MOPAR SUSPENSIONS

S-A DESIGN

COMPLETE GUIDE FOR BUILDING MOPAR SUSPENSIONS FOR STREET, AUTOCROSS, AND ROAD RACING

- SUSPENSION BASICS • CUSTOM SUSPENSION COMPONENTS
- UNDERSTANDING TERMINOLOGY • THE BRAKING SYSTEM
- CHOOSING TIRES & WHEELS • TORSION BAR TECHNOLOGY

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MOPAR SUSPENSIONS

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MOPAR SUSPENSIONS INTRODUCTION

TO THE READER

This book will help you understand the dynamics of vehicle handling and—even more important—show you how to apply these theories to Chrysler torsion bar suspension systems. In a nutshell, it will show you how to make *your* Mopar handle—handle nearly as well as the world's finest road cars—without spending a fortune!

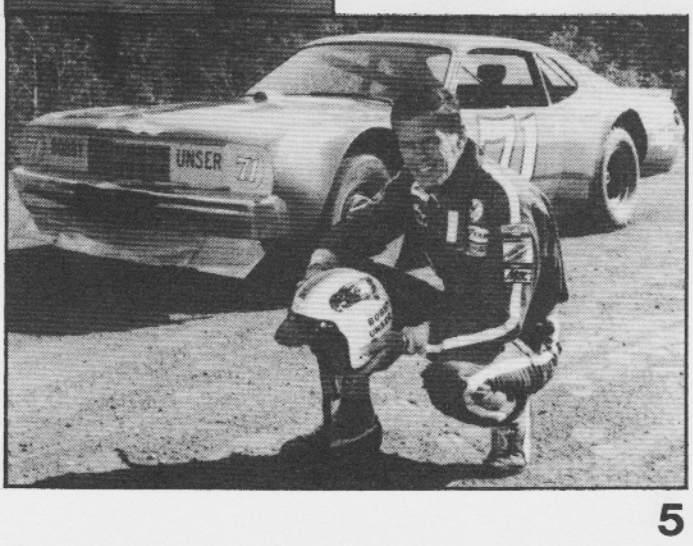
While the sports car magazines extoll the virtues of a well handling car, many assume that their readers drive as well as their test drivers. This writer has made no such assumptions. If you are only familiar with boulevard cruising or drag racing, you will have a lot to learn or (re-learn) about driving and controlling an automobile with a superior road suspension. For those of you who have never heard the term "autocross" or "slalom," I suggest that you locate a local sports car club and attend with the sole purpose of learning. The best way to find out how you rank as a high-performance driver is to participate. Your ego will surely get bruised, but with the correct outlook, you may discover why an apparently tame looking VW went through the course a lot faster than you. I know this to be true, because it happened to me and began my understanding of how to make my Mopar handle.

For those of you who own Mopars and are interested in improving handling, but can't find solid, understandable information, this book may become your "bible." If you would like to build a well-handling "grocery getter" or a flat-out competition car for road racing, this book will help you do it right. And if you wish to delve even deeper into the science of handling, a list of additional publications is included that will guide you beyond the techniques and tricks revealed in the following pages.

INTRODUCTION

The science of handling is full of conflicting theories—as are most areas of automotive performance. Suspension systems are particularly open to controversy since each driver develops his or her own style and preference for suspension response. Some drivers may prefer a firm ride, others may choose a softer ride; and both may run identical lap times on the same course. This is not uncommon but is remarkable since the suspension adjustments may be entirely different. This book will introduce a base line for suspension design and then show you how to "tune" your chassis and suspension to suit *your* driving requirements.

The biggest controversy surrounding handling theory for production based cars (sedans, sports cars, etc) involves spring rates versus anti-sway-bar rates. One "school" prefers stiff spring rates and weak anti-sway-bar rates. The other prefers just the opposite: stiff anti-sway-bar rates and slightly stiffer than stock spring rates. Neither side will budge on their beliefs, but you can surmise that the track surface and racing conditions dictate which system to use. For example, a vehicle traveling "at speed" on a high-banked oval would need much stiffer spring rates and less stiff anti-sway-bar rates in order to maintain good stability and control, whereas a vehicle on a back road or road-racing course would need



Suspension design is quite different for dirt tracks, off-road rallies, and hill climbs (Bobby Unser is shown here at Pike's Peak). For optimum performance, the suspension design must be carefully matched to the road/track surface conditions, car weight, intended use, etc.

Road racing more closely approximates street driving than any other automotive sport. Although most of the exotic "race-only" cars are not practical for street driving, the basic suspension design and safety equipment added to these vehicles can turn a mundane "grocery getter" into a safer, better-handling G/T machine.

stiffer anti-sway-bar rates and softer spring rates to prevent it from "bouncing" off the road.

When referring to a "well-handling automobile," all parties agree on certain "constants" that give the feeling of well-handling. These constants are: directional stability, cornering ability, positive road feedback, aerodynamic stability, predictability, security, and reliability. The question is: how does one go about acquiring these characteristics? There are two ways. The first is to buy the vehicle already built to these characteristics, such as a top-line Porsche or BMW. The second approach—and the one we will address in this book—is to address each of these desirable characteristics and build them into an existing chassis.

An important fact about handling is: Not all cars are created equal. A car with leaf springs at all four corners and a high center of gravity—such as many four-wheel-drive vehicles—will never handle as well (on pavement) as a car with a low center of gravity that utilizes a combination of leaf springs and torsion bars or coil springs. Therefore, if you want your D-100 to handle as well as an early Dart, Barracuda, or late-model Shelby Charger, you'll have to redesign the entire suspension system.

Fortunately, Mopars with torsion bar suspensions can be modified relatively easily to handle much better than they presently do, including station wa-

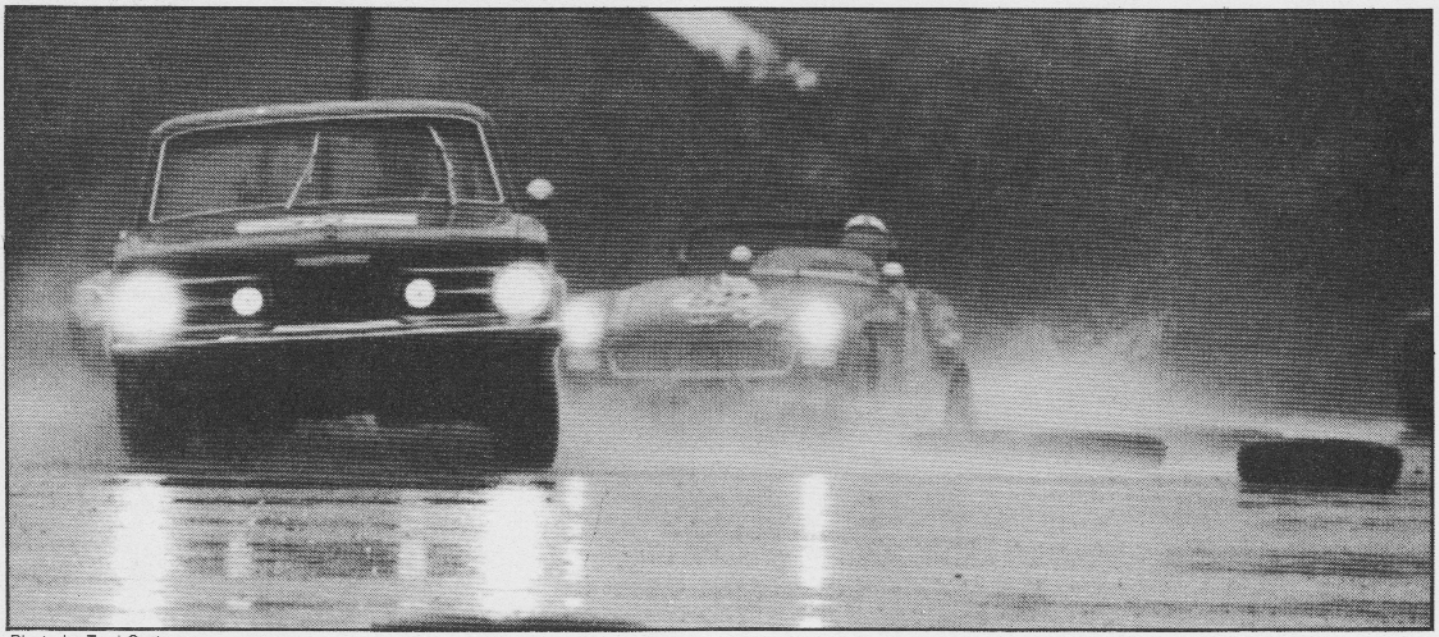
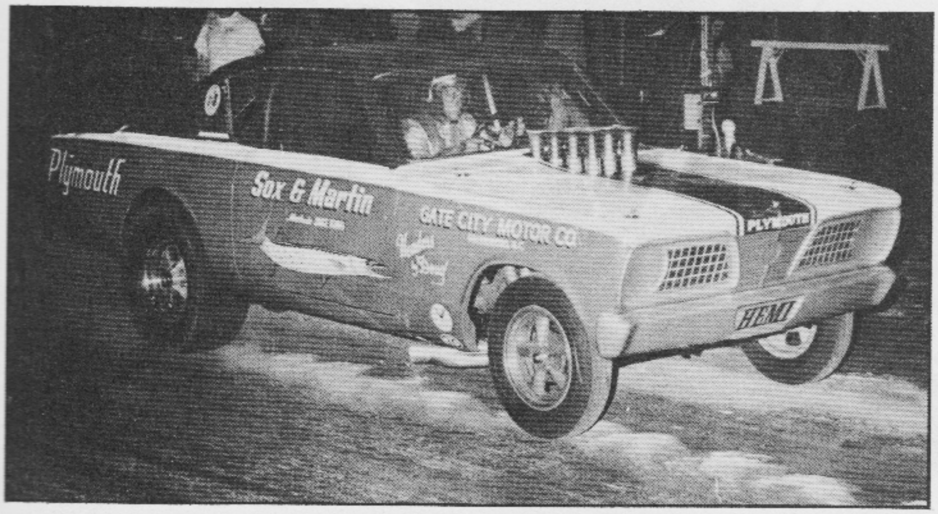


Photo by Toni Cortes

Mopars can be easily modified to improve handling on the street or track. The straightforward torsion-bar front suspension and the center-offset rear leaf springs that Mopars have pioneered are relatively easy to work on and tune for performance. A careful evaluation of requirements, applying the proven principals discussed in this book, and always keeping saftey in mind, will produce a suspension system that gives predictable response to driver input, even in adverse road and weather conditions.



While this may bring tears to the eyes of a 1/4-mile drag buff, it is not the way to build a car for cornering. The CGH (Center of Gravity Height), engine position, and front shock dampening are designed to shift weight to the rear; the front tires are very narrow for friction reduction; and ballast has been added to the rear end for traction. All of these modifications will improve straight-line acceleration *and* reduce cornering ability.

gons. The short and light "A" body chassis with an "A" engine (not the heavy "B" block or the venerable Hemi) under the hood is an excellent starting point. (Note: We will address the front-wheel drive cars in a separate chapter.)

While there isn't any direct correlation between handling and wheelbase or track (track is the distance between the centerlines of the right and left tires), the majority of well-handling vehicles fall into the region of 95- to 106-inch wheelbases with a 54- to 60-inch track. This doesn't mean that if your car has a 112-inch wheelbase it won't handle. It just means that with the emphasis on downsizing, most cars fall into a shorter category. There is a definite correlation, however, between handling and the amount of weight on the front wheels. Too much weight on the front end reduces the ability of the car to respond well to steering inputs; too much weight on the rear end will induce oversteer. So if you are thinking of building a super suspension under your blown Hemi Dart or 'Cuda and "attacking" the Turbo-Carrera down the street—don't!

For purposes of handling, the following rule of thumb is provided: "A"-Body cars will handle best with "A" and "LA" engines; "B"-Body cars with "B" and "RB" engines; and "E"-Body cars with "A" and "LA" engines. If you must build that blown Hemi, put it in an Imperial, Superbird or Daytona; that'll turn some eyes, and it should handle too!

Here is a well-built road-racing engine—the 355-cubic-inch version of the venerable 340 Mopar. The lack of exterior "flash" belies the internal modifications that make this a reliable, high-output endurance engine, rather than a 1/4-mile sprinter.

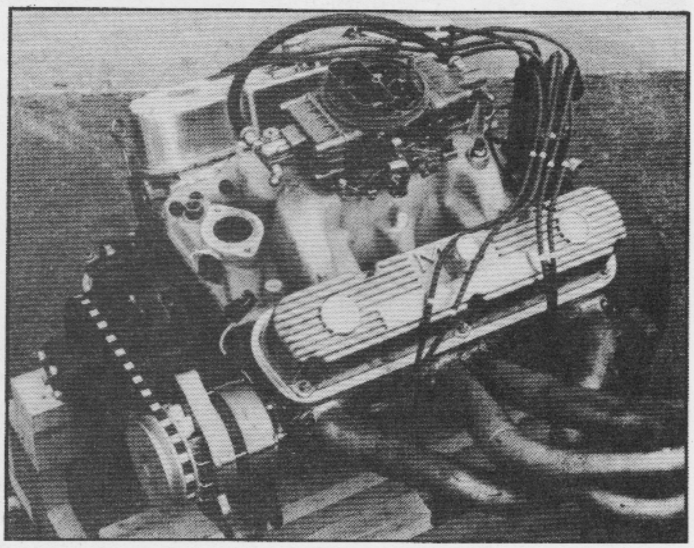
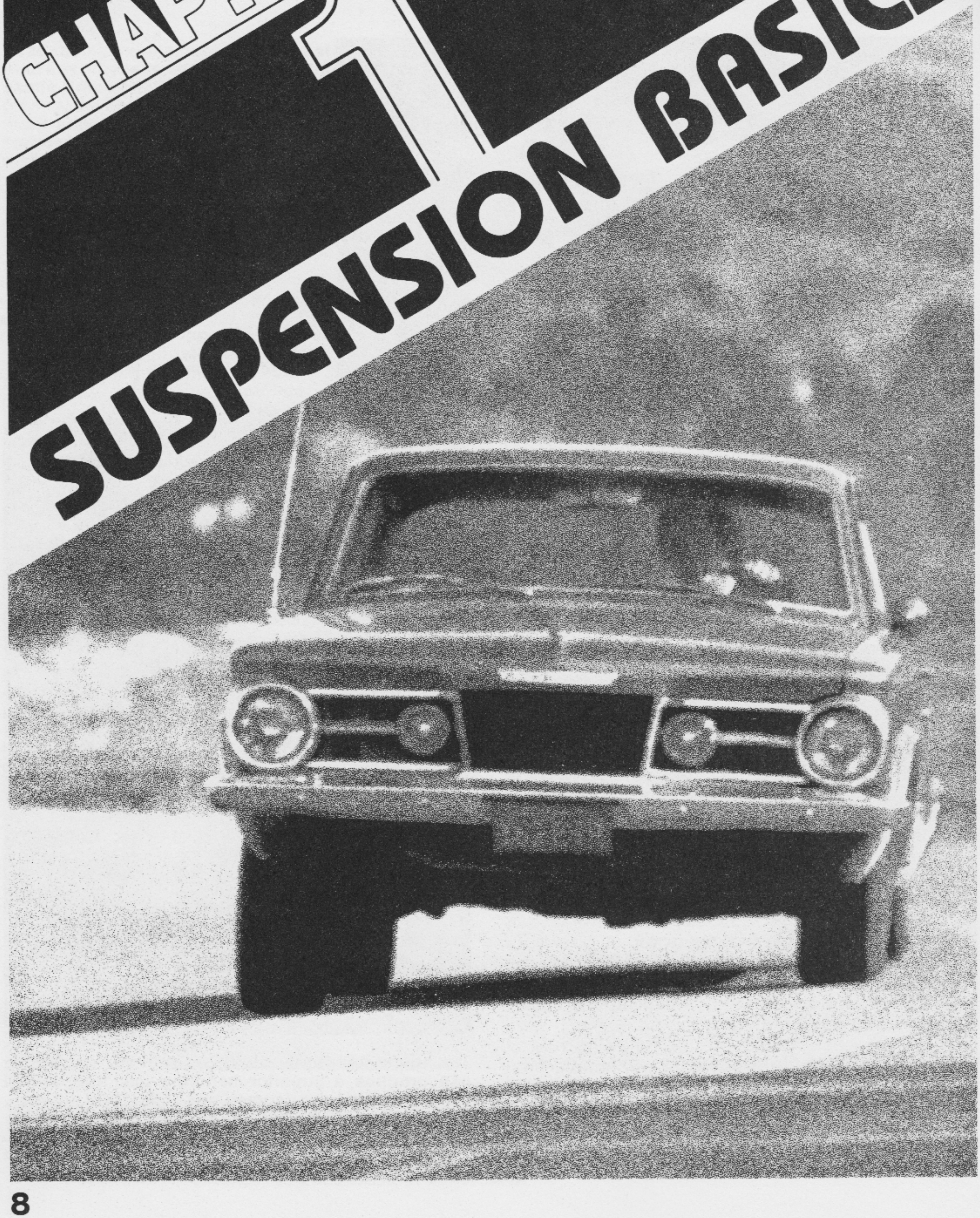


Photo by Toni Cortes

The Mopar "A" Body, combined with an "A" engine, can produce a superior handling car. The overall vehicle weight is low (sometimes under 3000 pounds), and the lightweight "A" engine keeps weight distribution close to 50/50 (considered ideal). Also, there are plenty of parts available in junk yards and from Chrysler's Direct Connection parts program.

