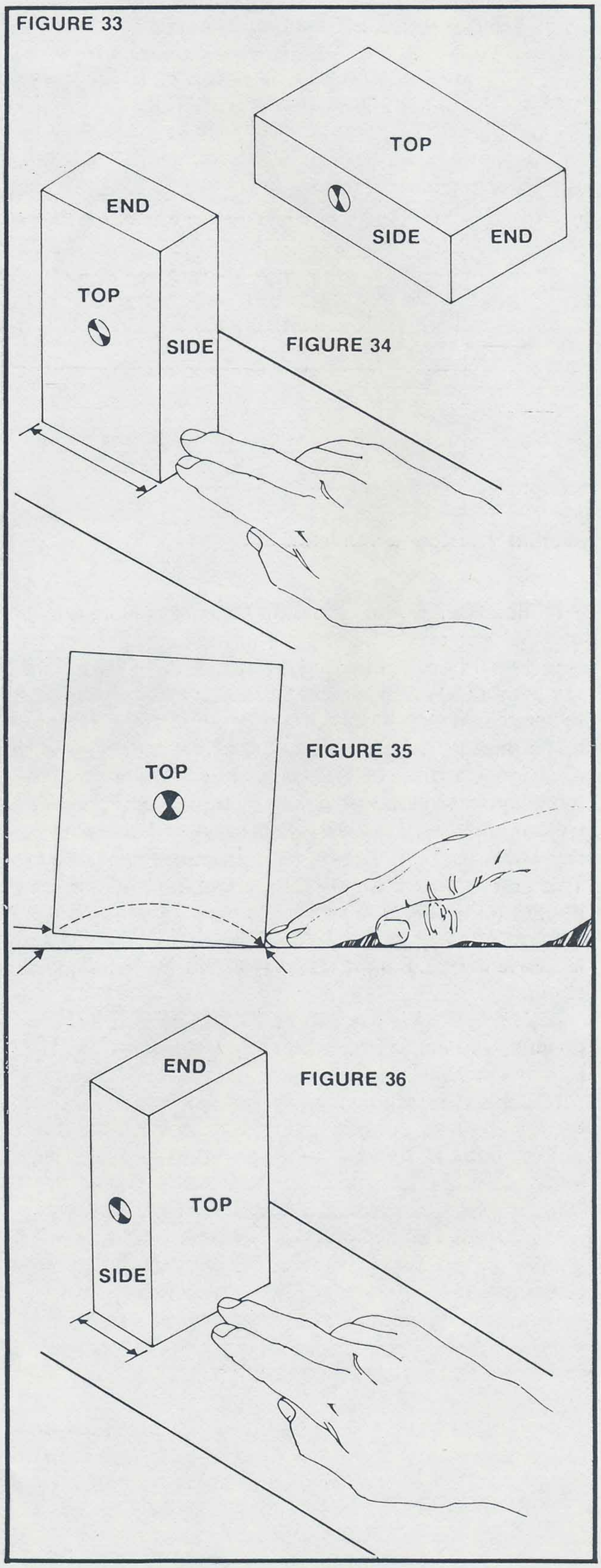
The concepts are more abstract than anything presented here previously and may be slightly more difficult to grasp. However, you should not be intimidated by, or take casually, the theoretical nature of this material.

The first part of this chapter is concerned with **weight transfer.** This dynamic condition happens every time your motorcycle moves under its own power and it has a profound effect on every aspect of your motorcycle's handling behavior.

The second part of the chapter focuses on a rear suspension condition known as **anti-squat.** Anti-squat is a dynamic force in the rear suspension that can add or subtract to the downforce at the point the tire contacts the ground. This force varies with the application of engine power or the brakes and its value is determined by the geometry between the swing-arm and the chain and the ground.

The anti-squat concepts presented here are derived from research performed by Trevor Harris of Harris Dynamics in Costa Mesa, California. To the best of our knowledge, these ideas are completely original and new as applied to motorcycles and have not been published anywhere prior to this publication. We thank him for making these concepts available and for his editorial contribution.

What exactly is weight transfer and what does it do to your motorcycle? Specifically, it is the change in load at the tire contact patch as a result of acceleration of the vehicle. For a clearer understanding of this statement, follow this text using **Figure 33** through **Figure 38** as a visual guide. It would be even better if you would perform this series of tests yourself while following along with the text: Place a small box about the size and proportions of a box of kitchen matches on a smooth table, so that it is standing on end. **Figure 33** establishes the end, side and top. Now, push it across the table with your fingers at a point as close to the table as you can, as in **Figure 34**. Start slowly and gradually build up the speed as you push the box across the table. Do this at a faster and faster speed until the box tips over backwards. At this point, there is total weight transfer. All the weight that was being supported at the front of the box is now being supported at the rear of the box. **Figure 35** diagrams this condition. Now, perform this same procedure, but this time push on the topside of the box, again as close to the table as possible, as shown in **Figure 36.** Notice the box tips over much easier this time.



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Very carefully, climb aboard the motorcycle and assume your normal riding position. Have your friend make a mark on the masking tape where the plomb-bob intersects it. This is the center of gravity. Needless to say, this almost upside down riding position is very fatigueing so be prepared to read the plomb-bob quickly. Also, don't perform this procedure unless you have full confidence in your overhead hoist and your attachment to it.

Here are some figures taken from a Suzuki RM 370B with half a tank of gas. The rider is 175 pounds and 6'2" tall. The wheelbase is 57.5".

	WEIGHT FRONT	WEIGHT REAR	TOTAL WEIGHT	FORE & AFT C.G.	VERTICAL C.G.
BIKE ONLY	112 LBS. 46%	133 LBS. 54%	245	31.05" back FROM AXLE	21.25" FROM GROUND
BIKE/ RIDER	161 LBS. 38%	266 LBS. 62%	427	35.65" BACK FROM AXLE	30.5" FROM GROUND

Now, let's plug these figures into the formula:

$$\text{Weight Transfer} = \frac{(.71)\,(427)\,(30.5'')}{57.5''}$$

Weight Transfer = 160.8 pounds

Notice the amount of weight transfer is just short of the original load on the front tire. This indicates the motorcycle is just on the verge of doing a wheelie. Take note of the fact that almost all of the weight transfer has taken place before the front wheel has lifted off the ground. Weight transfer, to some degree, is happening all the time as long as there is acceleration or deceleration. It is not necessary for the motorcycle to pitch or change attitude for there to be a weight transfer. Conversely, pitching of the motorcycle does not necessarily change the amount of weight transfer. This is true only if the pitching does not change the C.G. height, as is the case when the front raises and the rear lowers. However, most pitching *is* accompanied by a major increase in C.G. height, as is the case when doing a wheelstand.

If you desire to keep the front tire in contact with the ground, you can compensate by altering any of the components in the formula or you can alter the static weight distribution so that after weight transfer there is some remaining load on the front tire. If you don't want to alter the static weight, you can reduce the weight transfer which will also leave you with some remaining load at the front tire.

When you look at the formula, you will notice that the wheelbase is divided into all the other components. This means that in order to reduce weight transfer you must either lengthen the wheelbase or reduce any or all of the other components. The difficulty lies in the fact that alteration of any of the components creates compromises somewhere else in the motorcycle. The trick is to select the compromise that best suits your purpose.

Before you decide on which compromise to tolerate, let's look at the effects of weight transfer. You have seen that it alters the load at the front and rear tire contact patches. However, up to now we have just talked about weight transfer to the rear under acceleration. When the brakes are applied, exactly the same dynamic condition is produced, but in the opposite direction. The front tire load increases and the rear gets lighter; again a shuffling of the loads at the tire contact patch. The effect of these load changes is the traction between the tire and the ground varies accordingly.

In order to interpret these load changes, you must know something about the interaction of the tires and the ground. *The traction increases as the load increases.* This statement is the most important of this entire chapter and again will be important in the discussion of anti-squat. It is always true when the track surface is dirt and generally true for pavement, but not always.

With this idea firmly in mind, let's look again at the options available for reducing weight transfer. The first option is to lengthen the wheelbase. This will definitely lower the weight transfer, but the static weight distribution will be effected, depending on how the chassis is lengthened. If it is lengthened at the rear only, the static load will be less there and more in the front. If the motorcycle always tended to wheelie too easily and was generally unpredictable while turning, this modification would help considerably. But, if the motorcycle was balanced before, it probably will now experience too much loss of traction at the rear. This means the acceleration will be less (lower G force), which means even less weight transfer, further reducing traction. *The traction is contingent on the amount of weight transfer fed to it, which is contingent on the amount of acceleration fed to the C.G., which is contingent on the traction available.* This cycle of events feeds on itself in a closed-loop fashion. This is an important concept to understand and will help explain why some minute changes have a drastic effect.

If the wheelbase is lengthened at the front only, it will cause the static load to be less at the front and more in the rear. This will produce less weight transfer, but since there is now more weight in the rear, the traction will be increased which could cause faster acceleration which, in turn, could return the weight transfer back to the same level as before. But now the load on the front is less so the wheelstand problem may become worse. When attempting to turn or brake this new configuration, you will really regret the effect of not enough front tire loading.

Let's consider some of the other options in the formula that are written over the wheelbase. The first figure is the **g's** of acceleration. In order to reduce weight transfer, it is necessary to make this figure smaller and/or acceleration slower. We have never heard of a race where less acceleration is a virtue, so this option is out.