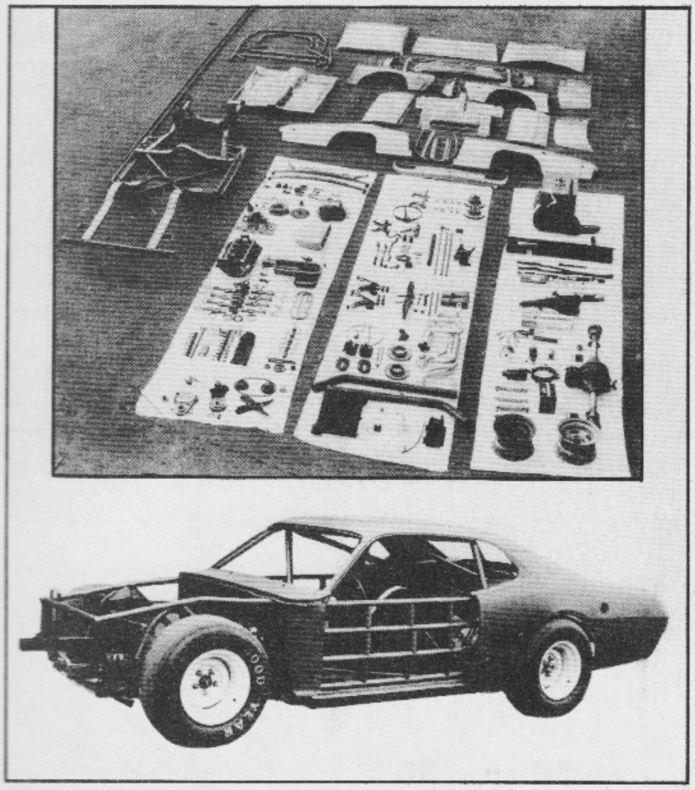
CHAPTER 1
SUSPENSION BASICS



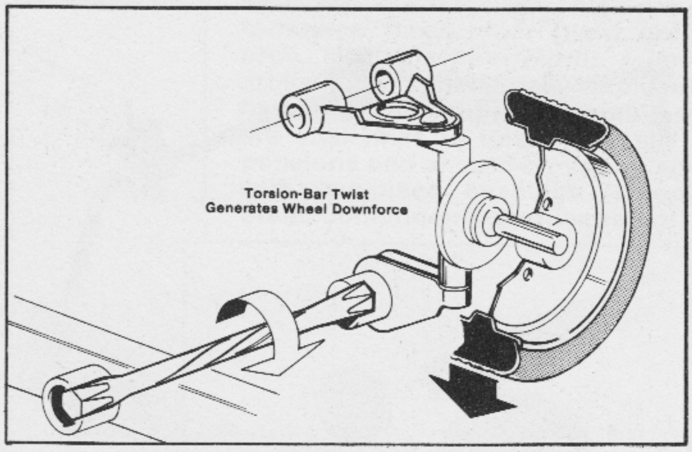
SUSPENSION BASICS

The torsion bar suspension on all Chrysler products is very simple: The front suspension is independently sprung with torsion bars and telescoping shock absorbers are used with an anti-sway bar (optional on some models). On the rear, a Hotchkiss-type differential is sprung by unequal-length leaf springs (unequal length refers to the mounting of the differential closer to the front spring eye, rather than in the center of the leaf), telescoping shock absorbers and, on some vehicles, an optional rear anti-sway bar. All of these parts are attached to a strong unit-body chassis.

While the standard Chrysler suspension system doesn't seem as impressive as a totally independent design, it does the job—and does it well. Although an exotic suspension design may seem superior to all others, no design is really superior over "all" others. A notable exception to this is the front and rear leaf-sprung suspension, such as those found on four-wheel-drive vehicles. While these heavy-duty vehicles can be made to handle reasonable well for certain applications (dirt track, off-road), they are generally inferior for high-performance handling, as discussed in this book. Pick-up trucks can be made to handle, however, but not without sacrificing their usefulness as a utility vehicle.



Here are the critical parts that work together to make a high-performance vehicle move, handle, stop, and protect the occupant(s). An automobile ranks among the most complicated and sophisticated mechanical devices in the world; but the sum of years of design and development have produced a vehicle that is both reliable and fun to drive.



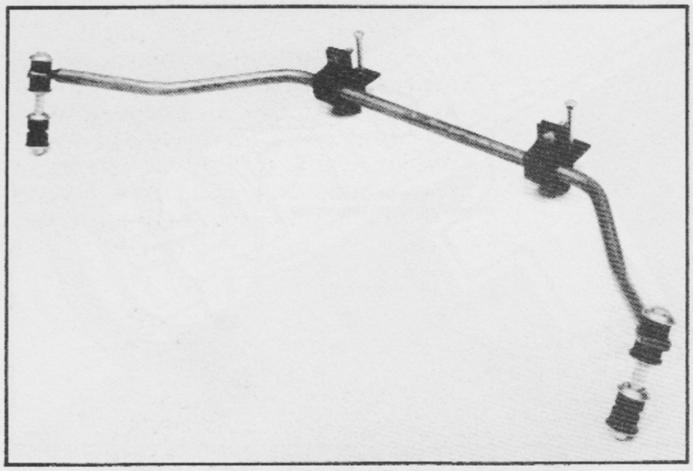
Chrysler torsion-bar suspension is a simple, easy-to-work on design. One end of the bar is attached to the chassis; the other end is connected to the lower control-arm pivot. When the bar is placed under torsional load (twist), it generates a downforce at the wheel, counterbalancing the front-end weight.

The basic components of the Chrysler suspension (springs, torsion bars, etc.) are certainly important in effecting good handling, but there are other, less-obvious parts that are equally vital to overall function. Suspension bushings, shackles, mounting brackets/bolts, ball joints, and rod ends are such pieces. They all work together to make the car handle. Worn bushings and/or ball joints will sabotage the best suspensions, so plan to use all new parts. You wouldn't use year-old oil in a new engine; don't make a similar kind of mistake with your suspension.

Every suspension build-up should include a re-evaluation of the braking system. Capable brakes are as much a part of good handling as high-quality oil is to a race engine. The same can be said for tires and wheels. All three sub-systems (tires/wheels, brakes, and suspension) must thoroughly meld together or handling will be less than optimum. Each of these sub-systems will be detailed in the following chapters. But for now, some more basics.

Suspension bushings prevent shock and vibration (induced by irregularities in the road) from being transmitted to the chassis and the passengers. The shock absorption of rubber bushings will also reduce component failure. While some race cars use solid bushings, or bushings made of extremely hard rubber, this is not recommended for street-driven machines. The days of "rock-hard" suspensions and a masochistic driving philosophy are over. Excellent handling results can be achieved without shaking loose your dental work.

Springs are designed to make a vehicle ride smoothly over road irregularities. The spring flexes so that the car won't. And rear leaf springs perform an additional function; they locate the differential and maintain proper track. But, to keep the car from bouncing and re-bounding uncontrollably, motion dampeners are required to reduce the continued



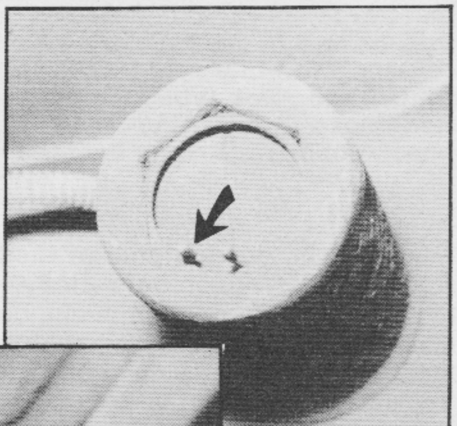
The front anti-sway bar is nothing more than a torsion bar that connects the right- and left-front suspension elements. As the body begins to lean in a turn, the sway bar resists the lean—or "roll"—and applies a counterforce to minimize this tendency.

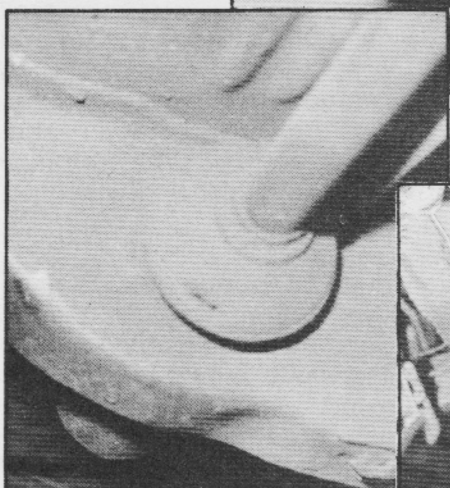
oscillation inherent in springs. These dampeners are commonly called shock absorbers, even though they don't actually absorb shock. What they do, technically, is dampen the continuing oscillations of the springs by converting the rebound energy into heat.

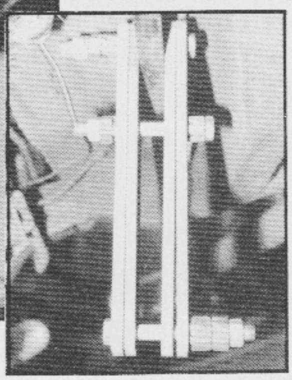
The final major suspension component is the anti-sway bar. It is called by various names, such as "sway bar," "roll bar," or "anti-roll bar." By any name, it provides an essential function in high-performance handling by preventing the car body from leaning precariously. The anti-sway bar is very similar to the torsion bar in design and function. In this case, however, the "torsioning" (twist resistance) of the bar not only provides resistance to inhibit body lean, but also plays a major role in keeping the front wheels on the ground as the tire moves over irregularities in the road surface.

The reader should remember that every time a component of the suspension is changed (hopefully improved), the handling characteristics of the car will change. Experienced drivers will take time to get acquainted with the new tendencies/peculiarities of the car before trying any "all-out" driving techniques. A typical example is the addition of wider wheels and tires. If no other changes are made to the suspension, alterations in handling response—particularly in steering feedback—will be pronounced (exactly what changes will be noted depend greatly on the design of the new tires and wheels).

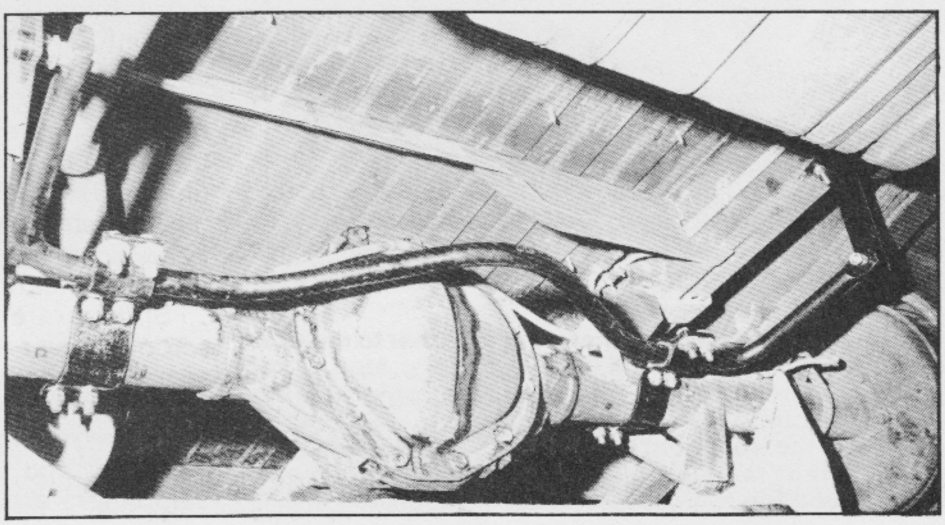
Most people will find that using bolt-on equipment, such as stiffer shocks, front and rear sway bars, etc., will improve handling. This philosophy is perpetuated by many of the aftermarket suppliers. *However, while simple bolt-ons will improve the handling of the vehicle, they are not the complete story (by a long shot), especially if you wish to be competitive with well-prepared cars.* You may wish to look at one of the complete suspension kits sold for Datsun Z cars,



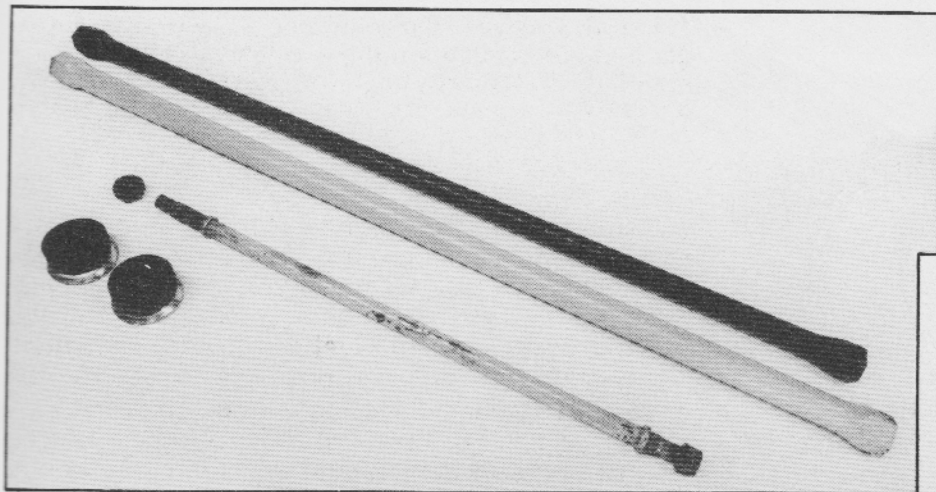


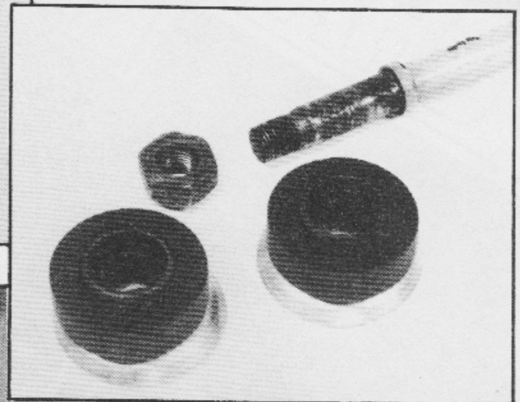


This is the torsion-bar retaining socket located on the chassis. By removing all torsional load (completely unscrewing the adjustor located in the lower-control arm), and extracting the small spring clip (arrow), the torsion bar can be removed toward the rear of the car. Bar removal may require attaching a clamp onto the bar (Note: do not scratch or damage the bar; fatigue cracks can develop from surface irregularities).

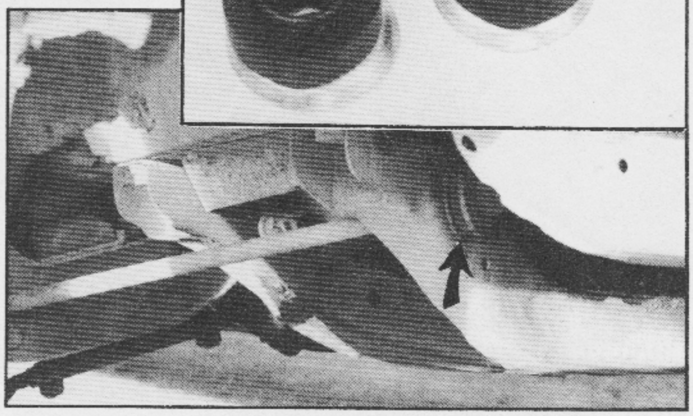


The massive rear-sway bar illustrated here is just plain "over-kill." A bar this large will almost never be required on a van, truck, or automobile.





Bars and bushings: the high-performance torsion bar (light colored, available from Martin Automotive) can substantially improve handling. The strut-rod assemblies are vital links in torsion-bar suspensions and should be upgraded with two-piece bushings (Moog brand components are illustrated).



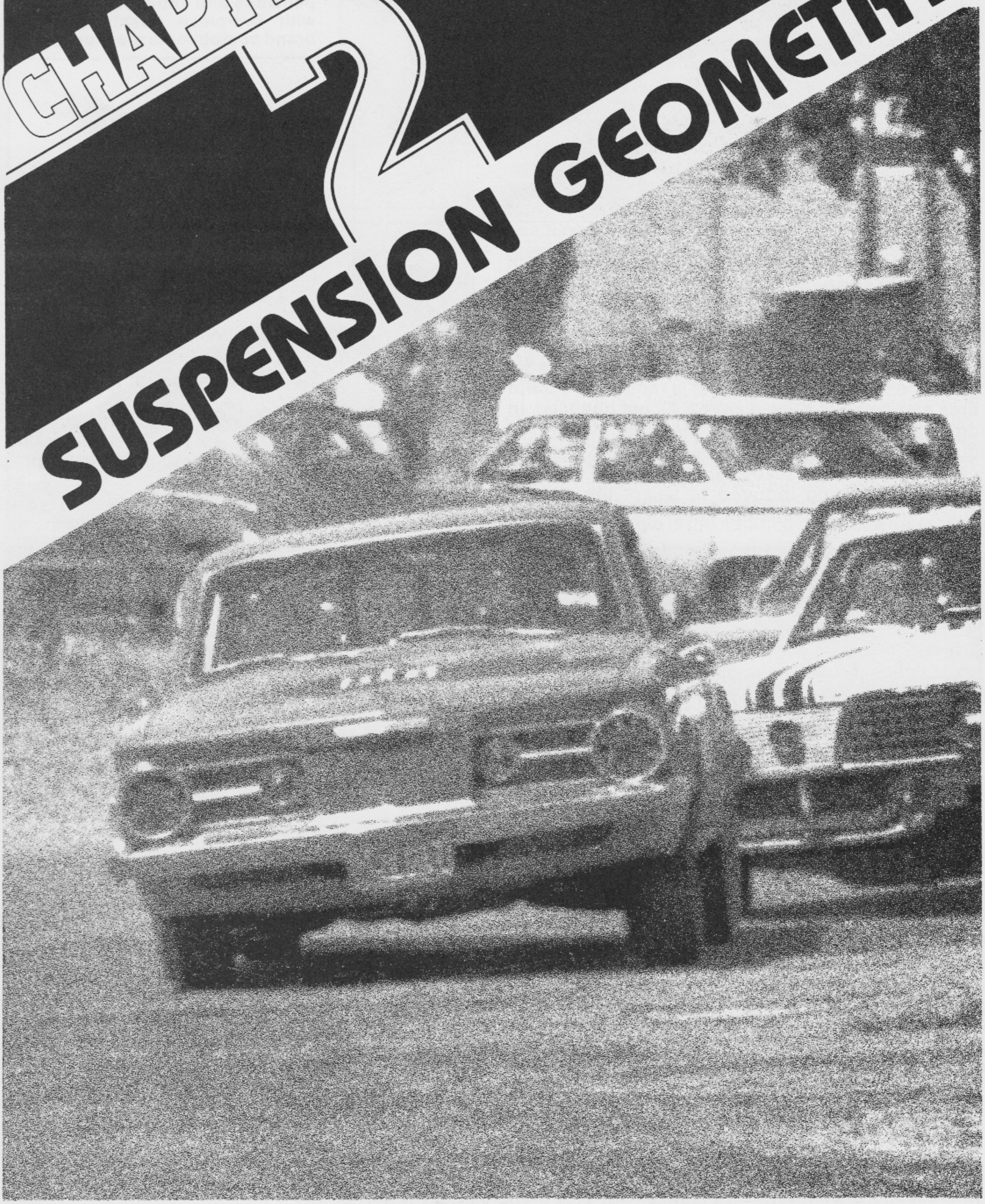
as it illustrates what a *complete* suspension modification looks like. Besides the front and rear anti-sway bars, the kit includes three levels of springs (from heavy-duty street rates to all out competition), all new bushings (including steering bushings), and a tension/compression kit for the strut link. This kit gives the buyer virtually all that is required—except tires and rims—to build a car for all out handling. In order to apply this "package" theory to Mopar products, we must explain a few technical facts. A careful study of the following material will give you the understanding required to select the correct suspension pieces...the first time!

Photo by Toni Cortes

The halcyon days of sedan racing mixed stiff competition with a feeling of low pressure; the high-tech environment had yet to develop. But racing teams were soon concerned with the intricacies of suspension design, and large sums of money were spent in an effort to best the competition. In a short time, racing development leapfrogged ahead to relegate the earlier "run-what-you-brung" days to the past. But because Mopar torsion-bar suspension systems are easily modified for race-track-like cornering, many street machines now capture the enthusiasm of these early seat-of-the-pants racers.

CHAPTER 2

SUSPENSION GEOMETRY



A well-set-up Mopar can generate high lateral acceleration, exhibit excellent handling stability, and generate very little body lean.

Photo by Toni Cortes

LATERAL ACCELERATION

A discussion of vehicle handling is not complete without reference to "lateral acceleration." This is a measure of the side forces applied to a vehicle in a turn. Lateral acceleration is often used to judge how well a vehicle handles (just as 0-to-60 mph acceleration times are used to judge straight-line acceleration). Cars that have 0-to-60 times under 10 seconds are considered fast, and those that generate lateral accelerations of .75g (75% the force of gravity) or more are considered "good handlers." The higher the lateral acceleration, the faster the car will negotiate a turn. Most sports car magazine road test summaries include the maximum measured lateral acceleration (G) along with the 0-to-60 and quarter-mile elapsed times.

Now you may ask, how is lateral acceleration determined and how do I put the G-value of my Mopar into the respectable category? To start off, let's look at the *measurement* of lateral acceleration, since for the most part, the rest of this book is devoted to ways of improving lateral G and handling.

The first "essential" in G measurement is a large, flat, paved area (a parking lot is ideal), used for what is called the "skid-pad" test. This area must be paved (absolutely), flat, and free of obstructions such as curbs, trees, other cars, pedestrians, etc. A circle is marked out on the pavement, usually 200 feet in diameter (although a larger—400 or 600 foot—test circle will yield more accurate results, the larger size may not be practical). The car is then driven on the circle while the elapsed time for each lap is carefully measured. The car is driven faster and faster until it slides out of the circle. Acceleration must be applied smoothly, as heavy throttle inputs will cause the car to prematurely slide out of the circle. After the first test, the vehicle is sometimes driven in the opposite direction to verify that it can generate the same lateral acceleration in both left and right turns. After the fastest lap-time is recorded, the lateral acceleration can be calculated. First the circumference of the test circle is found by using the formula:

FORMULA 1:

Circumference = 2 x (Pi) x Radius

or

Circumference = 2 x 3.14 x 100 = 628 feet

An industrial lot is ideal for a skid pad test; it is large, relatively flat, and free from obstructions and traffic. The vehicle is driven around a fixed circle until it slides out (by either oversteering or understeering). The fastest lap time is then used to calculate the maximum lateral acceleration.

Skid-pad results establish the baseline for suspension design. But it is on-track testing, evaluation and modification that fine tune the suspension to obtain that competitive, winning edge.

(Where 100 is the radius of a 200-foot diameter circle). The answer of 628 feet is the distance that the car travels as it makes one "lap" around the circle.

Speed can be determined by simply dividing distance by time (an old "trick" in basic physics):

FORMULA 2:

$$\textbf{Speed} = \frac{\textbf{Distance}}{\textbf{Time}}$$

For our application the distance is the circumference of the skid-pad circle, so we can substitute the circumference formula (formula 1) for "distance" in the above equation. This yields:

FORMULA 3:

$$\textbf{Speed} = \frac{\textbf{2 x (Pi) x Radius}}{\textbf{Time}}$$

Now, let's set up one more substitution. The next formula (again from basic physics) will find the acceleration for an object moving in a fixed circle when the speed of the object and the size of the circle are known:

FORMULA 4:

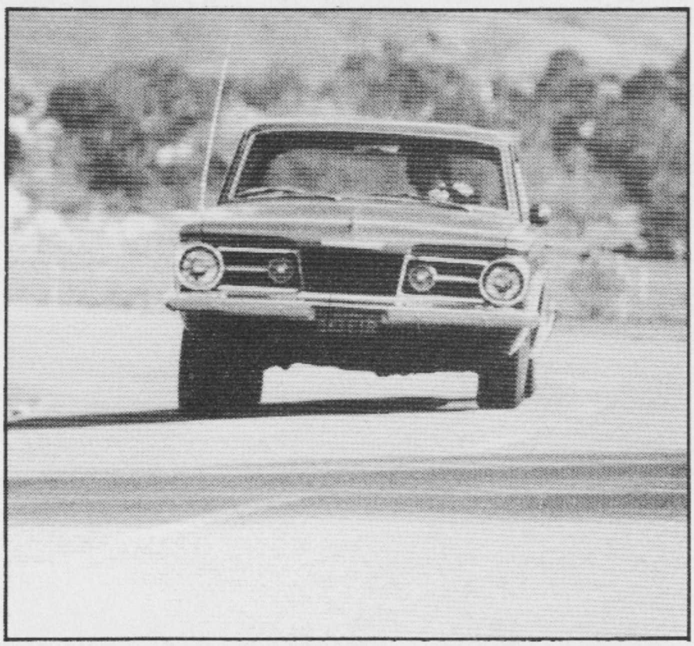
$$\textbf{Acceleration} = \frac{\textbf{Speed}^2}{\textbf{Radius}}$$

But since we know the time of each lap (by using a stopwatch) rather than the speed of the car, substituting formula 3 for "speed" in the above equation will produce a formula for calculating acceleration that uses *the data we have measured*:

FORMULA 5:

$$\textbf{Acceleration} = \frac{\dfrac{\textbf{2 x (Pi) x Radius}^2}{\textbf{Time}}}{\textbf{Radius}}$$

At this point we can "clean up" formula 5 by combining some of the terms and multiply out Pi, etc.



There are few places on the open road that lend themselves to "skid-pad" testing. This test was conducted on a 360° loop of private roadway with a very wide shoulder. Since skid-pad testing pushes the vehicle beyond its limits (speed increases until it the car spins out), saftey must be a principal concern.

The result is:

FORMULA 6:

$$\text{Acceleration} = \frac{39.5 \times \text{Radius}}{\text{Time}^2}$$

The acceleration measured in this equation can be thought of as a force trying to push the car out of the circle. (Technically the force is the product of the acceleration and the mass of the object, but for our purposes, the differences between acceleration and force can be ignored.) For the acceleration equation to produce the desired results, it is very important that the units of measurement (feet, seconds, etc.) are kept in order. The acceleration in the above equation is the lateral—also called *centripetal*—acceleration. It is measured in feet and seconds (to be precise: ft/sec/sec; or feet per second per second) when the units of measurement for the radius of the circle are in feet and the time for one lap is measured in seconds. To convert this into the popular "G" figure used in road tests, the calculated acceleration is divided by the acceleration of gravity—32.2 ft/sec/sec.

AN EXAMPLE OF "G" FORCE

Now let's look at a practical example. Suppose a 1965 Valiant made its quickest lap around a circle with a *radius* of 100 feet in 14 seconds; the lateral acceleration would be calculated as follows (using formula 6):

FORMULA 7:

$$\text{Acceleration} = \frac{39.5 \times 100}{14 \times 14} = \frac{3950}{196} = 20.15 \text{ Ft/sec/sec}$$

If we divide the acceleration by 32.2 ft/sec/sec (the acceleration of gravity), we arrive at 0.63g, which is quite low, even compared to the econobox sedans of today; they usually fall in the 0.70- to 0.74g range (meaning that the poor Valiant is just not going to "make it" in any test of performance handling).

Now that we can measure lateral "Gs," the next step is to raise this value as much as possible. Decreasing quarter-mile elapsed time on a drag car requires work, money and knowledge. The same investment is required to improve lateral acceleration. There are plenty of parts on the market that can improve handling, but knowing which parts to get, how to install them, and which pieces will properly work together to improve overall handling takes more than an understanding of terminology; it requires a *basic knowledge of the concepts that operate within a suspension system.* So on with the basics.

CENTER OF GRAVITY

The next basic concept is "center of gravity." The center of gravity (CG) is an imaginary point (usually located within the vehicle body) where, for the purpose of theoretical calculations, all of the mass of the car is located. A CG that is located four feet behind the front wheels and two feet off the ground, means that, for theoretical purposes, the car will act as if all of its weight is concentrated at this spot. CG is measured in terms of "height" (CGH—center of gravity height off the ground), and "location" (how far behind the front wheels). A CG closer to the front wheels will place more weight on the front end, and vice versa. And, in general, a CG located lower to the ground—a lower CGH—will allow a vehicle to corner flatter and faster.