The next available option is to lower the overall weight. Since most current motocrossers are right at the minimum allowable weight, there is little that can be done in this area.

The last, and perhaps the best option of all, is to lower the C.G., since the static weight distribution does not necessarily have to be changed. One of two things to be aware of at this point is that the C.G. height (30.5") is roughly one-half of the wheelbase (57.5"). What this indicates, as far as weight transfer is concerned, is one inch of C.G. height change is equivalent to two inches of wheelbase change. This may help you decide which is the easiest part of your motorcycle to modify.

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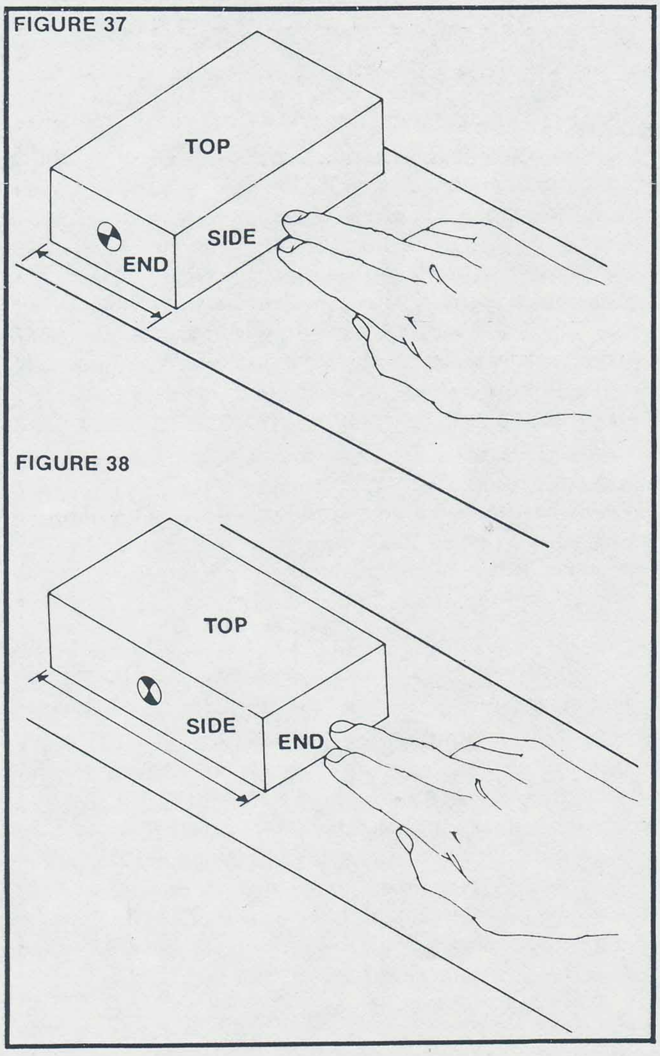
Several points are evident from this sequence. First, the box, which obviously has no wheels or suspension, has weight transfer when it is accelerated; second, the weight transfer is increased if the acceleration is increased; and third, weight transfer varies in relation to the size of the base. (The base is broader in **Figure 34** than in **Figure 36** and the box can be moved faster without tipping over).

Now, push the box again as in **Figure 34** and make a comparison as to the ease of tipping over with the box positioned as in **Figure 37**. Notice that in the **Figure 37** configuration the box is very difficult to tip over even though the size of the base is no greater than in **Figure 34**. In this case, the center of gravity of the box is moved much lower in relation to the point being pushed on. From this it is established that the lower the center of gravity is in relation to the point of reaction, the less the weight transfer.

In **Figure 38** the box is positioned to have the broadest base and the lowest center of gravity. In this configuration, it is virtually impossible to tip the box over, which indicates a combination of broad base and low C. of G. has the least weight transfer of all.

FIGURE 37

FIGURE 38



These simple tests show that several factors have a direct and predictable bearing on weight transfer, and therefore, that weight transfer is a mathematical function capable of calculation. In order to set up this calculation, we must transform the components of our box test to those of an accelerating motorcycle. Begin by substituting the motorcycle's front and rear tires for the lower corners of the box. Now, the wheelbase becomes the "base." The point where the rear tire contacts the ground is now the reaction point and corresponds with the point where you pushed on the box. The engine now provides the acceleration instead of your finger and the motorcycle/rider combination has its own center of gravity.

These components are applied to a formula:

$$\textbf{Weight Transfer} = \frac{\textbf{(g's) (Weight) (C.G. Height)}}{\textbf{Wheelbase}}$$

Weight Transfer is the amount of weight in pounds subtracted from the load at the front tire and then added to the load at the rear tire.

g's is the rate of acceleration of the motorcycle. (For a typical open class motocross bike, a theoretical **g** figure of .71 will just lift the front wheel off the ground with the rider sitting in an upright position).

Weight is the total weight in pounds of the motorcycle/rider combination.

C.G. Height is the height of the center of gravity of the motorcycle/rider combination measured in inches from the ground.

Wheelbase is the distance in inches between the front and rear axles.

If actual figures are plugged into the formula, several additional aspects of weight transfer are revealed. These figures can be collected by placing your motorcycle on two bathroom scales, one tire on each. Then run a tie-down strap from each end of the handlebar straight out to the nearest adjacent immovable object. The tie-downs will keep the motorcycle from falling over without influencing the weight shown on the scales. Now, get on the motorcycle and have a friend take some readings at the front and rear. For your own curiosity, try several different riding positions to get a feel for the effects of body english on the weight distribution. From this procedure, you can get the total weight by adding the two readings together and then can establish the fore and aft position of the center of gravity by proportioning the two readings in relation to the wheelbase. (If the total weight is 427 pounds and the weight is distributed 161 pounds front and 266 pounds rear, the percentage of weight distribution will be 38% front and 62% rear. The fore and aft C.G. will be closer to the rear tire and, specifically, 62% of the wheelbase back from the front axle or 35.65" back for a 57.5" wheelbase).

Finding the center of gravity height is slightly more difficult, but not much. First, take a strip of masking tape and put it straight up and down on the motorcycle exactly where the calculated fore and aft C.G. is located. Now, attach the front tire to some overhead hoist or crane and raise the motorcycle until it is hanging vertically from the front only. Attach a plomb-bob to the front axle so the bob weight can swing freely across the strip of masking tape.

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Notice that no mention is made of the fore and aft position of the C.G. in the formula. This tells you the static weight can be altered without effecting the weight transfer as long as the C.G. height remains constant. However, you have already seen that altering the static weight does change the behavior of the motorcycle's handling.

If you analyze the type of "body english" necessary to make your motorcycle behave correctly, you will know whether to play with the C.G. height, the wheelbase or the static weight. If you find yourself moving forward under acceleration and backward while turning or braking, it is a good sign the wheelbase is too short. If you move around in the opposite manner, it is a sign the wheelbase is too long. If your motorcycle is already pretty short and you find it necessary to move in the same directions as if it were too long (back on acceleration and forward for turning), it's a good bet your C.G. is too low. If you have a long motorcycle and you find you still have to climb over the bars on acceleration and scoot back for turning, the C.G. is probably too high. If you always end up sitting way forward, the static weight distribution is biased too far to the rear. The converse is also true.

The best compromise between static weight distribution and weight transfer is achieved when very little "body english" is required. This is not to suggest that "body english" is inherently bad. It is a very useful tool and if your motorcycle responds to a lot of moving around, by all means do it. However, it is less fatiguing to ride a well-balanced motorcycle that allows you to sit or stand in just one spot.

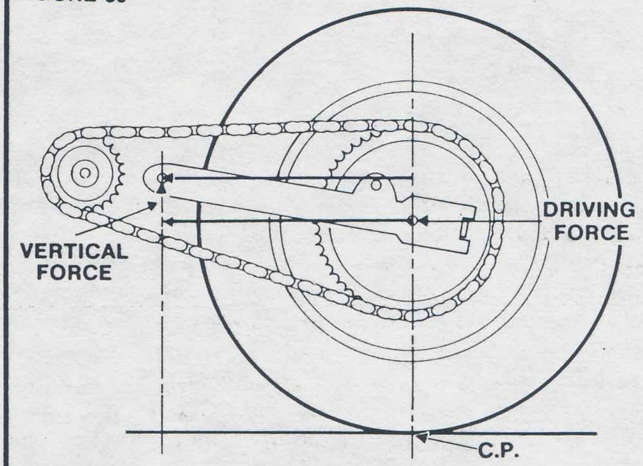
Thus far, we have been searching hypothetically for a means to reduce weight transfer. It is likely that your motorcycle probably doesn't need less weight transfer. We have used this example merely to illustrate the components that do effect weight transfer. The fact is, the optimum amount of weight transfer is the most you can use without causing any real compromises elsewhere.

ANTI SQUAT

As noted earlier, anti-squat is a force in the rear suspension that changes the load at the tire contact patch. To fully understand this dynamic phenomenon, it is necessary to go back to some fundamentals.