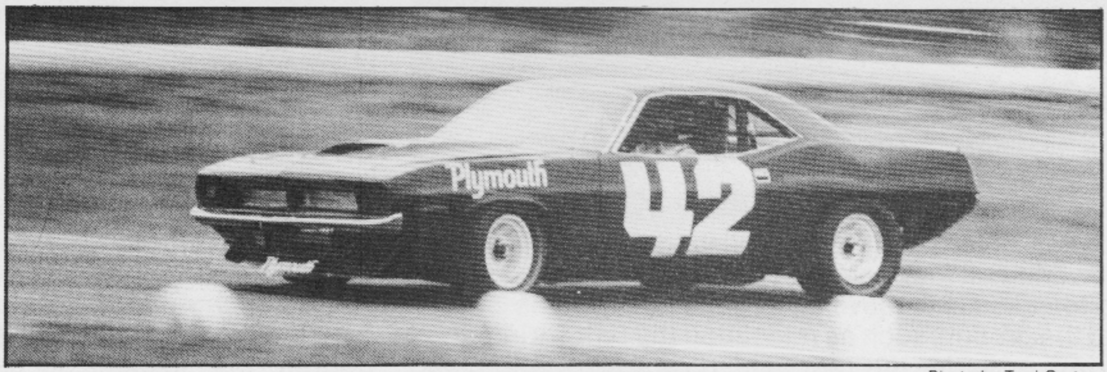
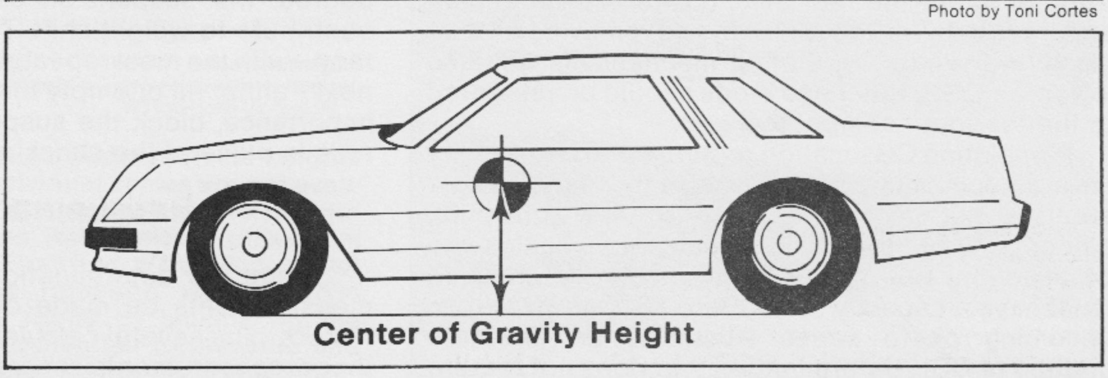
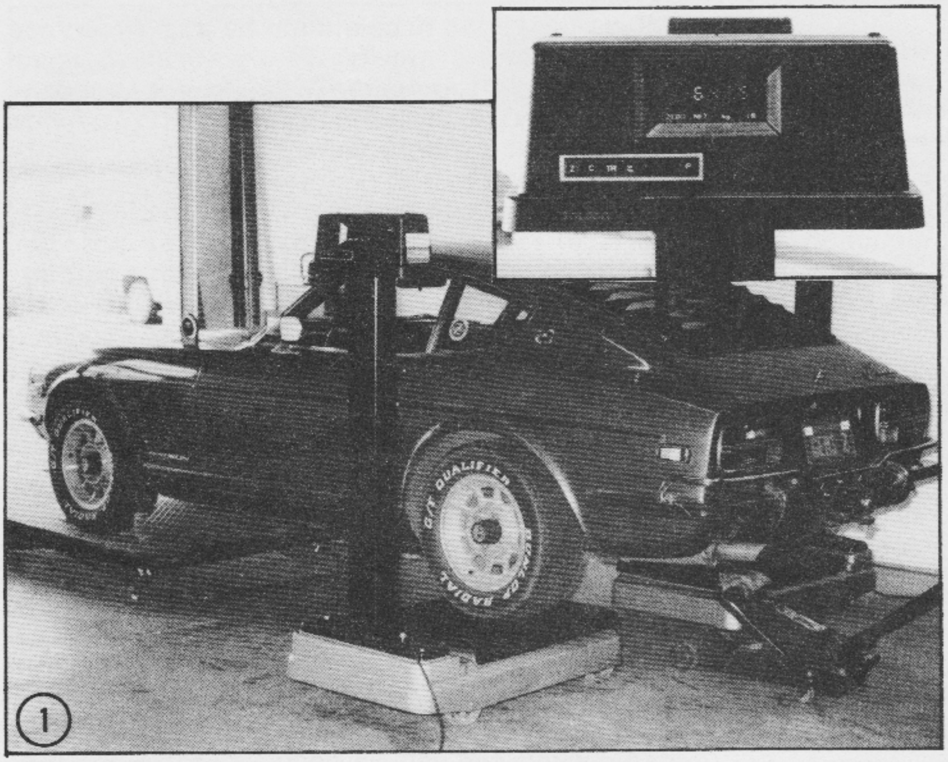


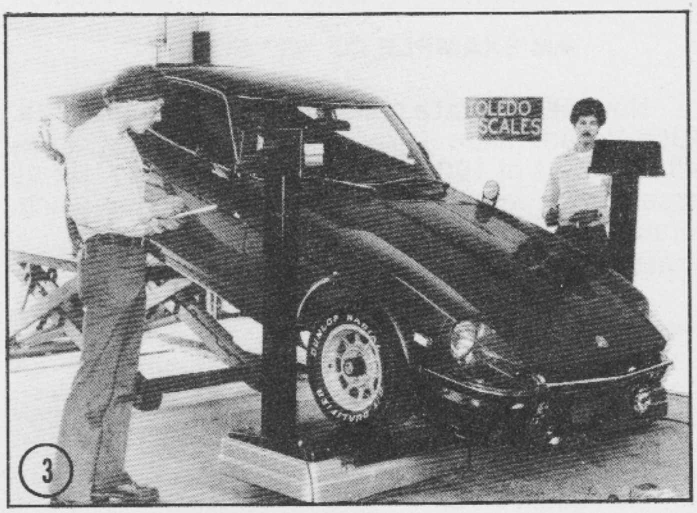
Photo by Toni Cortes



The center of gravity is an imaginary point where, for the purpose of calculation, all of the mass of the car is located. The CG is usually measured from the centerline of the front axle (front/rear position) and from the road surface (height). The center of gravity height (CGH) is a good indicator of cornering ability—the lower the better; typical values fall between 15 and 20 inches.



This is the preferred method to obtain the weights required for CGH determination. Ideally, four scales would be used; however, in this testing session only two scales were available. First the rear of the car was weighed to find *WTrto*, with the value read from the digital display. Next, the front of the car was weighed to find *WTfto*; the sum of these weights produced the overall weight of the car *WTtot*. Finally, the rear of the car was raised a measured amount *RHT* and the front end reweighed to find *WTftn*—the final value needed to calculate the CGH.



CENTER OF GRAVITY HEIGHT

For handling purposes, the lower the CGH the better. This can be accomplished most directly by lowering all four corners of the car. Another method often used is to move all large or heavy items closer to the ground, such as relocating the battery into the trunk or under the rear seat. (Lowering the engine and transmission, the radiator, and removing all side glass and window operating mechanisms will also lower the CGH, but these mods should be relegated to the "serious" competitor.)

Pinpointing CG location requires the use of a very flat area, some blocks that will raise the rear of the car about 20 inches (to measure front/rear weight shift), a floor jack, and four weight scales. Two scales can be used, but four make the job easier. Each scale must have a capacity of 1000- to 1500-lb. (Note: An upcoming photo series illustrates an alternate method of CGH determination that does not require using individual weight scales.)

When the car is raised to measure front/rear weight shift, a more accurate reading will be obtained if *all weight shift not caused by the fixed mass of the car is reduced or eliminated*. The greatest source of unwanted weight shift is due to liquid fuel "sloshing around" in a partly-filled tank. To a lesser degree, the suspension can compress/relax and contribute to weight shift. To obtain the best accuracy—with the most repeatability from one test to the next—either fill or empty the fuel tank, and of lesser importance, block the suspension with solid links/rods in place of the shock absorbers.

MEASURING THE CGH

Since CG determination requires that several measurements be made of car weight and other factors, plus several calculations be performed with these measurements, recording the data as they are taken is advised. The best method of organization is to construct a table of all the needed measurements and just "fill in the blanks" as you go along. The following table illustrates the data required and the order in which it will be measured/recorded:

WBo—Original wheelbase as measured on a level surface
WTrto—Weight on rear tires measured with car level
WTfto—Weight on front tires measured with car level
WTtot—Total weight of car
CG—Location of CG behind front wheels
RHT—Height rear of car is raised with blocks
WTrbk—Weight of both blocks used to raise rear of car
WTrtn—New weight on rear tires with rear of car raised
WTftn—New weight on front tires with the front of car raised
WBn—New wheelbase as measured with rear of car raised
CGH—The center of gravity height of vehicle

The first data item needed is the original wheelbase length. This is easily found by either reading manufacturer's literature or measuring the distance from the centerline of a front spindle—the front wheels must face straight forward—to the centerline of the rear axle, with the car on a level surface. Do not take this measurement with the car up on jackstands, as the rear axle can swing forward and give an erroneous reading. Once the wheelbase is known, record this length in the table next to **WBo**, the original wheelbase (the meaning of **o** for "original" and **n** for "new" wheelbases will be explained shortly).

The next required measurement is the weight on the front tires with the car in a level (original) position. The trick here is to carefully raise the car and place a scale under each wheel. With the car sitting level on the scales, the weight on each wheel can be measured. By adding the front scale readings, we can find the weight on the front end—**WTfto**—and the sum of the rear scales will provide the weight on the rear end—**WTrto**. The total weight of the car—**WTtot**—is the sum of all the scales. Record these values in the table next to **WTfto**, **WTrto** and **WTtot**.

The center of gravity position (fore/aft) is directly related to the wheelbase length and the weight distribution. And since we have measured the weight on the front and rear wheels, the overall car weight, and the wheelbase, we are now able to calculate the CG position. The first step is to divide the total weight of the car into the weight on the rear wheels. This will reveal the proportion of the total car weight on the rear wheels. The formula below details this calculation:

FORMULA 8:

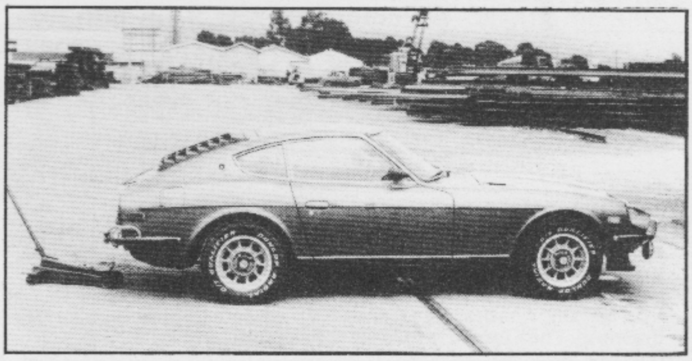
$$\text{Proportion of total weight on rear wheels} = \frac{\text{WTrto—rear wheel weight}}{\text{WTtot—total car weight}}$$

Now multiply the rear-wheel-weight proportion by the wheelbase in inches. This will locate how far behind the front wheels (in inches) the center of gravity is located.

FORMULA 9:

$$\text{CG}_{\text{behind front wheels}} = \text{Wheelbase} \times \text{Proportion of weight on rear wheels}$$

We have completed the first step in finding the exact CG location. We now know its fore/aft position relative to the front wheels, but before we can pinpoint its position in space, we still must determine how far off the ground it is located; i.e., find the center of gravity *height*—**CGH**. This is accomplished by raising the rear of the car with blocks that are placed on top of the scales. The new scale readings will



Here is an alternate method for finding vehicle weights to calculate CGH when individual scales are not available. A public truck scale will produce results that are remarkably close to those found with professional individual-wheel scales. First find *WTtot* (top photo) and then determine the front-end weight *WTfto* (middle photo). *WTftn* is found by jacking up the vehicle 15 to 20 inches (remember to measure and record *RHT*) and reweighing the front-end weight.

indicate how much weight "shifts" from the rear to the front of the car. Knowing this weight shift—and performing a little math—will give us the CGH, from which we can determine the location in space of the theroetical center of gravity for the vehicle.

After weighing both blocks—so their weight can be subtracted from the indicated rear-end weight—record this weight in the table—**WTrbk**. Now raise the rear of the car and insert the blocks between the scales and the rear wheels (both rear wheels must be raised the same height). Add the rear scale weights *and subtract the weight of both blocks*. The result is the weight on the rear tires with the car in the "new" raised position—**WTrtn**. Record this value in the table. For the sake of completeness, add the new front scale weights and enter this value also—**WTftn**.

As the rear of the car is raised, the measured wheelbase—that is, the distance between the center of the axles *as measured along the ground*—will be shorter than the true wheelbase. In the following steps we will calculate the length of this "new" wheelbase—**WBn**—which is the last data point required to determine CGH.

FORMULA 10:

$$\text{Length of new wheelbase} = \sqrt{(\text{WBo})^2 - (\text{RHT})^2}$$

Formula 10 calculates the new, shorter wheelbase for the raised car. (This formula is an application of the Pythagorean solution for triangles.) The original wheelbase—**WBo**—is squared and subtracted from the height of the blocks—**RHT**. The square root of the resulting value is the length of the new wheelbase—**WBn**—and should be recorded in the table.

DETERMINING CENTER OF GRAVITY HEIGHT

The next formula calculates the CGH and is somewhat complex, but this method eliminates the need for a difficult-to-construct engineering drawing, since the only non-formula method for CGH determination involves a graphic representation. In fact, the graphic method was used to develop the following CGH formula, and those who are interested in how this was done, refer to the accompanying section "Derivation of the CGH Formula." Now the formula:

FORMULA 11:

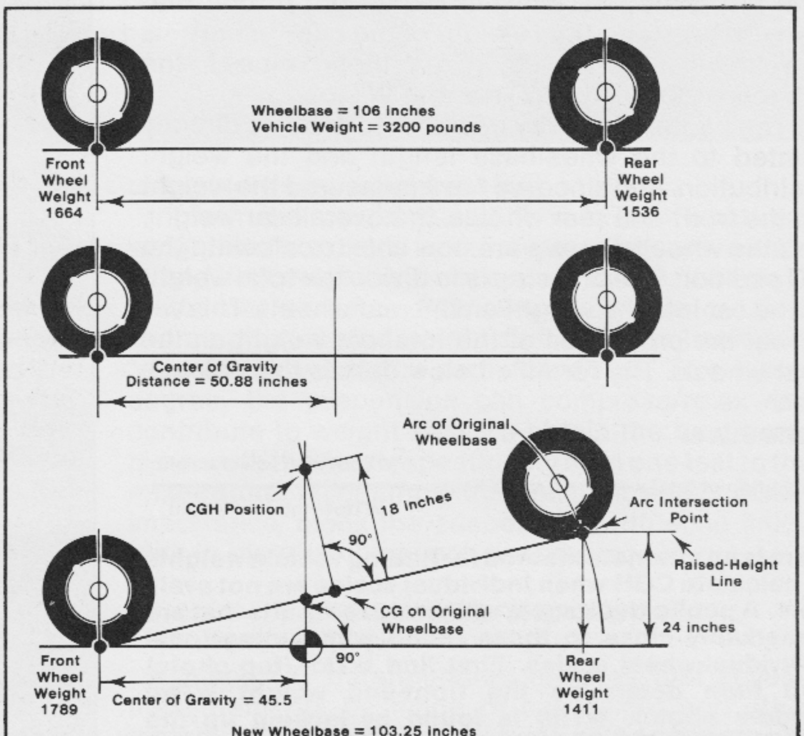
$$\text{CGH}_{\text{ground}} = \frac{\text{WTrto} \times \text{WBo} \times \text{Cos}\left[\text{Sin}^{-1}\dfrac{\text{RHT}}{\text{WBo}}\right] - \text{WTrtn} \times \text{WBn}}{\text{WTtot}\left[\dfrac{\text{RHT}}{\text{WBo}}\right]}$$

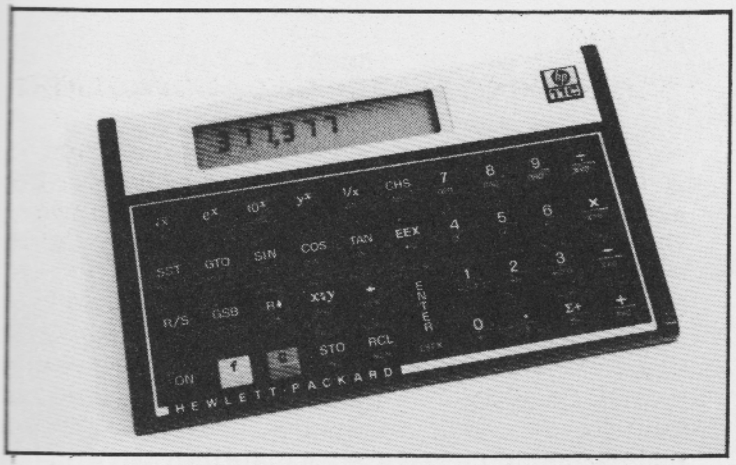
While this formula looks rather imposing, it is not difficult to solve with the aid of a simple hand-held calculator that will perform cosine and inverse-sine functions (see note below), such as the Hewlett-Packard model 11C or similar models. The Hewlett-Packard 11C is a programmable calculator; and once the program is keyed into the memory of the calcula-

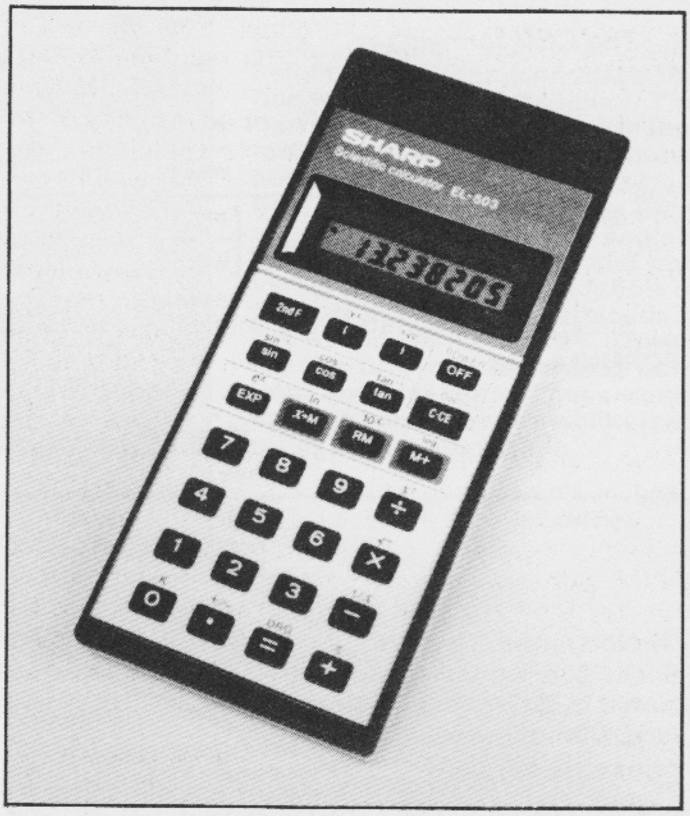
Note: The inverse sine, or "arc sine" as it is often referred to, is the angle that generates a known sine value. In other words, if the sine of a 45-degree angle is .707, then the angle whose sine is .707—the inverse sine—is 45 degrees. The inverse sine is represented in the CGH formula by a "Sin" raised to the -1 power.

DERIVATION OF CGH FORMULA AND GRAPHIC SOLUTION OF CGH

The CGH can be determined by using a scale drawing or the mathematical formula presented in the text. (In fact, the CGH formula in the text was derived from this graphical technique.) First draw the wheelbase—using an appropriate scale—and measure and record the weight on the front and rear wheels (upper drawing). Then locate the CG position using formula 9 (middle drawing). Now, draw an arc on the original wheelbase line (lower drawing) and move the rear wheels up as high as the blocks used during the weight-shift measurement (see photo series, page 16 and 17). Drop a vertical line from the rear-wheel contact point to the original-wheelbase line to find the "new-wheelbase length." Recalculate the CG position using the new wheelbase and the new wheel weights (weights found when rear of car was raised). Note on the lower drawing that the CGH is pinpointed at the intersection of two lines: 1) a perpendicular line—drawn from the horizontal baseline—located at the new CG position point, and 2) a perpendicular line—drawn from the now-slanting original wheelbase line—at the original CG location. The intersection of these two lines in our example occurs at a scaled distance of 18 inches above the original (slanting) wheelbase line.







The Hewlett-Packard Model 11C (above) is a very capable programmable calculator that can determine CGH within a few seconds after the variables are stored in memory. Inexpensive non-programmable calculators, similar to this Sharp EL-503, can also be used to determine CGH and other variables, although they take a longer time for each calculation. Before you make your selection, ensure that the calculator can perform these advanced functions: square, square root, sine and arc sine (and optionally, arc tangent).

tor, easy calculation and recalculations of the CGH can be performed. To aid those readers who have access to this popular scientific calculator, the HP-11C CGH program follows.

CENTER OF GRAVITY HEIGHT PROGRAM FOR THE HEWLETT-PACKARD 11C CALCULATOR

Switch the calculator to the program mode (**g P/R**), clear program memory (**f PGRM**), and enter this short 20-step program:

STEP	KEY	KEY CODE
001	LBL A	42,22,11
002	RCL 4	45,4
003	RCL 1	45,1
004	÷	10
005	STO 7	44,7
006	SIN^{-1}	43,23
007	COS	24
008	RCL 1	45,1
009	RCL 2	45,2
010	X	20
011	X	20
012	RCL 5	45,5
013	RCL 6	45,6
014	X	20
015	-	30
016	RCL 7	45,7
017	RCL 3	45,3
018	X	20
019	÷	10
020	RTN	43,32

INSTRUCTIONS: Load the data from the table—constructed in the last section—into storage registers 1 through 6 as follows:

WBo—(Original Wheelbase) Store in Register **1**
WTrto—(Original Weight on Rear Tires) Store in Register **2**
WTtot—(Total Weight of Car) Store in Register **3**
RHT—(Height Rear of Car is Raised) Store in Register **4**
WTtrn—(New Weight on Rear Tires) Store in Register **5**
WBn—(New Wheelbase) Store in Register **6**

When the data has been stored in the appropriate registers, just push the **f** and the **A** keys to start the program. In a few seconds the calculator should display a number. This is the center of gravity height (CGH) for the data from the table. To use the program for other data, just store the changed data in the proper registers and re-run the program by again pressing the **f A** keys.

USING A NON-PROGRAMMABLE CALCULATOR

If you do not have a *programmable* calculator, you must solve this formula manually by carefully entering the data in the proper sequence. Remember that it is best to solve an equation from the inside to the outside. Solve the CGH formula by substituting the data values from the table below and first solving the inverse-sine function:

SAMPLE DATA TABLE

(DATA FROM THE BARRACUDA EXAMPLE IN THE NEXT SECTION)

DATA SYMBOL	DATA VALUE
WBo	106 inches
WTrto	1536 pounds
WTtot	3200 pounds
RHT	24 inches
WTrtn	1411 pounds
WBn	103.25

The CGH formula:

FORMULA 12:

$$CGH_{ground} = \frac{WTrto \times WBo \times Cos\left[Sin^{-1} \frac{RHT}{WBo}\right] - WTrtn \times WBn}{WTtot \left[\frac{RHT}{WBo}\right]}$$

Substituting the data table values:

FORMULA 13:

$$CGH_{ground} = \frac{1536 \times 106 \times Cos\left[Sin^{-1} \frac{24}{106}\right] - 1411 \times 103.25}{3200 \left[\frac{24}{106}\right]}$$

Now, solve the inverse-sine function. First divide 24 by 106 (equals 0.226) and find the angle whose sine is 0.226—the inverse sine of 0.226. The answer is 13.086 degrees:

FORMULA 14:

$$CGH_{ground} = \frac{1536 \times 106 \times Cos\,(13.086) - 1411 \times 103.25}{3200 \left[\frac{24}{106}\right]}$$

Next—in the numerator—find the cosine of 13.086 (0.974). Then multiply the result by 1536 and multiply that result by 106; record this answer on paper (158587.8). Now multiply 1411 by 103.25 and also note this result (145685.75). Now move on to the denominator; divide 24 by 106 and multiply the result by 3200 (answer is 724.5). The CGH formula has now been simplified as follows:

FORMULA 15:

$$CGH_{ground} = \frac{158587.8 - 145685.8}{724.5}$$

Next, subtract the two numbers in the numerator and divide the result by the denominator. The result is 17.81 inches, which is the CGH measured from ground for the data in the table.

This manual technique, if done carefully, will produce values within a tenth of an inch of the CGH calculated by the programmable HP-11C—rounding errors account for the slight differences. But because of the greater chance for error using the manual method, always recalculate the CGH to make sure you get the same (hopefully correct) answer.

FINDING THE CGH ON AN EARLY BARRACUDA

Now that we have developed the formulas for locating the CG position and the procedure for solving them, working through an example will help clarify the entire process. Let's use an early Barracuda with a 106-inch wheelbase; therefore **WBo**, the first value we must write down in our table is 106 inches. Scales are now installed under each wheel of the Cuda. The front two scales indicate a front-end weight of 1664 pounds and the sum of the rear scales indicate a rear-end weight of 1536 pounds. The third and fourth values in the table—**WTfto** the weight on the front tires (1664-pounds) and **WTrto** the weight on the rear tires (1536-pounds)—can now be filled in. The total car weight is the sum of all the scales: 3200 pounds. Record this value under **WTtot** in the table.

Before we raise the car to determine the remaining values for the table, we can calculate the CG position behind the front wheels. This is done with formula 9. We divide the rear weight (1536 pounds)

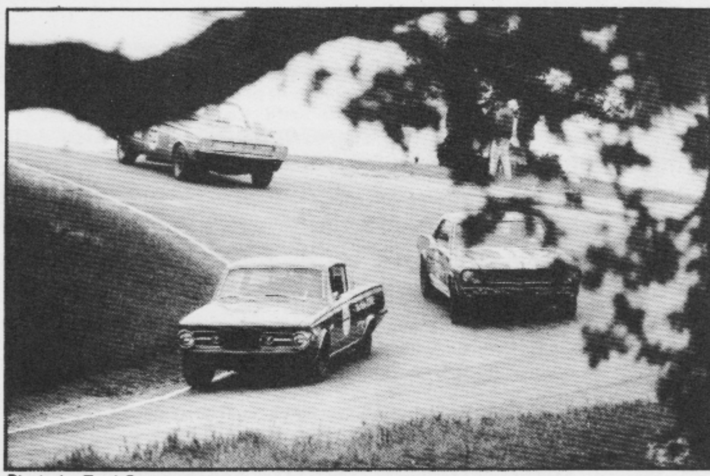
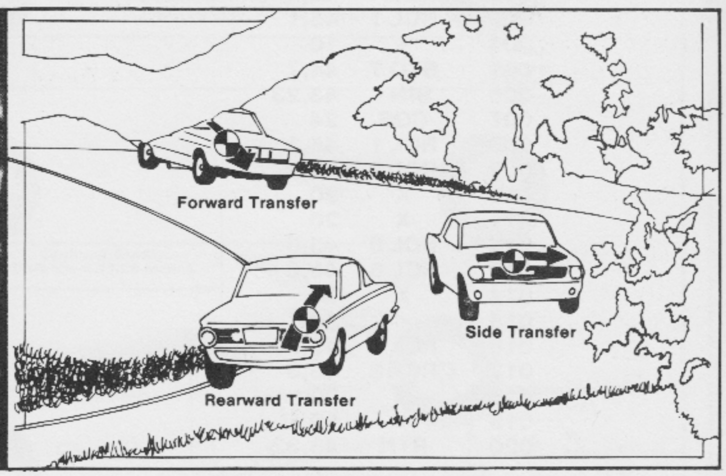


Photo by Toni Cortes



The CG location is not a fixed point within the chassis; rather it can move within the vehicle body during braking, cornering, and acceleration. During hard braking (Dart entering turn) the CG moves forward as weight shifts to the front of the car. Under lateral acceleration loads (Mustang in turn) the CG shifts to the outside of the chassis; placing greater traction demands on the outside tires. And under acceleration (the Barracuda leaving the turn) the CG shifts to the rear; aiding traction by increasing the weight on the rear wheels.

Photo by Toni Cortes

Ride height, center-of-gravity height, and alignment are all important variables in handling. The Dart and Mustang (left), and Barracuda (below) exhibit excessive body roll causing diminished traction. In addition, the severe body roll generates odd camber angles, further reducing traction and cornering ability (note that only 50% of the outside tire on the Barracuda is contacting the pavement). Lowering the CGH and increasing both front and rear roll stiffness would improve handling and stability (and make driving more fun!).

by the total car weight (3200 pounds), producing 0.48. Multiply the 0.48 rear-wheel weight proportion by the 106- inch wheelbase. This yields 50.9 inches, which is the center of gravity fore/aft position. **CG** is located 50.9 inches behind the front axle and this value should be added to the table. After we find the last five entries in the table and determine how high off the ground the CG is located (the CGH), we will know the exact center-of-gravity position for the Cuda.

The rear of the car must now be raised by installing blocks in between the scales and the rear tires. For this example, the rear of the Cuda is raised 24 inches and the blocks weigh 40 pounds each. So the next two values of 24 inches for **RHT** and 80 pounds (both blocks) for **WTrbk** can be added to the table. Reading the scales with the rear of the car raised indicates a rear-wheel weight of 1491, but this includes the weight of the blocks. Subtracting **WTrbk** from the indicated scale weight gives the true weight on the rear wheels of 1411 pounds—**WTrtn**. The front wheel weight can be directly read by summing both front scales—**WTftn** is 1789 pounds.

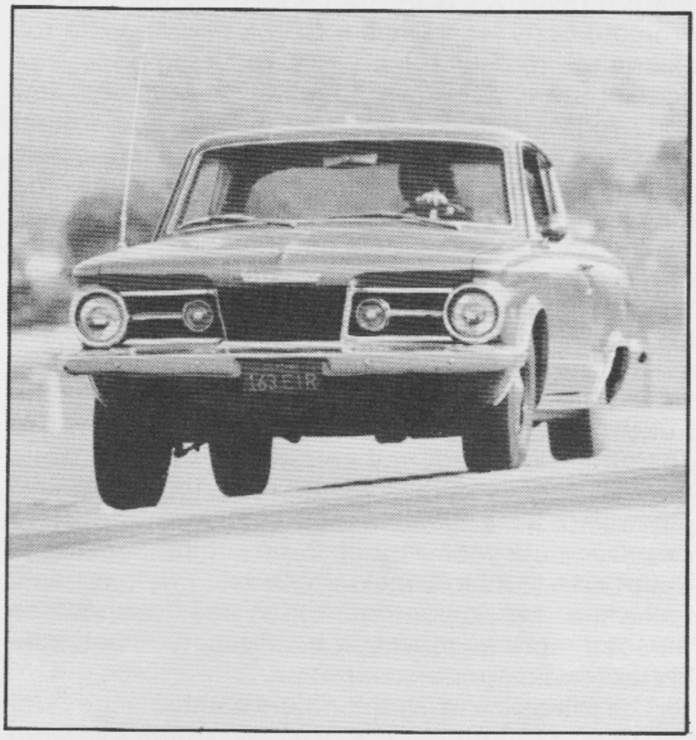
We have now obtained sufficient data to calculate the "new" wheelbase with formula 10 by: 1) squaring the original wheelbase, 2) subtracting the squared height of the blocks, and 3) taking the square root. This produces the "new" wheelbase length as measured along the ground:

FORMULA 16

$$\text{New Wheelbase} = \sqrt{(106)^2 - (24)^2} = \sqrt{10{,}660} = 103.25 \text{ inches}$$

The "new" wheelbase—**WBn**—is 103.25 inches. We have found all the values needed to calculate the CGH, and the table looks like this:

DATA SYMBOL	DATA VALUE
WBo	106 inches
WTrto	1536 pounds
WTfto	1664 pounds
WTtot	3200 pounds
CG	50.9 inches behind front axle
RHT	24 inches
WTrbk	80 pounds
WTrtn	1411 pounds
WTftn	1789 pounds
WBn	103.25



You may now either store these values in the memory storage registers of the programmable calculator and run the CGH program, or use the manual method of evaluating the CGH formula. Either method should determine a CGH value of about 17.8 inches. Therefore, the center of gravity is located 17.8 inches above the ground and 50.9 inches behind the front wheels—obviously a point within the car body, just ahead of center.

EVALUATING CGH VALUES

What do these CGH numbers mean? For a sedan like the Barracuda to be competitive, the center of gravity height should fall in the range of 13 to 18 inches (the lower the better). While a CGH of 17.8 inches is good for a stock Barracuda, the value could be reduced further by lowering the car, installing lower profile tires and wheels, and/or lowering or reducing the weight of some internal components (battery, seats, etc.). Every time the height of the car is changed or weight is moved around inside the car, the center of gravity can change. After the lowering and weight movement has been accomplished, reweigh the car and compile a new data table. Then plug the new values into the CGH formula and determine the new CG location; the value will either verify that the correct changes have been made (a lower CG) or indicate that more work needs to be done (a higher or unchanged CG).

CG AND ROLL AXIS

Analyzing the center-of-gravity position will help predict the handling characteristics of the car. One indicator is the CG location relative to the "roll axis," an imaginary line running through the car around which the body of the car pivots (rocks left/right) when turning. A CG located above the roll axis will promote body lean, while a CG on or below the roll axis tends to cancel lean. But, reducing body lean doesn't always improve handling. In fact, very low body roll can adversely affect side-to-side weight transfer and handling. But before we explore this principle further, we will find out how to locate the roll-axis line.