Figure 39 illustrates a typical motorcycle rear suspension. When power is applied to this mechanism, the rear tire claws at the ground at the contact patch (designated C.P.). This causes a reaction which propels the motorcycle forward. Since the tire revolves about frictionless bearings at the rear axle, the forward driving force is imparted to the motorcycle at the axle. The axle subsequently pushes on the end of the swing-arm in a direction that is parallel to the ground. However, the swing-arm is not parallel to the ground so the driving force, which is straight ahead, gets divided into two components: one, horizontal, and the other vertical. The vertical force becomes a portion of our mysterious anti-squat by pushing up on the chassis at the swing-arm pivot. Elementary — right? Wrong! This vertical force is only a portion of the force energized as power is applied and these other forces work in opposition and cancel part of the swing-arm vertical forces.

FIGURE 39



Before we go any further, let's make sure you understand how a force can be divided and go in two directions. Some simulations will help make this clear. Take an ordinary pencil and hold it lengthwise between the tip of your forefinger on one hand and the forefinger on the other hand (**Figure 40** illustrates the positioning of the pencil). Don't use a sharpened pencil or you will injure one of your fingers! If the pencil is sharpened, put an eraser over it. Now push on both ends of the pencil and envision the pencil as representing a swing-arm with one end being fixed and stationary, like the frame end, and the other as being free to swing in an arc, like the axle end. Maintain your pressure on each end of the pencil and swing the moveable end down out of line with the fixed end as shown in **Figure 41.** Notice the

FIGURE 40

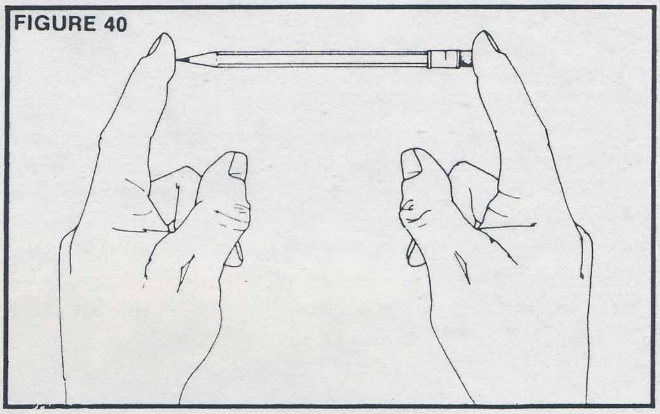
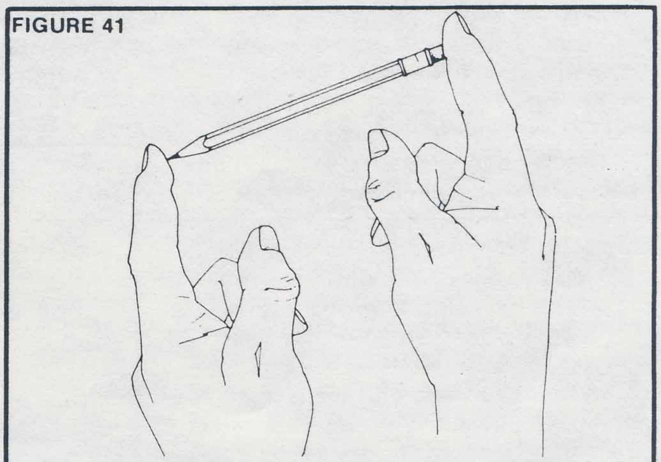


FIGURE 41



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different feel at your imaginary fixed end. Even though there is a force trying to push your finger over backwards, there is another force trying to push your finger upwards. Notice those forces trade progressively from horizontal to vertical the more the pencil is angled. This indicates the *angle* determines the distribution of the two components. Now notice the *upwards* vertical force at the stationary end of the pencil is counteracted by a *downwards* vertical force at the movable end. (For every action, there is an equal opposing reaction). This is an important point and illustrates the main benefit of anti-squat. This downwards force is translated to the motorcycle as an increase in load at the contact patch which increases the traction.

Diagramming the distribution of component forces is done as shown in **Figure 42.** In Part **A**, a rectangle is drawn around the swing-arm, having the top and bottom parallel to the ground. The length of the top and bottom represent the percentage of the original force that is horizontal. The length of the sides represents the vertical percentage. Part **B** shows what happens to the forces if the angle is increased. In this instance, the angle is 45° and all four sides are equal, forming a square. This means the horizontal force is less and the vertical force is more than before, but they are equal to each other.

Let's get back to the real world and look at the other forces that contribute to our anti-squat forces. If you look at **Figure 39** again, you will see the chain also works at an angle in relation to the ground. We know there is a force imparted to it from the engine. However, this time the force is a pull instead of a push. Because the chain is in tension, we can look on it as just another link in our system much the same as the swing-arm, except in the case of the swing-arm we can both push and pull. The ends of this link are considered to be at the tangent points where the chain leaves the rear sprocket and enters the engine sprocket. **Figure 43** shows the component forces of the pulled chain.

The next step is to devise a system to diagram these component forces to give us their resultant force and how it relates overall to the chassis. **Figure 44** illustrates the technique for determining the relative anti-squat. Begin by drawing a straight line from the center of the axle through the swing-arm pivot and continue it out into space. Then draw a line connecting the two chain tangents and extend it into space. These two lines will cross each other somewhere ahead of the engine sprocket. From the intersection point, draw another line back down to the tire contact patch. This third line indicates the degree of anti-squat in the system. The angle between this line and the ground is the measure of the amount of anti-squat. Again, the horizontal and vertical components can be derived by drawing a rectangle around this angled line.

FIGURE 42A

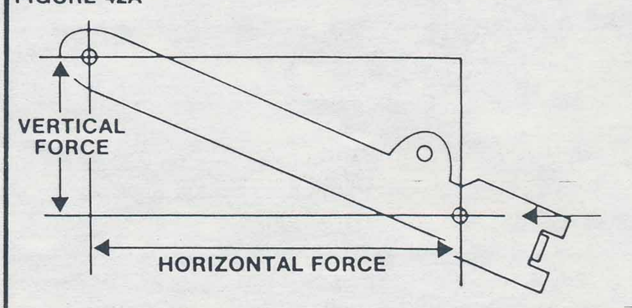


FIGURE 42B

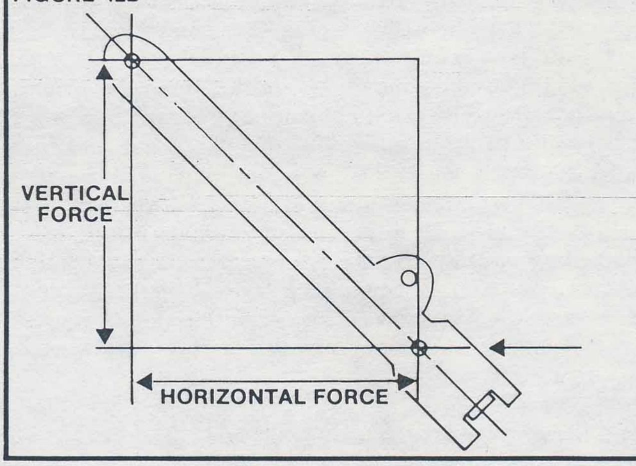


FIGURE 43

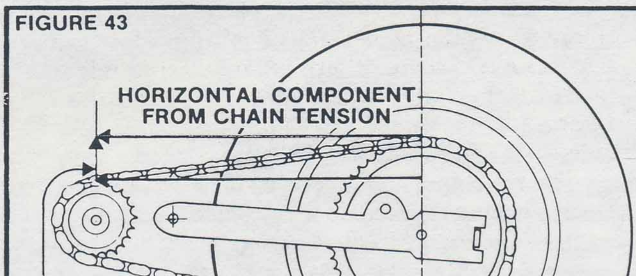
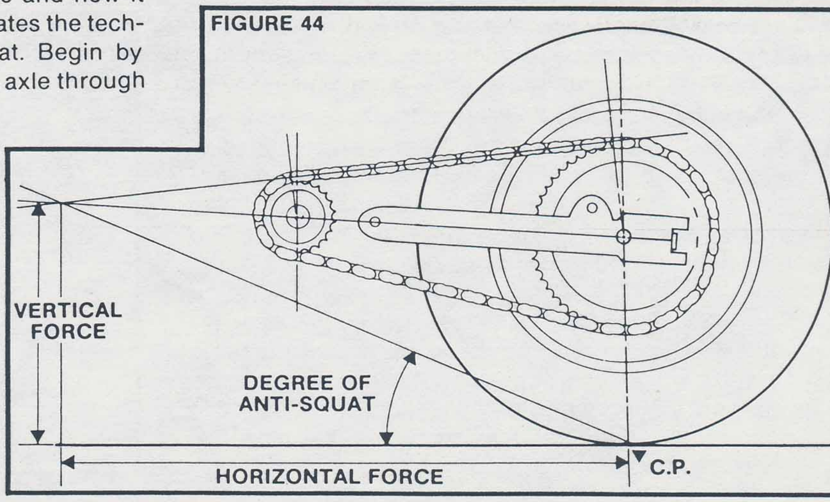


FIGURE 44



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If you know the driving force, you can calculate the vertical force with this simple formula:

Vertical Force = $F \times \frac{A}{B}$

where: F = **the driving force in pounds**
A = **the vertical component**
B = **the horizontal component**

The point where all three lines intersect is known as the *instantaneous force center* and is the location where the component forces act on the chassis. Watch what happens to the force center when the rear suspension is compressed **(Figure 45).** Notice the force center has moved down and forward, introducing a drastic reduction of the anti-squat angle. This shows that the anti-squat somehow varies according to the position of the suspension — a condition not conducive to promoting a stable, predictable motorcycle.