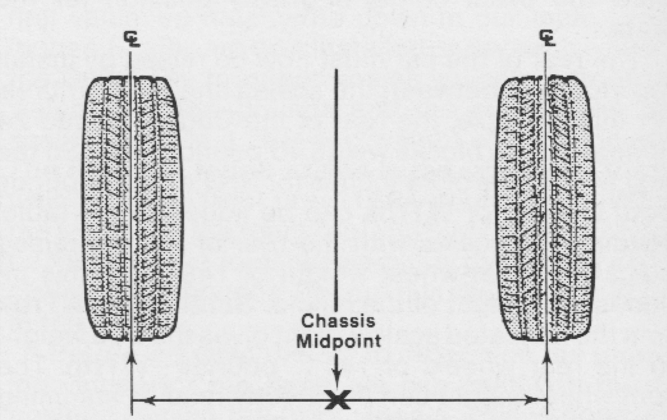
The roll-axis location is determined by the basic geometry of the suspension. Cars that utilize an independent front suspension and a live rear axle—like the Barracuda in our CGH calculation—will have a roll axis that passes through the center of the vehicle, gently sloping downward from the rear to the front. A roll axis of this type will force the front end to absorb greater body roll and weight transfer, while the rear end distributes weight more uniformly. This proves to be a good design because the more sophisticated geometry of the front suspension is better able to absorb inside-to-outside weight transfer, while the power-transferring rear suspension "sees" a more even weight distribution, resulting in better traction and acceleration.

The roll-axis line (by definition) connects the front and rear "roll centers". The front roll center is located in the center of the car, equidistant between the front wheels; and the rear roll center is similarly located in between the rear wheels. The roll centers are the theoretical points around which the front of the vehicle (front roll center) and rear of the vehicle (rear roll center) pivot in a turn. Locating the roll centers will determine the precise position of the roll-axis line.

FINDING THE FRONT ROLL CENTER

Locating the front roll center is a tricky operation; and unless you have access to a factory blueprint of the front suspension, determining the precise roll-center position generally requires some guesswork, even with a carefully prepared scale drawing. Accuracy in taking, recording, and reproducing measurements is important if the results are to be at all reliable.

The Barracuda used in the earlier CGH determination will provide a good example of the process used for roll-center measurement. The first step is to locate the centerline of the front tires (the midpoint in the tread) and then the centerline of the car (halfway between the centerlines of the front tires):

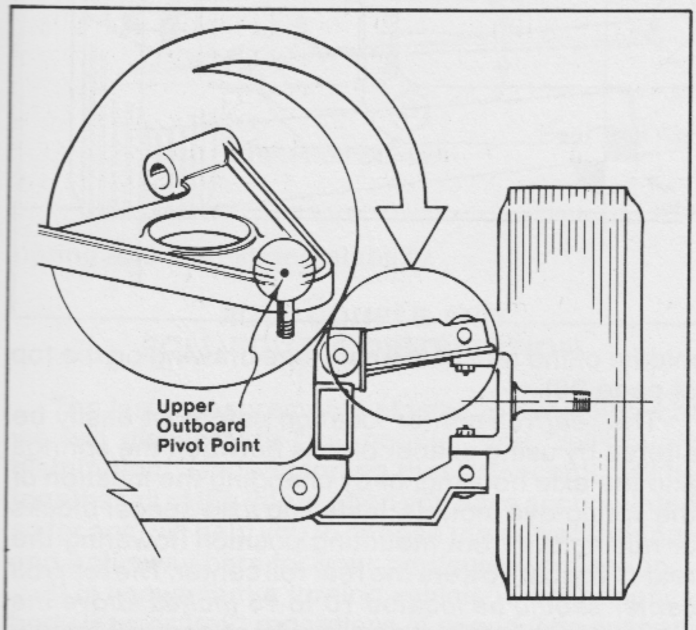


The chassis midpoint is located at the centerline of the track (equal distance between each tire). It is on this vertical line that the roll center is located.

Photo by Toni Cortes

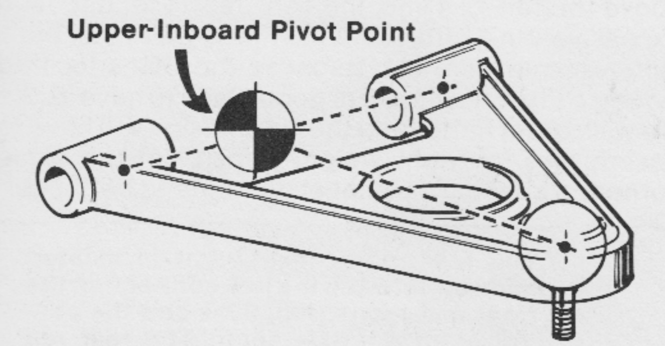
While the roll-center may be static at a particular ride-height, it is certainly not static when the car is in motion. As the suspension moves up and down, the location of the front roll-center also moves. The extent of this movement is determined by the basic suspension design and geometry; particularly important is the CGH location relative to the roll-axis line, which greatly influences body-roll forces.

Next, measure the distance from the centerline of the car to the center of the left-side upper and lower control-arm pivot points (except for the upper-inboard pivot location) and locate these measurements on your drawing. NOTE: The ball-joint pivot point is not located where the ball joint connects to the control arm; it is where the "ball" pivots in the ball-joint socket (see insert):



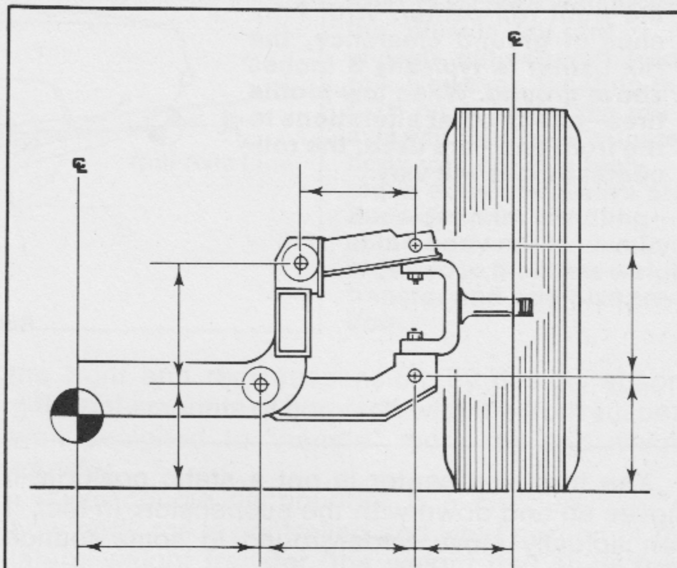
The ball-joint pivot point is inside the socket, not where the ball-stud is attached to the spindle. Disregarding these subtleties will induce small errors that will generate inaccurate angles and measurements. Remember: accuracy counts.

The upper-inboard control-arm bushings are not in a horizontal line, and the pivot point position must be located as indicated:



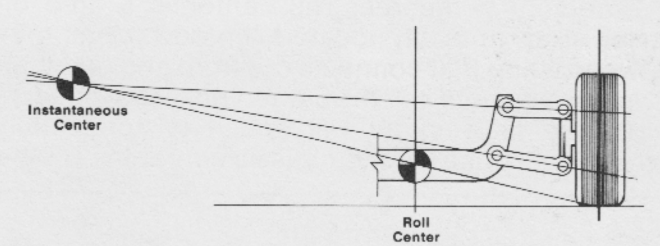
The pivot point is located (for the purpose of calculation) along the sloping line connecting the front and rear upper-control-arm bushings (the upper control arm is mounted at a slight angle to generate anti-dive characteristics). The precise point is located by extending a perpendicular line to the center of the ball joint; the intersection of these two lines marks the pivot point.

Now, measure the distance from the control-arm pivot points to the ground. These measurements will allow you to transfer the control-arm pivot point locations to the scale drawing:



Take your measurements carefully and record them on a scale drawing similar to this one. Double check each measurement to ensure that all the pivot points have been properly located. An error at this point will certainly result in an erroneous roll-center location. With all measurements recorded, the roll center position can be plotted either in inches above (as in this case) or below ground level.

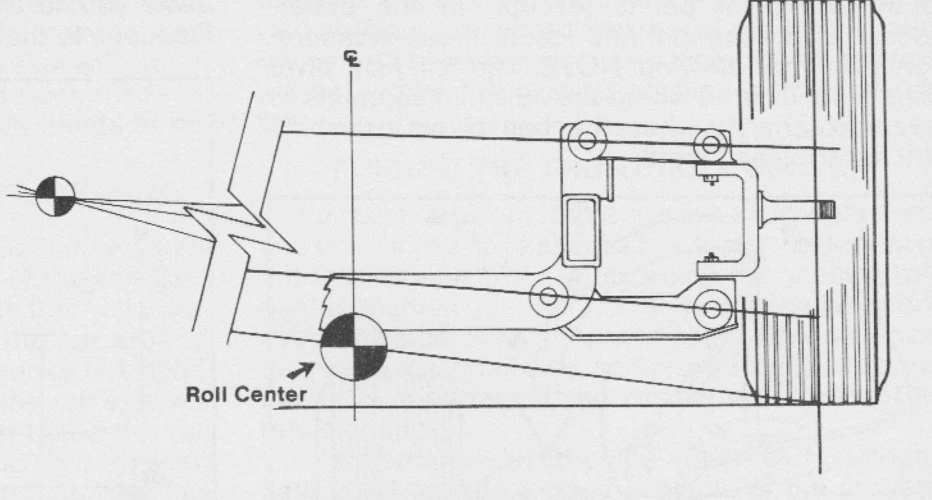
The next step is to locate the "instantaneous center"—a point of intersection for the lines drawn through the pivot points of the upper and lower control arms. It is called the "instantaneous center" because, without the physical constraints imparted by the "real-world" design of the front suspension, this is the theoretical point around which the suspension travel would pivot.



Connecting the suspension pivot points will pinpoint the instantaneous center. The roll center is found along a line passing through the instantaneous center and the tire contact patch. The intersection of this line and the vertical chassis centerline is the precise location of the front roll center.

With the instantaneous center located, the front roll center can be found. Draw a line from the instantaneous center to the point where the centerline of the tire contacts the ground. Where this line crosses the car centerline is the location of the front roll center.

For Mopar suspensions set to the factory ride-height, this is the usual relationship between the K- member, front geometry, and the front roll center. With 7 inches of ground clearance, the roll center is typically 5 inches above ground. When low-profile tires—or any other alterations to the front end—are used, the roll-center height will vary.



The front roll center is not a static position; it moves up and down with the suspension. In fact, it can actually move underground in some jounce situations.

A low roll-center height will optimize handling. *Most Chrysler front suspensions locate the roll center approximately 5 inches above ground*, unusually low for a production car. This position moves closer to the ground when the car is lowered for competition use, reaching an optimum height of about 3 inches. In addition, the long lines converging to the instantaneous centers help to reduce camber changes during jounce and rebound. In other words, the basic Chrysler design keeps the tires more "upright" during hard cornering by minimizing camber change. This upright positioning maintains a larger tire contact patch on the pavement, improving handling and maximum cornering speed.

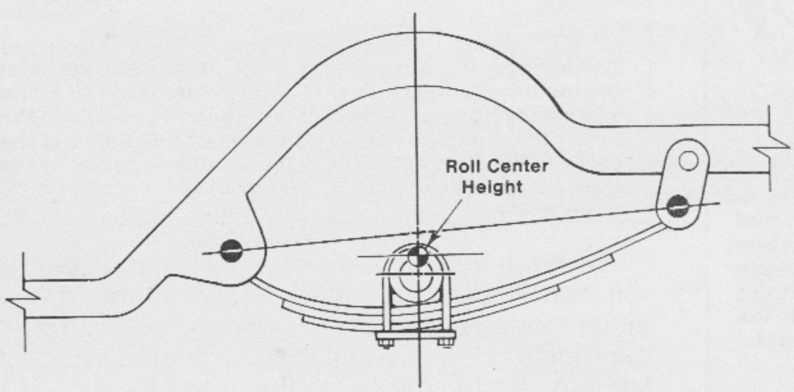
FINDING THE REAR ROLL CENTER

Determining the rear roll center is a little bit easier. It is vertically located (up/down) half way between a line that connects the front and rear leaf-spring eye mounts and the centerline of the rear axle shaft (see drawing below); and it is horizontally located (left/right) on the chassis centerline (in the middle of the rear of the car—see drawing on the top of page 26).

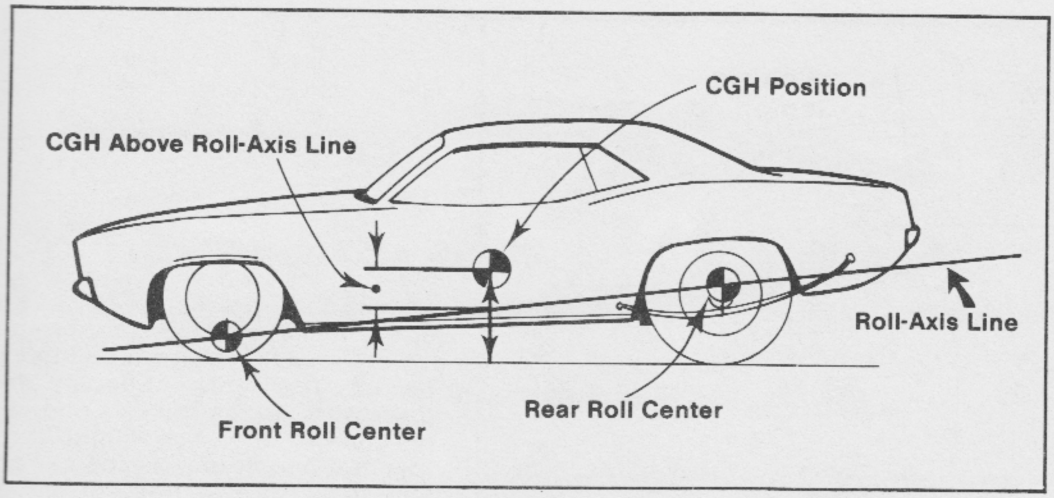
The rear roll-center location can most easily be altered by using spacer blocks between the springs and the axle housing, or by changing the location of the spring-eye mounts. Installing axle-spacer blocks or raising the front mounting position (lowering the rear of the car) lowers the rear roll center. *The rear roll center should be located 10 to 15 inches above the ground.* Within this range, the front end will handle most of the weight transfer during cornering.

LOCATING THE ROLL-AXIS LINE

Once the position of both roll centers has been determined, these points can be located on a scale drawing of the car. The roll-axis line connects the front and rear roll centers, and its position relative to the CG will indicate *what type* of suspension modification techniques should be employed to improve handling. If the CG is located more than 16 inches above the roll-axis line, the vehicle will tend to lean excessively in a turn, requiring heavy anti-sway bars, stiffer suspension, etc. Because the CG is located closer to the roll-axis line (a good place to have it), the car will corner flatter and the suspension will have to absorb less left/right weight transfer at the same cornering speed. This almost guarantees better cor-



The rear roll-center height is located midway between the line connecting the front and rear spring eyes and the centerline of the axle shaft. The rear *roll center* is found on the centerline of the chassis (in the middle of the car) at a height above ground equal to the roll-center height (see the drawing at the top of page 26).



This is the roll-axis line and the basic relationship between the front/rear roll centers and the center of gravity height. When the CGH is located well above the roll-axis line, the chassis will generate body roll. However, locating the CGH on or below the roll axis will reduce/eliminate body roll; but this technique does not always produce optimim handling—some body roll is usually required to facilitate weight transfer and optimize traction.

nering and high-speed stability.

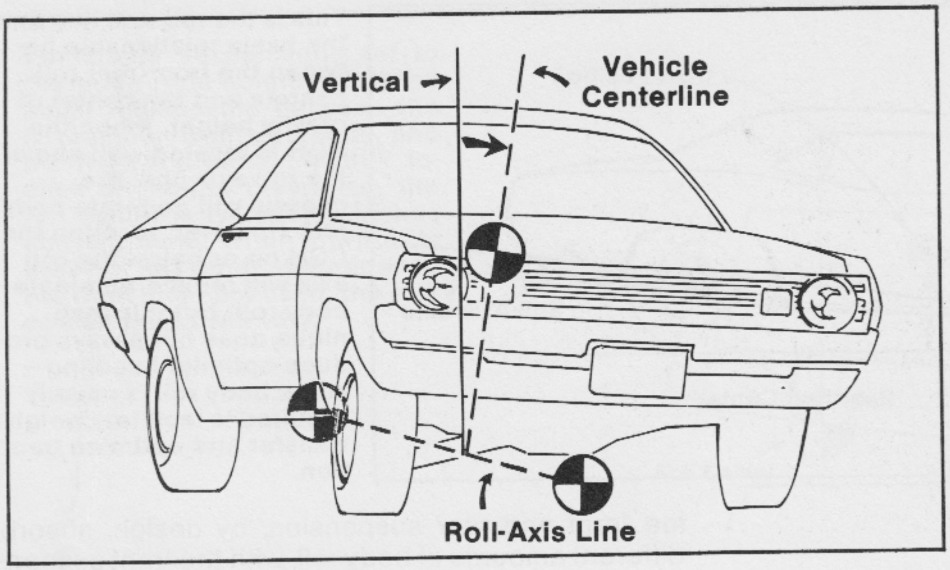
ROLL COUPLE AND ROLL-COUPLE DISTRIBUTION

The last measurements of vehicle dynamics that we will discuss are "roll couple" and "roll-couple distribution." Understanding these concepts will tie together all of the information you have accumulated so far and will help you to choose the correct springs and anti-sway bars for your particular application.

It is known that a turning vehicle will experience body lean or "roll" regardless of speed. The suspension system controls that motion and minimizes tire slippage and loss of road contact. We know (from the discussion on roll centers and roll-axis position) that the front and rear suspension, by design, absorb different amounts of body roll, with the front suspension designed to "handle" more roll and weight transfer.

"Roll-couple distribution" is a measurement of how body roll is controlled and what portion of the lateral-weight transfer (the weight that shifts from the inside to the outside wheels in a turn) is handled by the front or rear suspension. Roll-couple distribution is often explained (although not always accurately) by saying: the stiffest part of the suspension system will be the first to break away in a turn. A typical example is a car set up for drag racing; it is often a poor-handling vehicle in turns—prone to loose rear traction because the rear suspension is usually far stiffer than the front.

The right parts—springs, sway bars, shocks, tires, bushings, etc.—will produce the right results: flat, responsive handling. The calculation and analysis of center-of-gravity height, roll-couple distribution, and the position of the roll-axis line all helped to give the owner of this fast and nimble Barracuda a competitive edge.



As centrifugal cornering forces are applied, the roll-axis line forms a "pivot base" around which the car rotates. The center of gravity also pivots around the roll axis and can be thought of as the weight of an inverted pendulum; as the car accelerates, decelerates, or turns, the CG will "swing" from vertical, always pivoting about the roll axis.

DETERMINING THE ROLL-COUPLE DISTRIBUTION

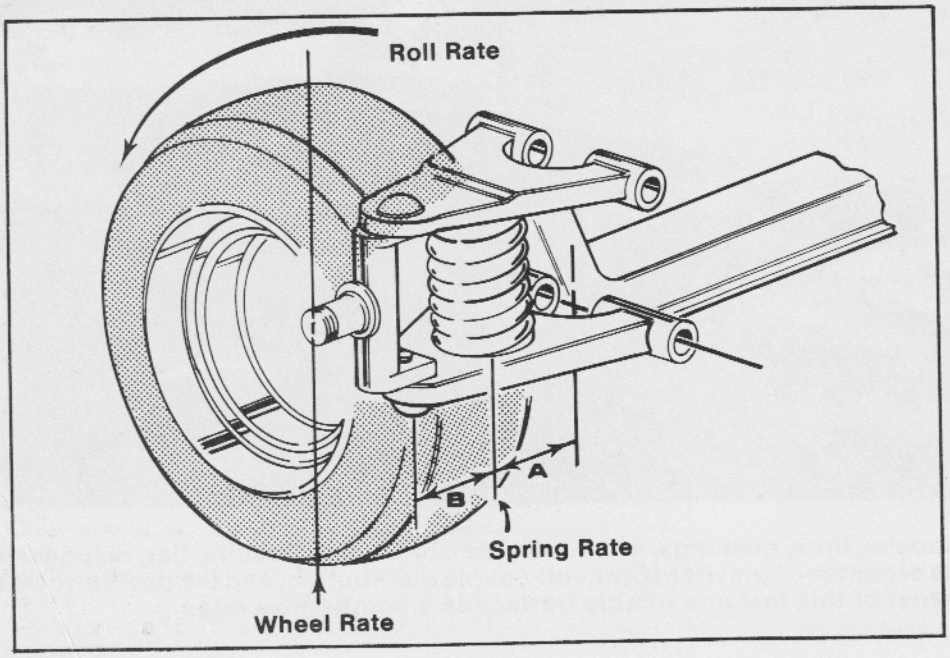
Chrysler engineer Larry Rathgeb defines roll couple as: "the fore and aft distribution of lateral weight transfer—the whole concept revolves around understanding and controlling tire-slip angles, front to rear." To the chassis engineer, the concept of roll couple and roll-couple distribution are relatively straightforward; for the rest of us however, they are more easily understood when some basic spring theory is kept in mind.

IT'S SPRING TIME

Automotive springs are rated three separate ways: 1) the static or "spring rate," 2) the "roll rate," and 3) the "wheel rate." The static spring rate (referred to as **Ks**) is the amount of force (in pounds) required to *compress the spring one inch*. The static rate measures only the spring; that is, all measurements are made with the spring removed from the suspension. The roll rate and wheel rate, however, are measurements of spring resistance while installed in the suspension. The roll rate (**Kr**) is a rating of the spring/suspension resistance to body roll while cornering. The wheel rate (**Kw**) is the measured spring rate at the tire contact patch.

When a spring is installed in a suspension system, it acts only indirectly on the wheel. Its load is often reduced by the geometry of the suspension design. Because of this, values are different for wheel and roll rates, and in some cases (as with coil spring suspensions) the values differ significantly. Regardless of which "rate" is used to measure springs, we will consider all measurements to be in pounds-per-inch of deflection (more on this later).

The three spring rates (static, wheel, and roll) each measure the spring doing a different job. The wheel and roll rates are often lower than the static rate since the "lever-arm" action of the suspension (control arms) reduces the effective load of the spring at the wheel. Most Ford and GM cars use an upper-and-lower control-arm front suspension, with the coil spring attached between the frame and



The motion ratio for coil-spring suspensions can be determined by dividing distance A (distance from the inner control-arm pivot to the spring centerline) by distance A + B (the overall length of the lower-control arm), as shown in formula 17. This lever ratio establishes how much spring rate—the amount of force required to compress the spring one inch—is transferred into wheel rate—the measured spring rate at the tire contact patch; see formula 19. The roll rate, calculated by formula 18, is a rating of the spring/suspension resistance to body roll while cornering.

This is the original Chrysler "mule" prototype Kit Car. Before it was retired, Chrysler Product Development and Petty Engineering did extensive chassis research to discover the "ultimate" for their package-racer. Mathematical analysis, similar to the formulas presented in this book, plus many hours of on-track testing were invested before the final blueprint was drawn for the production Kit Car.

either the upper or lower arm. Since the spring does not act directly on the wheel but is located inward from the spindle, only a percentage of the static rate is transferred to the tires and wheels. how much the static rate is reduced can be determined by the "motion ratio," a factor based on the suspension design.

To illustrate this point, let's examine a Camaro coil-spring suspension. To determine the motion ratio, we must measure the length of the lower control arm and pinpoint the spring centerline position (how far from the control-arm inboard pivot the center of the spring is located). These measurements are typically 16.0 inches (length of control arm) and 8.0 inches (inboard mounting from the pivot). Formula 17 shows the relationship between these lengths and the motion ratio:

FORMULA 17

$$\text{MR (Motion Ratio)} = \frac{\text{Spring Centerline Distance}}{\text{Control Arm Length}}$$

OR

$$\text{MR} = \frac{\text{8 inches}}{\text{16 inches}} = 0.5$$

(Note: This formula is simplified to illustrate our point. The complete equation takes into account wheel offset, distance from the lower ball joint to the wheel centerline, etc. and is rather complicated for our purposes.)

With the motion ratio known, the roll rate is easily calculated from the known static rate of 800 lbs/in with the following formula:

FORMULA 18

$$\text{Roll Rate} = \text{Static Rate X Motion Ratio}$$

OR

$$\text{Kr} = \text{Ks X MR} = 800 \text{ X } 0.5 = 400 \text{ lbs/in}$$

Wheel rate is used by most engineers to calculate roll-couple distribution, and the following formula will find the wheel rate from a known static rate and motion ratio:

FORMULA 19

$$\text{Wheel Rate} = \text{Static Rate X (Motion Ratio)}^2$$

OR

$$\text{Kw} = \text{Ks X (MR)}^2 = 800 \text{ X } (0.5)^2 = 800 \text{ X } 0.25 = 200 \text{ lbs/in}$$

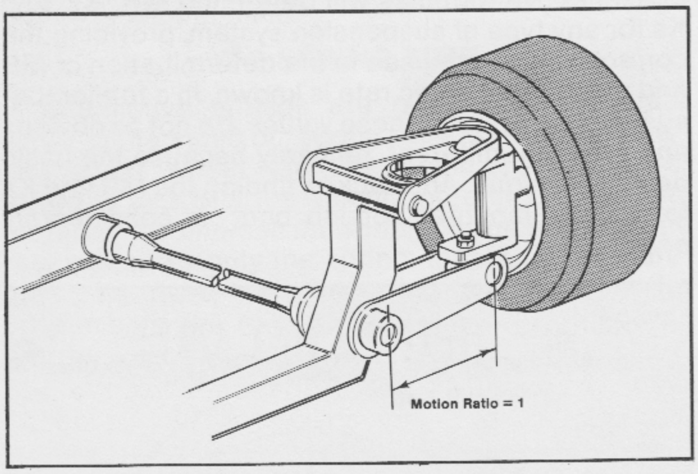
The above formulas will determine **MR, Kr,** and **Ks** for any type of suspension system, providing the correct values are used in the determination of **MR** and the correct static rate is known. In a torsion-bar suspension system, these values are not as obvious and are quite different, primarily because the basic design is unique. And, luckily, finding the MR and Ks for parallel-mounted torsion bars is *considerably easier.*

A final "production" version of the Kit Car was driven by Boddy Unser at Pike's Peak. Taking the checkered flag was proof that the investment in chassis development was well spent.

TORSION BAR RATING

A typical coil-spring suspension, as we discussed in the previous section, may mount the spring about halfway outboard on the lower control arm, producing a motion ratio of 0.5. In a Mopar torsion bar suspension system (of the parallel-mounting design), the torsion bar is directly connected to the lower control-arm inner pivot. In this design, the torsion bar is rated with a motion ratio equal to 1. Therefore the spring rate is equal to both the roll and wheel rates. (Check this out for yourself using formulas 18 and 19. Remember that 1 squared equals 1). In fact, in a Chrysler torsion-bar system, bars are rated solely by their vehicle application (based on the lever arm designed into the suspension and the stiffness of the bar).

Now that you know how torsion bars are rated, you should not be confused by the large static spring rates used in coil spring suspensions. A typical Camaro or Firebird may need a coil spring with a static rate (**Ks**) of 800 to 900 lbs/in to deliver a wheel rate of 180 to 200 lbs/in. This is *definitely not the case with torsion bars.* Don't let yourself get caught up in a numbers game—trying to "out spring rate" the other guy. Because of Chrysler's rating method (**Ks** equals **Kw**), the 200-lb/in wheel rate of the Camaro suspension would be equal to using a 200-lb/in torsion bar on a Mopar.



With torsion bars mounted parallel to the frame, the motion ratio is equal to one (1), because the "A" and "A + B" lengths are the same; see formula 17 and the lower drawing on page 26. With a motion ratio of one, the bar rating is equal to the wheel rating; i.e, a torsion bar rated at 200 lb/in will apply 200 lb/in at the wheel.

Photo by Toni Cortes

Different suspension designs require different spring rates. This Barracuda is using a 216-lb/in torsion bar, while the Corvette—with a coil-spring suspension—must use coils with two to three times this rate to produce the same wheel and roll rates.

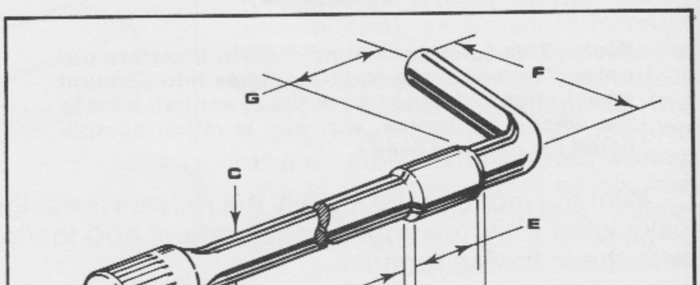
TRANVERSE TORSION BARS

Many 1976 and later Mopars used torsion bars that are mounted transversely (across the front of the vehicle). Because both the mounting techniques and functional dynamics of transverse bars are similar to standard sway bars, the formula for rating transverse torsion bars (TV-bars) closely approximates the math used to determine sway-bar ratings and is rather complex. The following formula calculates the spring rate **Ks** for a transverse bar in pounds per inch:

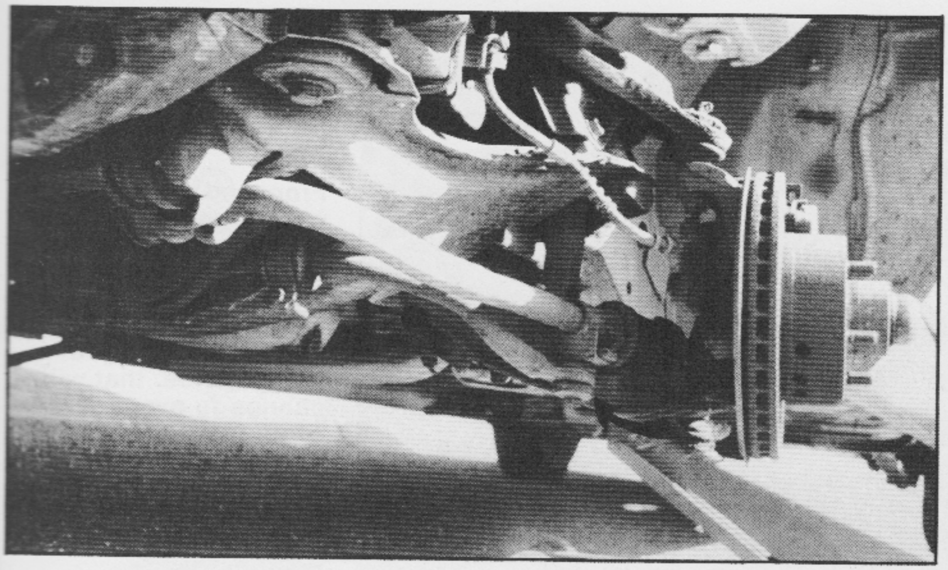
FORMULA T-BAR 1

$$\mathbf{Ks\ (lbs/in)} = \frac{\mathbf{Pi \times R^{4} \times G \times \left[Tan^{-1} \frac{1}{A} \right]}}{\mathbf{L \times A \times (114.6)}}$$

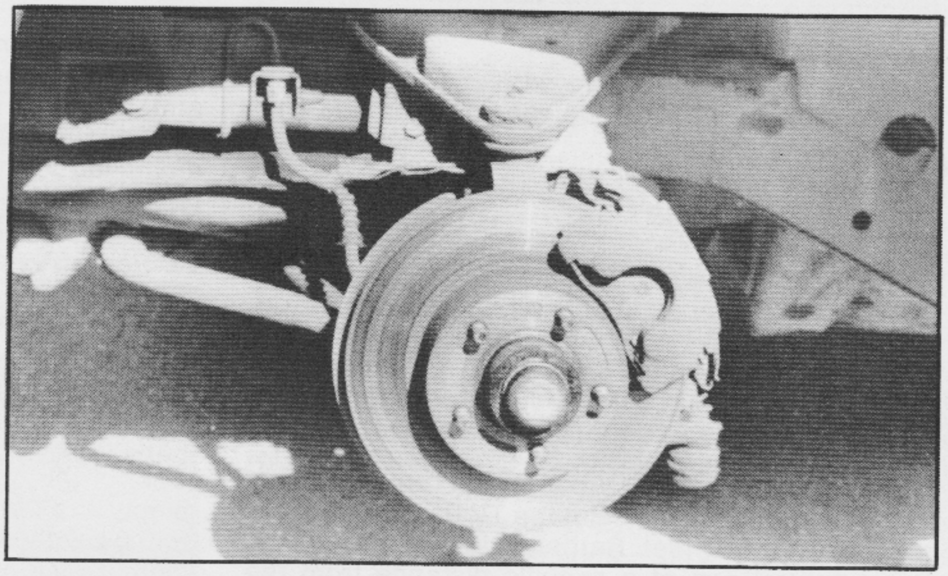
From left to right in the numerator: **Pi** is 3.1416, **R** is the *radius* of the TV-bar (measured at the effective diameter—see illustration) raised to the fourth power, **G** is the *modulus of elasticity* (a measurement of the material structure of the bar—a typical value for TV-bars is 11,500,00), and **A** is the TV-bar working-arm length. In the denominator **L** is the *effective* bar length (distance from the spline end to the retainer end—see illustration) and 114.6 is a constant. This equation employs an arc-tangent term (represented



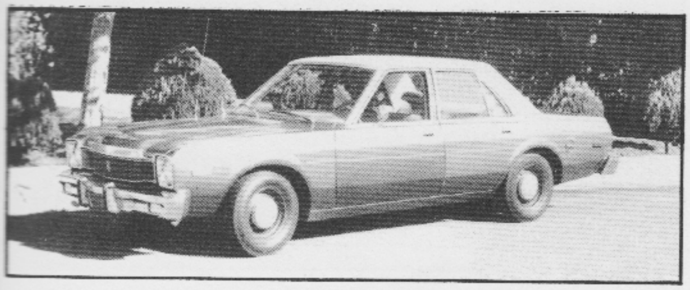
These are the critical dimensions needed to design a transverse torsion bar. While bar rating is similar to a straight torsion bar, the added dimensions (B, E, F, and G) add complexity to the design. Bar rating is still determined by lever arm length (F), working diameter (C), and working length (B).