The position of the instantaneous force center and the manner in which it moves determines the effect anti-squat will have on your motorcycle. There are many ways to modify the force center characteristics. It is important to decide on the exact type of anti-squat that will be beneficial rather than a liability. Very little empirical research has been done as to the optimum anti-squat. At this point, it is conjecture and we only can make an educated guess as to the perfect compromise.

Immediately, it seems reasonable that anti-squat should not vary with suspension movement, especially if it diminishes as the suspension compresses. It should at least stay constant all through suspension travel and, perhaps, it should increase with travel. This is one trait we can put on our list of preferences.

The real question is the amount of anti-squat to start with. Our testing revealed that too much anti-squat is as bad as too little, but for different reasons. Too much anti-squat will physically lift the rear of the motorcycle when power is applied as long as the traction remains constant. If the traction is intermittent, the suspension will "relax" with every interruption of traction and tense up or "flex" as the traction is restored. To the rider, this will read out as a chatter in the rear suspension. This chatter can be caused by an irregular track surface or it can be induced by the anti-squat itself. As the anti-squat is energized by engine power, the tire is momentarily pushed against the ground with extra force which increases the traction which in turn increases the driving force which increases the anti-squat. Sound familiar? This closed-loop cycle is very similar to that discussed in weight transfer, earlier.

If the anti-squat gets too strong, the rear suspension virtually will become locked in one position. In this condition, the suspension will be unable to comply with track irregularities and will cause the entire chassis to follow the track undulations. The motorcycle will have enough of its own momentum (and perhaps upward momentum from suspension flex) so that it will not follow the ground on the downside of each bump. As the tire leaves the ground, the traction is interrupted for a moment, allowing the suspension to "relax." When the tire touches the ground again, the anti-squat is once again energized, causing the suspension to "flex." If this sequence is repeated over and over at a fast enough rate, a chatter will develop. The chatter can be so bad that intermittent daylight can be detected between the tire contact patch and the ground. However, chatter can be a problem and be felt by the rider well in advance of this extreme condition. Chatter will reduce the forward acceleration and the motorcycle will tend to wiggle from side to side.

Too little anti-squat causes the reverse effect. As power is applied, the tire is jerked away from the ground, causing a general reduction in traction. This condition is known as pro-squat and is the equivalent of negative anti-squat. Many of today's motorcycles actually have pro-squat when the suspension is fully compressed. In addition to reducing the traction, pro-squat also tends to keep the suspension compressed just at a time when you would like it to return to its original position quickly. Pro-squat has the same effect as shocks that cause "pumping down." Excessive anti-squat can do the opposite and launch your motorcycle off of jumps by quickly extending the suspension.

We believe the optimum anti-squat to be an amount somewhat less than most of today's motorcycles have when their suspension is fully extended, but only if the anti-squat does not change very much when the suspension is compressed. To take full advantage of this configuration, it might be necessary to increase the static weight and/or weight transfer to the rear tire to restore any traction lost by the reduction in anti-squat. Shock absorbers might need some redesign because those available on the market today are most likely compensating for excessive anti-squat.

In **Figure 46**, compare the effect on the instantaneous force center (I.F.C.) with that in **Figure 44**. In this case, we have lowered the swing-arm pivot in relation to the engine sprocket. This I.F.C. has moved down forward, partially achieving our goal of reduced initial anti-squat. The drawback to this modification is that pro-squat is more pronounced than ever when the suspension is compressed. This is unsatisfactory.

Study **Figure 47** to see the effect of gear ratio changes on anti-squat. Notice a smaller wheel sprocket moves the I.F.C. further forward and reduces anti-squat. Notice that a larger engine sprocket has almost the same effect.

FIGURE 45

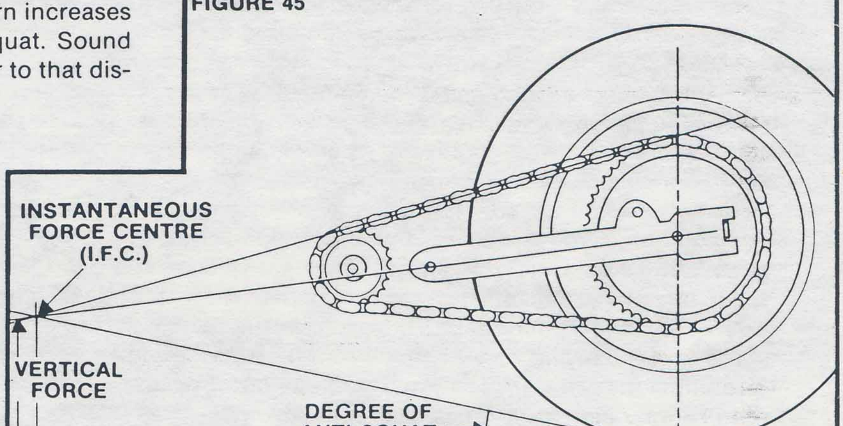

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FIGURE 46

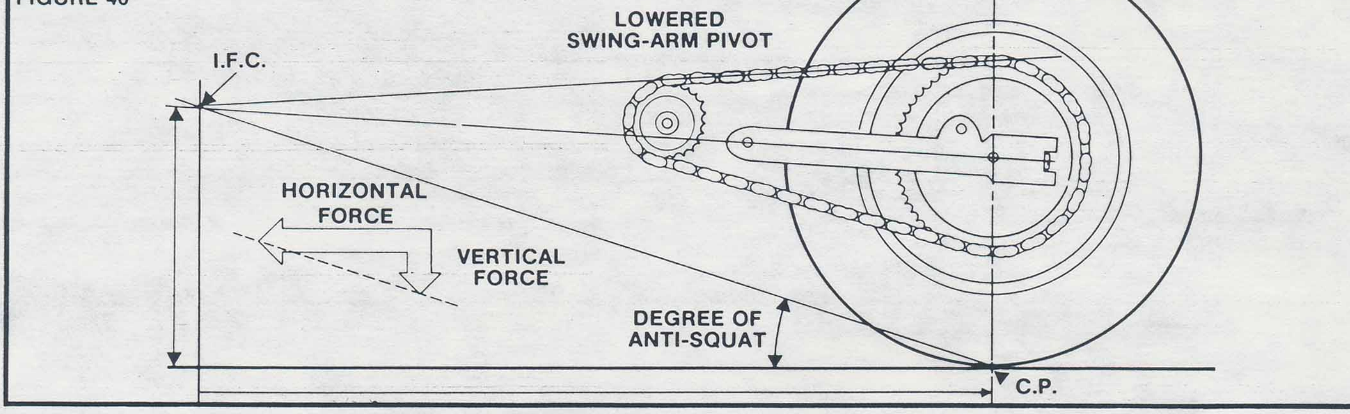


FIGURE 47

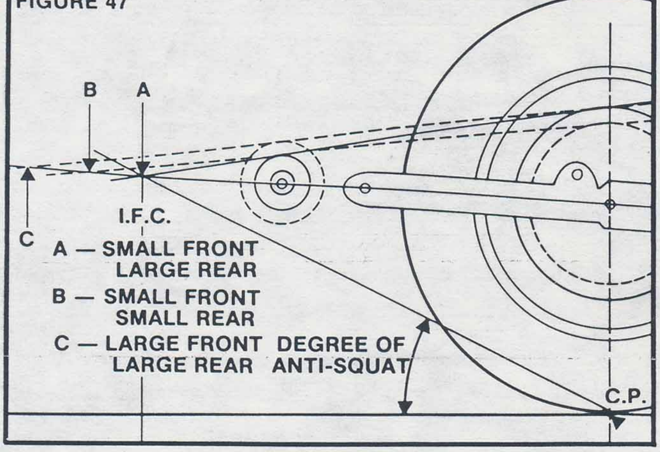
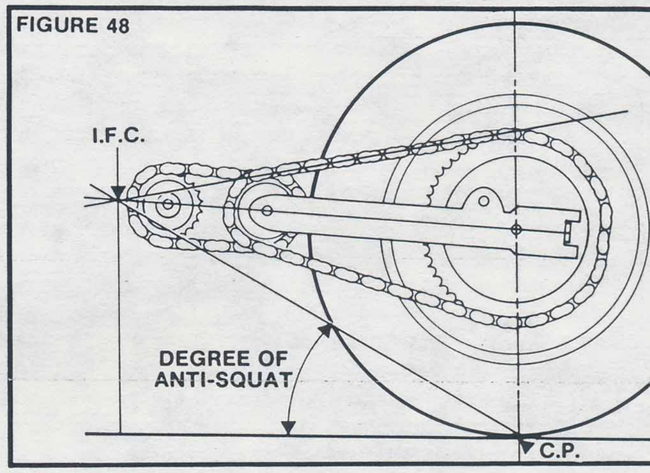


FIGURE 48



These gear-change effects can be useful for adjusting the anti-squat if you don't end up with gear ratios that are unusable. Some motorcycles have provision for altering the primary drive ratio (the gear ratio between the engine and transmission) to counteract any unfavorable final drive ratios. In any case, be aware that gear ratio changes also change the amount of anti-squat. Again, be aware that gear ratio changes don't solve the problem of pro-squat when the suspension is compressed.

Juggling the relationships of the swing-arm pivot and angle to the sprocket sizes and angle can bring the initial anti-squat to a more favorable amount and can reduce slightly the amount of anti-squat change due to suspension travel. But, in order to have any dramatic effect other methods are necessary. **Figure 48** diagrams an arrangement where the chain reacts on the chassis from a secondary sprocket mounted concentric to the swing-arm pivot. This configuration reduces the anti-squat change quite a bit but still falls short of the goal of no anti-squat change. Notice the only chain angle we are now concerned with is the second or final drive chain. This is because it is the one connecting, or acting as a link between, the sprung and unsprung parts of the chassis. For our anti-squat purposes, we will always consider the last fixed sprocket on the chassis as the point the chain is acting and the chain angle will be drawn through that point.

■ NOTE: For a topped-out suspension, or a rigid frame, none of the anti-squat forces discussed in these earlier paragraphs will apply.

The concentric sprocket design is the goal of most manufacturers today, as evidenced by all their efforts to position the countershaft sprocket and the swing-arm pivot as close together as possible. A few actually have made jack-shaft concentric arrangements, as shown in **Figure 48** and some are now designing extra long swing-arms that pivot right at the engine sprocket. Still, this design leaves something to be desired.

Figure 49A shows another approach to the problem. This time, two swing-arms are used in conjunction with a short upright to form a parallelogram that gives the effect of one very long swing-arm. If the arms are parallel to each other and exactly the same length, the effective swing-arm is the same as if there was just one central swing-arm. However, if the arms are not parallel **Figure 49B**, they assume an instantaneous center of their own and anti-squat diagrams must be drawn using this new instantaneous center as if it were the swing-arm pivot. This arrangement geometrically solves many problems like chain tension and braking torque, but still does not completely solve anti-squat change. Further, the double-link system pays a penalty in complication and extra weight.

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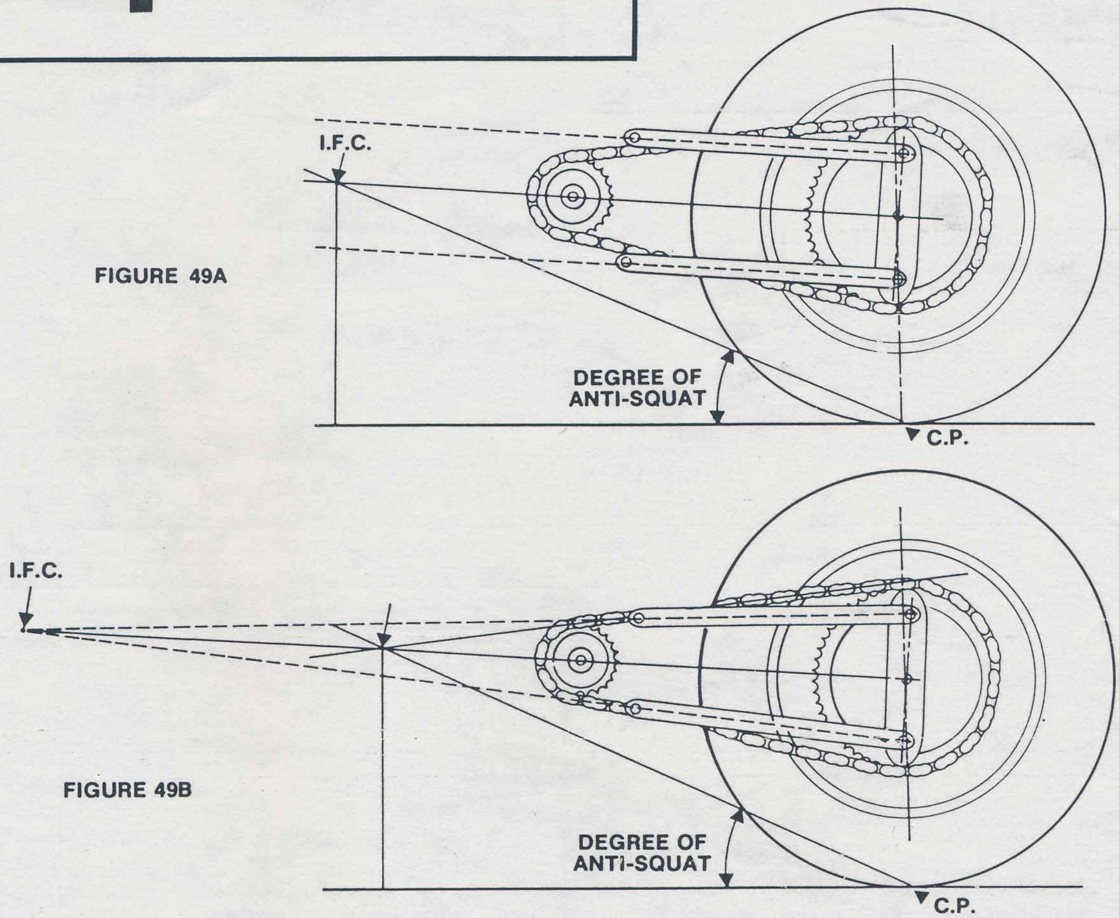
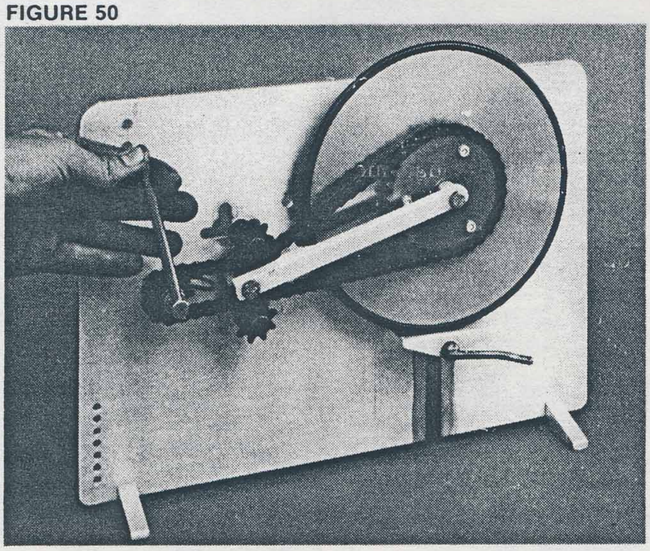


FIGURE 49A

FIGURE 49B

The most promising approach to anti-squat change is one developed by Trevor Harris. **Figure 50** is a photograph of a test fixture built by him to prove or disprove his theories. As you can see, it is a miniature motorcycle rear suspension with the positions of all the components completely adjustable. With this fixture, any combination can be tried and by measuring the torque required to rotate the engine sprocket, the traction between the tire and its support platform can be computed. This fixture confirmed beyond a doubt that anti-squat has a substantial effect on traction and the methods of diagramming anti-squat geometry presented here are accurate and realistic, not merely theoretical mumbo jumbo.

FIGURE 50



After determining the factors that influenced anti-squat, Trevor tried many variations in an attempt to minimize anti-squat change. The system with the best compromise of complication to effectiveness turned out to be a pair of idler sprockets that redirect the angle of the chain. **Figure 51** diagrams the new configuration both extended **(A)** and compressed **(B)**. Notice the top idler has no effect until the suspension is compressed. Notice the position fore and aft of the top idler changes its effectiveness. The farther back the greater the effect, but complications of chain length and swing-arm interference make installation more and more difficult on today's motorcycles. The bottom idler has two functions: One to improve the chain geometry during braking; the other to give up chain length as the top idler consumes chain length. This system does not solve the anti-squat change problem, but it does reduce anti-squat change more than any of the others. Additionally, it lends itself to an infinite number of variations which should provide the answers concerning anti-squat.

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FIGURE 51A

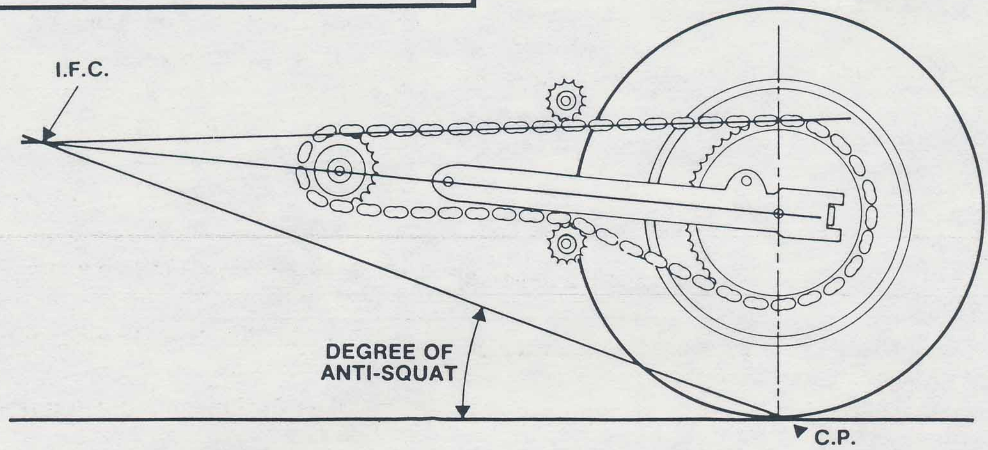


FIGURE 51B

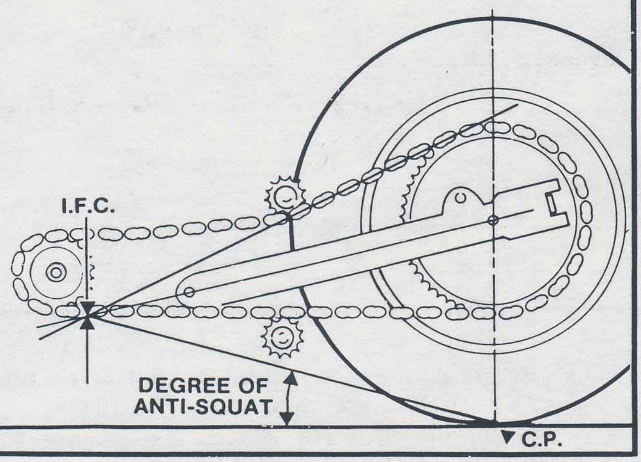
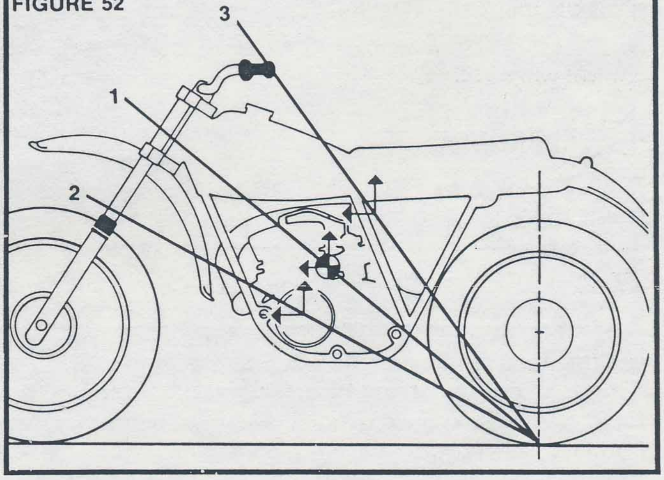


FIGURE 52



The chain idler system has been tested in competition in a flat track application with success that includes a win. The riders (one a former national champion) report a definite change immediately in the feel of the motorcycle when the idlers are installed. Further proof that anti-squat should not be ignored. These preliminary tests are just the tip of the development iceberg. Designs do exist that provide just about any anti-squat characteristics imaginable; however, the question of optimum anti-squat still haunts us.

Another look at the instantaneous force centers (I.F.C.) in the various diagrams reveals that as it moves up and down it almost always moves fore and aft as well.

There is another relationship to the I.F.C. that should be considered besides the anti-squat angle it prescribes. **Figure 52** shows a motorcycle with its center of gravity indicated. The center of gravity is the point where if you were to lift the motorcycle from that point, everything would be in equilibrium, or balanced. If you lift the motorcycle on either side of the center of gravity, the motorcycle will rotate until the center of gravity hangs directly under the lifting force.

In **Figure 52** three hypothetical optional I.F.C.'s are shown. How they differ is in their relationship to the center of gravity. The first option has the I.F.C. coincident with the center of gravity. This option will lift the chassis straight up. Option two has the I.F.C. ahead and below of the center of gravity. This configuration will tend to cause the chassis to rock clockwise for an instant. This rocking force gets more pronounced the greater the moment arm from the center of gravity. (The moment arm is a perpendicular line drawn from the anti-squat angle line to the center of gravity).

Option three will cause the chassis to rock in a counter-clockwise direction. The effective differences between options two and three is that the second option will tend to induce or promote wheelies whereas option three will tend to cancel wheelies for an instant. Notice also that any I.F.C., whether fore of aft of the center of gravity, whose anti-squat angle or force vector intersects the center of gravity will not impart a rocking motion to the chassis.

As is the case with the optimum anti-squat, the question of the optimum I.F.C. to center of gravity relationship is still very much unanswered. It is hoped that the information presented here will encourage further experimentation to resolve these questions. When that is done, we can all look forward to improved motorcycles and better motorcycling.

Glossary ...

ACCELERATION — The rate of change of velocity, or how quickly an object speeds up.

ACTIVE COILS — The working coils in a spring that are free to move.

AERATION — The condition when air bubbles are present throughout a liquid.

AIR SPRING — A device that uses air's natural ability to be compressed.

ANTI-DIVE — A system to resist a vehicle's nose-down pitch when braking.

ANTI-SQUAT — A system to resist a vehicle's nose-up pitch when accelerating.

BASE PIN — The restriction in a conventional shock absorber that gives compression damping.

BASE VALVE — The particular valve that gives damping in the compression direction, and free-flow in the rebound direction.

BLEED NOTCH — The small aperture in a shock valve that gives damping at slow piston speeds.

BODY ENGLISH — Instinctive use of body weight for control while riding.

BOTTOMING OUT — The condition when all suspension travel has been used up.

BOYLE'S LAW — A gas law stating that at a steady temperature, Pressure X Volume = a Constant.

CENTRE OF GRAVITY — The point in a mass where all the weights within the mass are equally balanced. Abbreviated to C.G. and shown on diagrams by the symbol

CHROME SILICON AND CHROME VANADIUM — Alloying elements in steel that give strength and toughness.

COIL BIND — The condition when a spring coil bears against the next coil.

COMPONENTS OF A FORCE — Parts of a force that act in specific directions.

COMPRESSION DAMPING — The resistance of a shock absorber to being pushed together at a given speed.

COMPRESSION RATIO — The ratio between the starting volume and the final volume when a piston moves in a cylinder.

CONTACT PATCH — The print of a tire on the ground.

DAMPING FORCE — The force required to move a shock absorber at any given speed.

DE CARBON — Name of leading authority on shock absorber design. Holder of many patents.

DECELERATION — The rate of slowing down of an object.

DEFLECTION — Movement of a part under load.

DIMINISHING RATE — Suspension where wheel rate decreases as the wheel moves up.

DIVE — Downward movement of the front of a vehicle (nose dive) caused by pitch.

DOWN FORCE — Total load at, for example, the tire contact patch.

DROOP — Suspension movement in the down direction. Wheel moves away from fender.

DYNAMIC — A system that's moving.

DYNAMOMETER — (Shock Dyno) A test machine for working a shock and measuring the loads it produces.

EMULSION — A dispersion of tiny air or gas bubbles throughout a liquid.

ENERGY — A body's ability to do work.

EXTENSION DAMPING — (Rebound or return damping) The resistance of a shock absorber to being pulled open at a given speed.

FADE — An unwanted reduction in damping control caused by heat.

FLOATING PISTON — Device to separate oil from gas in a De Carbon shock.

FORCE — An influence (as a push or pull) that causes motion or a change in motion.

FREE LENGTH — Natural length of a spring when no load is applied.

FREON CELL — Plastic bag with Freon gas trapped inside.

"g" — The acceleration due to gravity. (Or, what you feel in a banked turn.)

GEOMETRY — Descriptive term for the lengths and angles used in a chassis design.

G.V.W. — Gross Vehicle Weight. (For example, motorcycle, with fuel and oil, plus rider, passenger and luggage.)

HEAD ANGLE — Angle the steering axis leans back from vertical.

INSTANTANEOUS FORCE CENTRE — Location where component forces act on the chassis (I.F.C.)

LEVERAGE RATIO — (Of a rear suspension) Rear wheel travel divided by the shock travel.

(LEVERAGE RATIO)^2 — The leverage ratio multiplied by itself.

LOAD CELL — Device which produces an electric signal when loads are applied.

MASS — For motorcycles, think of mass as weight.

MEAN DIAMETER — The inside diameter of a spring plus one wire diameter.

MECHANICAL ADVANTAGE — (Of a rear suspension) Numerically, the same as leverage ratio

MECHANICAL PRELOAD — The amount either in pounds or inches, a spring is compressed when fitted to an extended shock absorber.

MOMENTUM — The momentum of a body is its mass multiplied by its velocity.

MUSIC WIRE — A good quality wire used for small springs.

ORIFICE — A passage, of exact and pre-determined size, for metering shock absorber fluid.

OSCILLOSCOPE — Electronic device for displaying information on a cathode ray tube.

Glossary ...

OVERSTRESS — The condition in a spring when the wire size cannot carry the loads applied.

PITCH — Rotation, or rocking of a vehicle in a vertical plane.

POLAR MOMENT OF INERTIA — A measure of the reluctance of a vehicle to rotate about its centre of gravity.

POTENTIOMETER — Device to sense movement and give an electric signal.

PRELOAD — See definitions of "mechanical" and "static" preload.

PRESSURE TUBE — The precision working cylinder in a conventional shock absorber.

PRIMARY SPRING — The long spring of a dual spring system.

PROGRESSIVE RATE SPRING — A one-piece spring with a rate that smoothly increases as the spring is compressed.

PUMPING DOWN — The condition (like a ratchet effect) when the ride height of a motorcycle progressively lowers over a series of bumps.

PUMPING UP — The condition when the ride height rises over a series of bumps.

REACTION — The push or pull that acts in opposition to (and is equal to) a force.

REBOUND — The extension or return direction of the shocks or suspension.

RISING RATE GEOMETRY — A suspension where the wheel rate rises with bump travel.

SEAL FRICTION — Mechanical drag of the seal on a rod or tube.

SECONDARY SPRING — The short spring of a dual spring system.

SHOCK ABSORBER — A hydraulic device used to resist movement or damp vibrations by pumping oil through orifices.

SPRING COMBINATION — Two (sometimes three) springs stacked together.

SPRING RATE — The amount of force required to deflect a spring a given distance.

STATIC PRELOAD — The mechanical preload plus the additional amount the spring compresses when it is supporting the chassis while at rest.

STATIC WEIGHT — Total of the weights at the front and rear tire contact patches with the motorcycle stationary.

TANGENT — Line touching an arc of a circle. (Example, a chain makes a tangent to a sprocket).

TORSION — A turning effect about a point.

TRAIL — Distance between the point where the steering axis intersects the ground and the centre of the contact patch.

UNSPRUNG MASS — Weight of the wheel, tire, brakes and suspension parts, etc.

VELOCITY — Speed.

WEIGHT TRANSFER — When braking, this is the amount of load subtracted from the rear wheel and added to the front.

WHEELBASE — The distance from front to rear axles.

WHEEL RATE — The amount of force required to deflect a wheel vertically by a given distance.

S & W Engineered Products wishes to thank **Motorcyclist Magazine** for their kind permission to reprint major sections of our Suspension Engineering Handbook which first appeared in **Motorcyclist** as a series in 1978. Their courtesy and cooperation was of tremendous value to our staff.

AUTHOR: BRUCE BURNESS

CONSULTANT ENGINEER/DESIGNER: MARTIN WAIDE

LAY-OUT AND ILLUSTRATIONS BY ROGER SAURIOL

Spring rate Selection

The use of the table and chart on this page will give an estimated rear suspension spring rate for all types of riding. Several variables have to be considered, and turned into a "correction factor" by using the small table (below, at right) which shows both off-road and street riding parameters. Total up the appropriate correction factors (plus cancels minus) and apply this factor to a particular box at the top of the main chart, which carries the description of your type of riding.

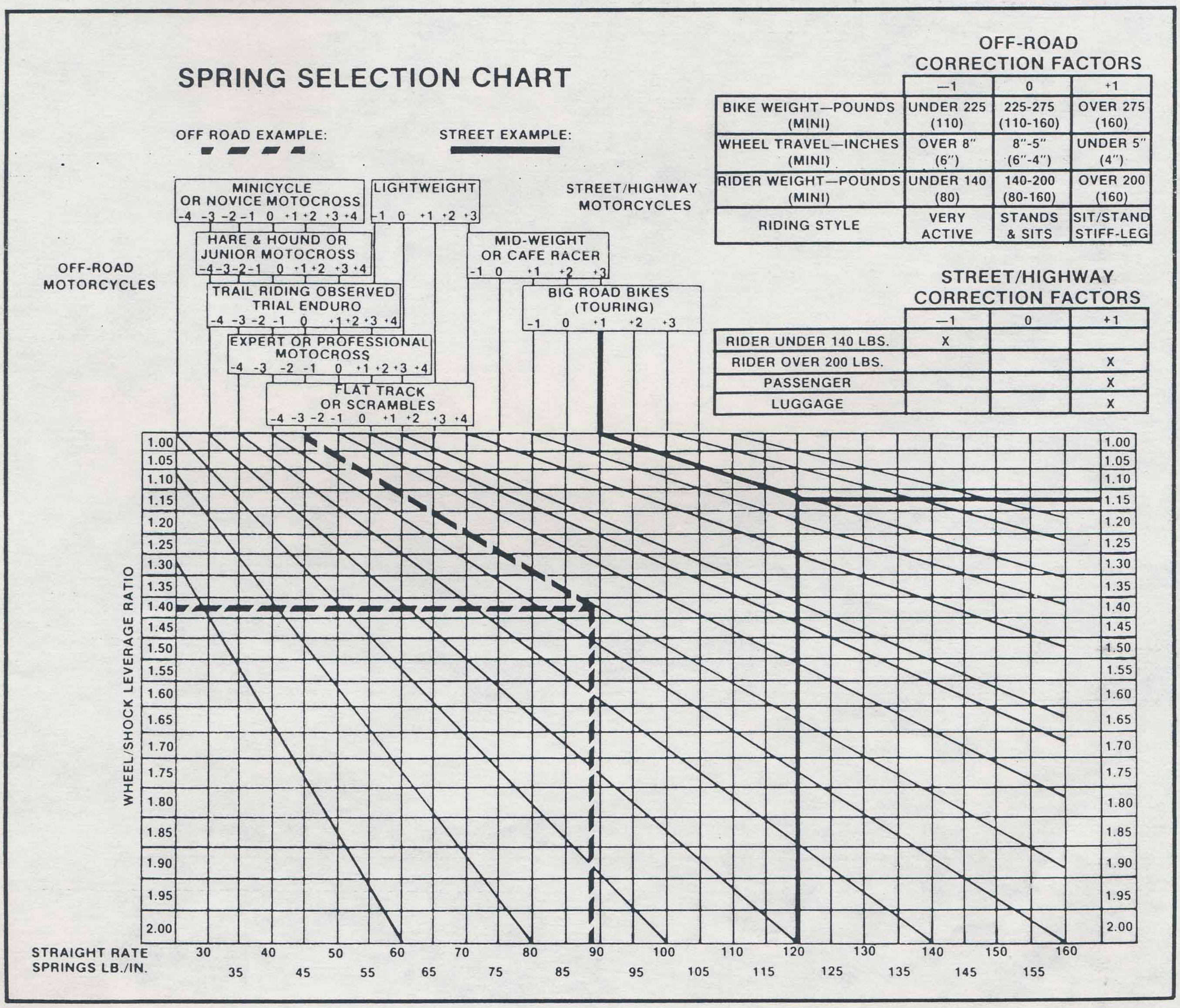
You will note that the leverage ratio of the suspension must be known. Draw a line horizontally across the chart for your particular leverage ratio. Refer to Page 9 for a method of obtaining this ratio for your bike's measurements if you don't already know it.

Follow the angled line down from your selected correction factor at the top of the chart until it crosses your horizontal leverage ratio line. Then follow vertically downwards to the spring rate scale, and read off the spring rate in lbs./in.

The resulting spring rate is a best estimate and your personal preference may eventually be for a rate slightly different from the one recommended. Here are two actual examples in the use of the table and chart.

Off-Road Example: First refer to the small table. A 280 lb. Bultaco (+1) with 8 inches of rear wheel travel (—1) has a 140 lb. rider (—1) with a riding style standing and sitting (0). Totaling the correction factors, +1, —1, —1, and 0, the result is —1. Transfer this factor to the main chart, and knowing that the rider is an expert at motocross, and that the Bultaco has a leverage ratio of 1.41:1, the chart shows a recommended Spring rate of 88 lb./in.

Street Example: Again refer to the small table. Take a Honda GL-1000 being ridden by a rider of over 200 lbs. (+1) but with no passenger (0) and no luggage (0). The resulting correction factor is +1, +0, +0, = +1. Transfer this to the main chart and knowing that the GL-1000 is a big road bike with a leverage ratio of 1.15, the recommended spring is 120 lb./in.



OFF-ROAD CORRECTION FACTORS

	—1	0	+1
BIKE WEIGHT—POUNDS (MINI)	UNDER 225 (110)	225-275 (110-160)	OVER 275 (160)
WHEEL TRAVEL—INCHES (MINI)	OVER 8" (6")	8"-5" (6"-4")	UNDER 5" (4")
RIDER WEIGHT—POUNDS (MINI)	UNDER 140 (80)	140-200 (80-160)	OVER 200 (160)
RIDING STYLE	VERY ACTIVE	STANDS & SITS	SIT/STAND STIFF-LEG

STREET/HIGHWAY CORRECTION FACTORS

	—1	0	+1
RIDER UNDER 140 LBS.	X		
RIDER OVER 200 LBS.			X
PASSENGER			X
LUGGAGE			X

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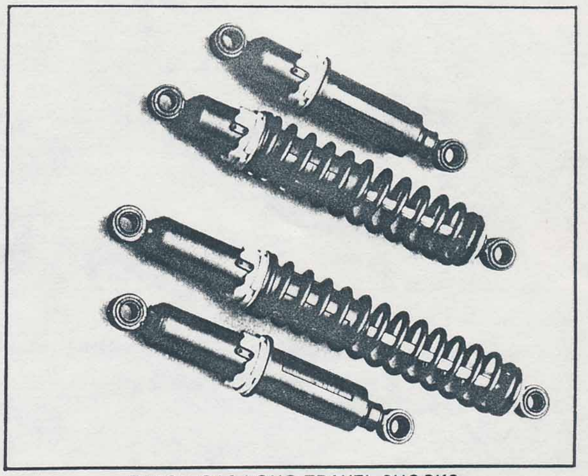
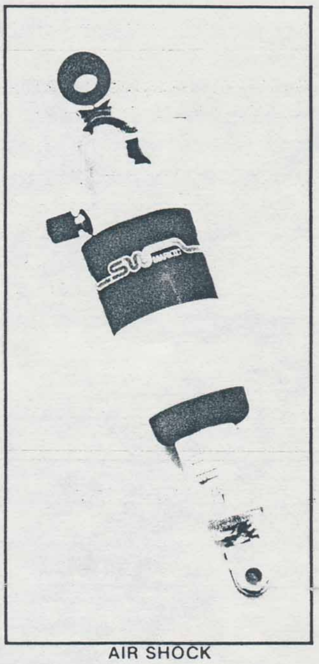
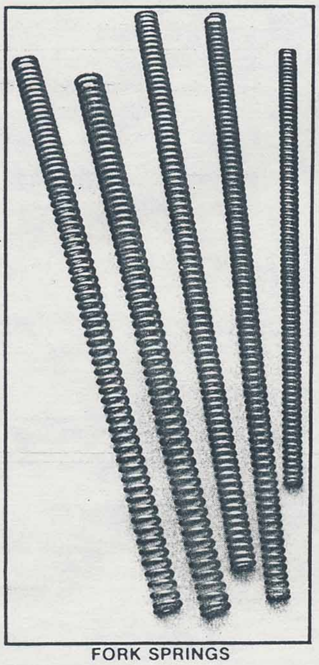


FREON GAS LONG TRAVEL SHOCKS

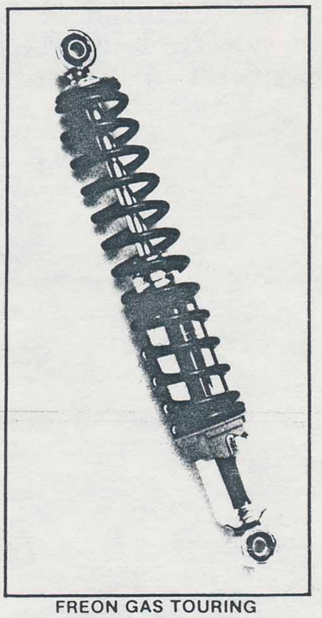


AIR SHOCK

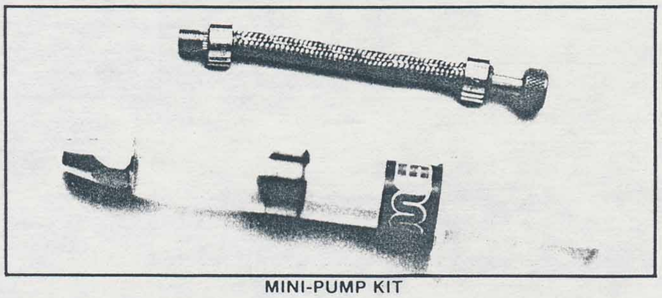


FORK SPRINGS

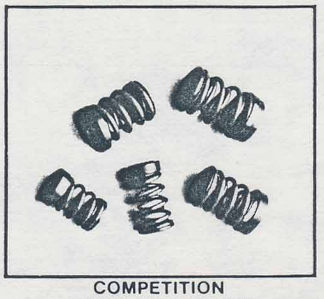
MOTOCROSS
DUAL SPRING SHOCK



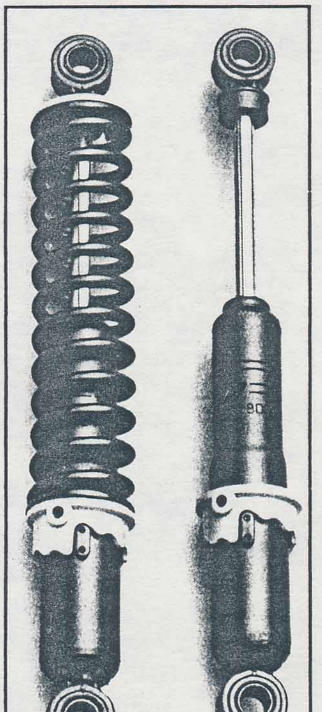
FREON GAS TOURING
SHOCK (CHROME)



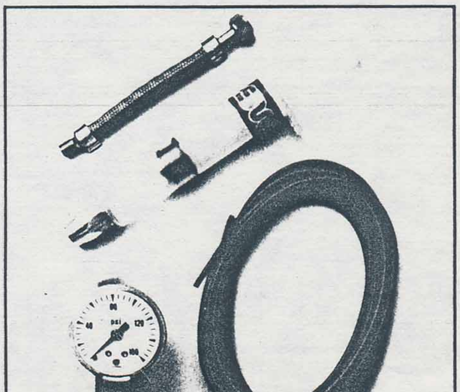
MINI-PUMP KIT



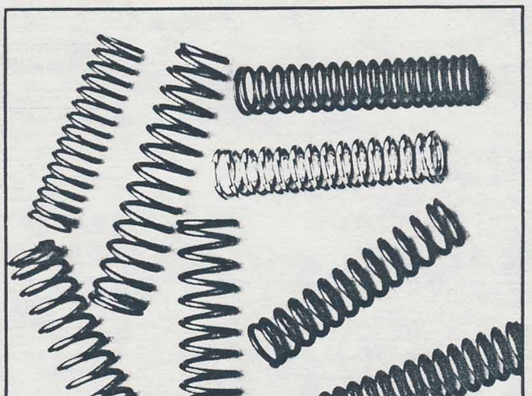
COMPETITION
VALVE SPRING KITS



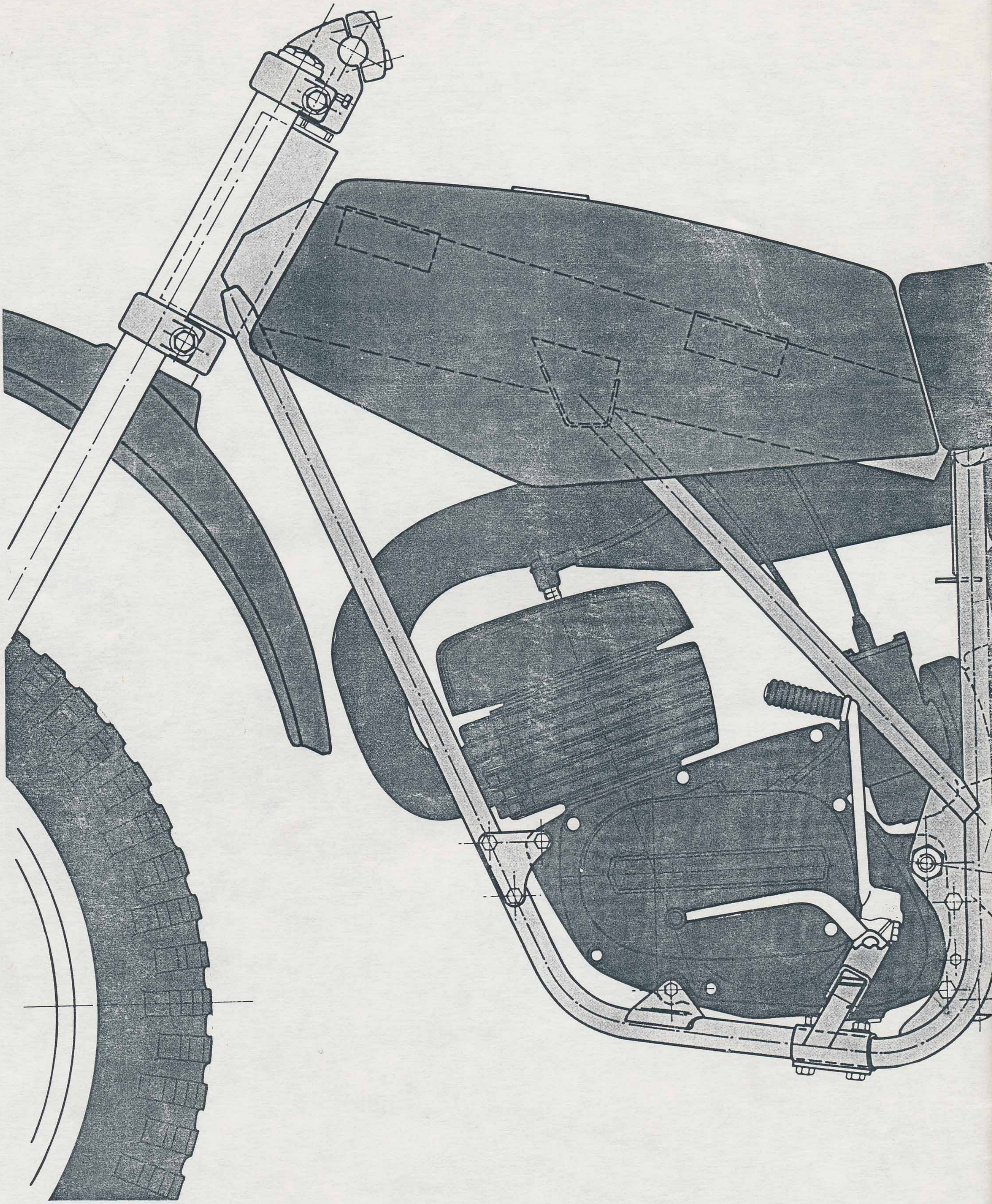
STANDARD
HYDRAULIC SHOCKS



PUMP AND GAUGE KIT
FOR AIR SHOCK



SHOCK SPRINGS



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