The unique transverse-torsion-bar mounting allows the torsion bars to function as both springs and strut rods. Note that the anti-sway bar is connected to the bottom of the TV-bar at the lower control-arm (arrow); this attachment is similar to sway-bar mounts used on early B- and C-body cars. Since some Aspen/Volare models were equipped with heavy-duty police suspensions, stiffer-than-stock T-bars are readily available.



The new transverse-torsion-bar mounting forced the factory to modify several suspension components: the upper and lower control-arms, the K-member, and front sway-bar. Disc brakes were standard on most TV-bar-equipped cars, including this 6-cylinder model with heavy-duty suspension.



While retaining some of the earlier Dart (Valient) body lines, many improvements and changes were made to the suspensions and brakes on the Dodge Aspen (and the Plymouth Volare). The most notable change was the transverse—rather than longitudinal—torsion-bar mounting.

by **Tan-1**). The arc tangent—like the arc sine used in the earlier CGH formula—is an inverse trigonometric function. It can be thought of as *the arc whose tangent is a known value.* This value can be found in a trig table or calculated by pressing the "TAN-1" key on the HP-11C or other scientific calculators. Make sure the calculator is set for decimal degrees (not radians or grads). If you look up the arc tangent in a table, ensure that the angle you find is a *decimal number.* That is, it is in degrees and tenths/hundredths of a degree (not degrees, minutes, and seconds).

The following example will help illustrate the procedure for solving the TV-bar equation. We will use the following dimensions from a 1981 Cordoba:

1981 Cordoba TV-Bar Dimensions

Dimension	**Value**
Radius of TV-bar	**0.5 inch**
Working Length	**13.75 inches**
Effective Bar-Arm Length	**25 inches**

The radius of the TV-bar is measured along the narrow section between the spline end and the retainer pivot. First measure the diameter, then divide this number by 2 to find the radius. The working length is the distance between the spline end and the retainer pivot, measured from the *base of the tapers at each end of the bar*—see illustration. The effective bar-arm length is measured from the bar centerline to the tip of the arm. This is the "working-arm" length of the TV-bar, a shorter arm length will

Photo by Toni Cortes

Front and rear roll couple work together to establish how the vehicle will respond during hard cornering. This Challenger has been setup for an easily controlled neutral slide, that will change to oversteer when the throttle is applied. This type of chassis is easy to drive because it "feels" stable, predictable, and responsive.

increase the spring rate.

Using these values and setting **G** (modulus of elasticity) to 11,500,000, we can solve formula 1 for the TV-bar spring rate.

FORMULA T-BAR 2

$$\text{Ks (lbs/in)} = \frac{\text{Pi} \times .5^{4} \times 11{,}500{,}000 \times \left[\text{Tan}^{-1} \dfrac{1}{13.75}\right]}{25 \times 13.75 \times (114.6)}$$

$$\text{Ks (lbs/in)} = 238.4$$

Therefore, **Ks**—the TV-bar spring rate in pounds per inch—equals 238.4. But the TV-bar does not act directly on the wheel. Rather, it is attached to the lower control arm at a specific distance from the wheel hub. So the motion ratio must be determined before the roll and wheel rates of the bar can be calculated. These further measurements are required:

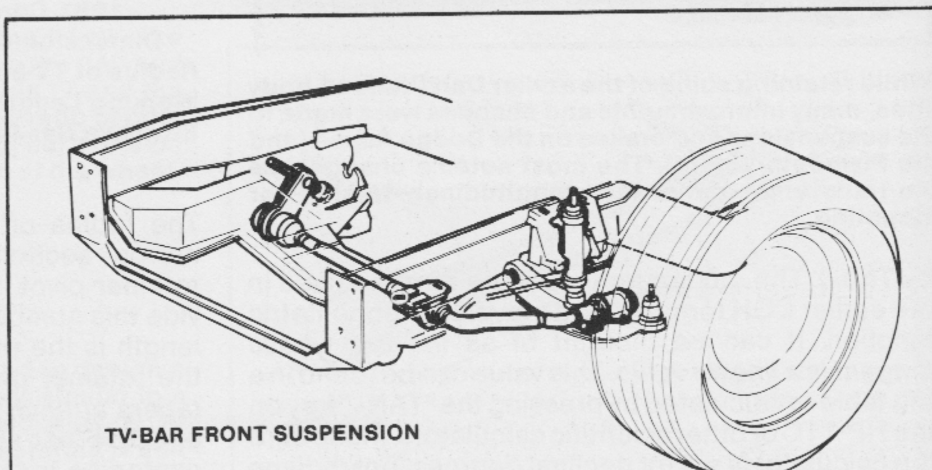
Dimension	Value
TV-Bar Attachment Point	10.0 inches
Control-Arm Length	12.75 inches

The motion ratio is determined by formula 17. The TV-bar attachment point (measured from the control-arm pivot) is divided by the overall control-arm length. The resulting motion ratio of 0.784 is then multiplied by the spring rate (formula 18) to find the roll rate for the TV-bar. Plugging in the numbers produces 187 lbs/in—a good value for a stock street car.

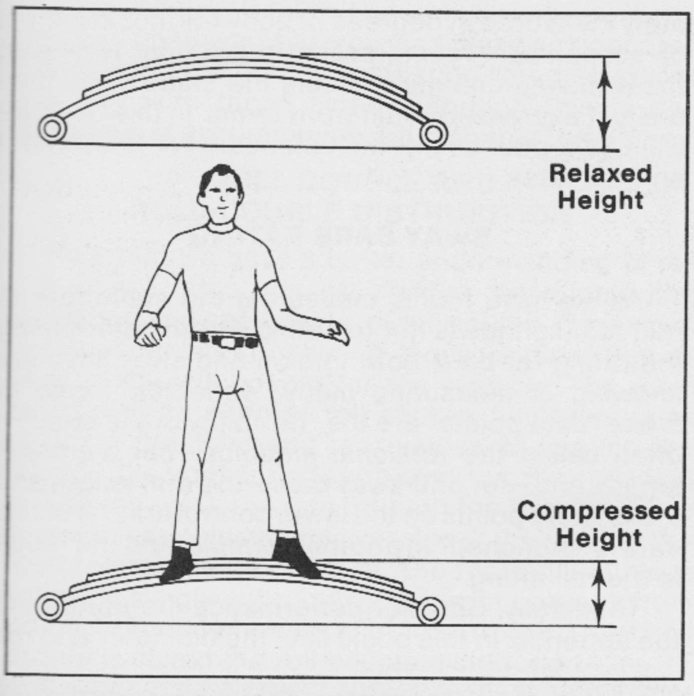
REAR-LEAF SPRING RATES

The calculation of rear spring rates is much easier. The rear roll rate is determined by multiplying the static rate of the spring by the "lever arm" (motion ratio) of the rear end. The rear-end motion ratio is simply the center-to-center distance between the spring perches divided by the center-to-center track width (track width is the distance between the centerline of the tires). For example, our stock Barracuda has a track width of 55.6 inches and a spring-perch distance of 43.0 inches. This produces a motion ratio of .773 (43.0 divided by 55.6). Multiplying this by the static spring rate will determine the roll rate at each wheel. The wheel rate is the motion ratio *squared* multiplied by the static spring rate (see

Many 1976 and later Mopars used torsion bars that are mounted across the front of the vehicle (transversely). These transverse bars (called TV-bars) function similar to sway bars, but are located at one end (with an adjustor) and attach directly to the lower-control arm at the other. The working length of the bar is the section between the adjustor and the retainer pivot (bracket mounted to opposite frame rail). The TV-bar motion ratio is found by dividing the bar attachment point (measured from the control-arm inner pivot) by the overall control-arm length.



TV-BAR FRONT SUSPENSION



Rear-spring static rate can easily be determined when the springs are removed from the chassis. After weighing yourself and measuring the relaxed height of the spring, stand on the spring as close to the rear-end attachment point as possible. Then measure the compressed height and divide your weight by the amount of spring deflection (relaxed height less the compressed height).

Photo by Toni Cortes

Optimum spring rates and sway-bar rates, combined with a low center of gravity, produce minimal body lean. As suspension technology has progressed, it has always been possible to upgrade the basic suspension system to current competitive levels. This has been possible because of easy component interchangeability in Mopar torsion-bar suspensions. Note: reduced body lean retains more optimum wheel camber angles and better traction.

During cornering the chassis rotates through an angle (roll angle) that causes the outside suspension elements to compress and the inside elements to relax. And although Barracudas don't have coil springs, the compression/relaxation concept is clearly illustrated here.

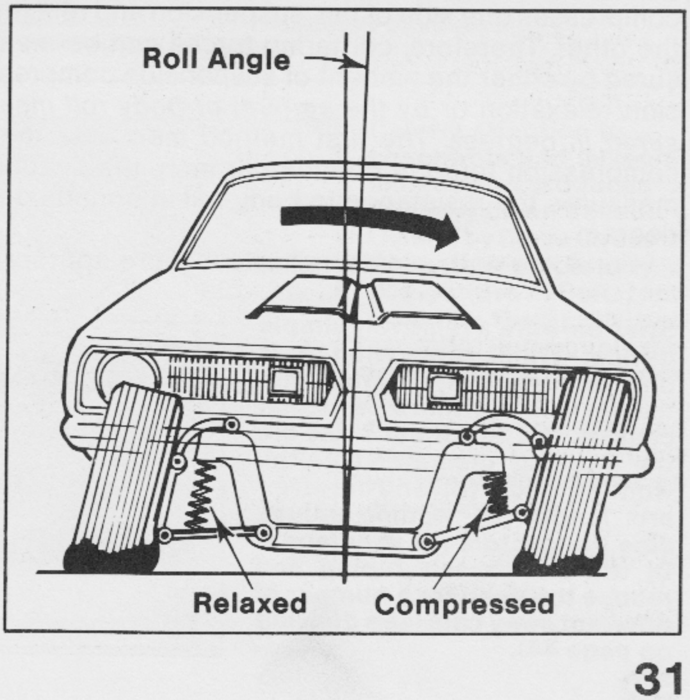
formula 19).

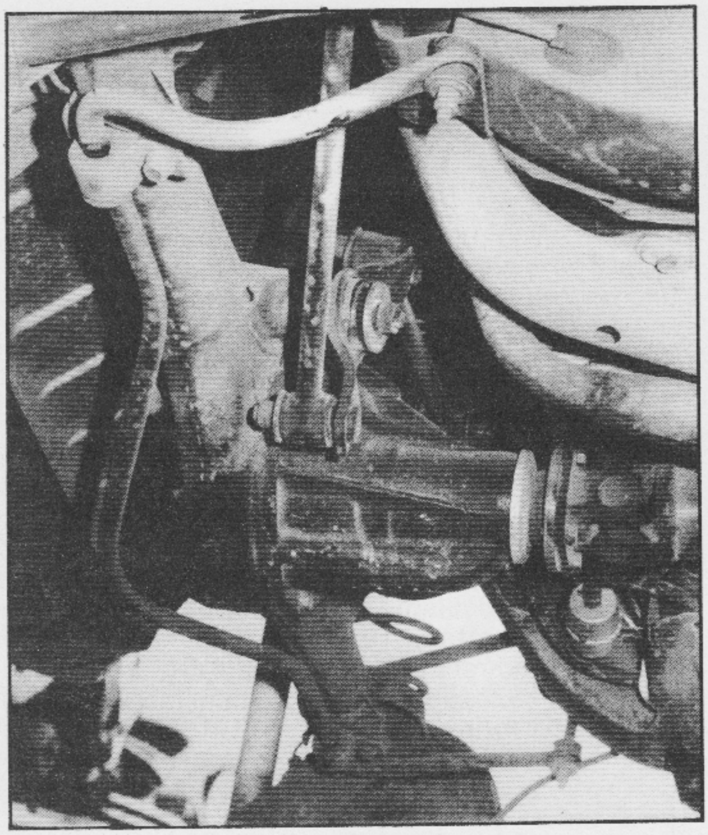
But these calculations only work if you know the static rate of the leaf springs. Chrysler literature can sometimes provide this information; but if all else fails, determining leaf-spring static rate is relatively easy (easy, that is, once the rear springs have been removed from the car). First, weigh yourself. Now position the spring on the ground so the center arch is up. Stand on the spring and have someone measure how much closer the center of the spring comes to the ground. Your weight divided by this measurement will give the static rate of the spring in pounds per inch.

Suppose that you weigh 189 pounds (the heavier you are, the more accurate the measurement will be); and when you stand on the spring, it deflects 3/4 of an inch. Divide 189 pounds by the deflection (0.75-inch) to get the static rate in pounds per inch: 252 lb/in. Now multiply this rate by the rear-axle motion ratio (0.773 calculated in the last paragraph). The result is a roll rate of 195 lb/in, which is pretty stiff even for a road-racing car. The wheel rate is the static rate (252) multiplied by the motion ratio squared (0.598): 151 lb/in.

OH, THOSE TERMS!

Up to this point, we have defined wheel rate (**Kw**) as the load imparted on the tire contact patch by the spring as the suspension compresses/relaxes. If you are familiar with Chrysler literature, you are probably aware that the "term of choice" for torsion bar, anti-sway bar, and leaf-spring ratings is not wheel rate but roll rate. As we have discussed, wheel rate and roll rate are the same measurement for torsion bars; since motion ratio is 1, each term describes the same force in the same units—Chrysler uses pounds per inch. However, for rear springs and sway bars, the motion ratio is not equal to 1.0 so the wheel rates must be calculated with formula 19.





The rear suspension on this Mazda RX-7 uses a typical rear axle with coil springs. However, radius rods and a watts linkage are used to prevent side-to-side axle motion, and a rear anti-sway bar helps the rear suspension to control body roll. Notice that the mounting points of the bar arms are considerably inboard from the wheels, reducing the "roll rate" of an obviously high "static rate" (large-diameter) bar. While the method of sway-bar mounting used on this Mazda adds to unsprung weight, it is probably the best alternative, considering the limited space.

Roll rate can be defined as: "The resistance of the sprung mass to rotate about the roll axis from forces generated in cornering." As the car body rolls, it compresses one side of the suspension and relaxes the other. Therefore, cornering forces can be measured by either the amount of suspension compression/relaxation or *by the amount of body roll measured in degrees*. The first method measures the compression force in pounds per inch; the second measures the resistance to body roll in pounds per degree.

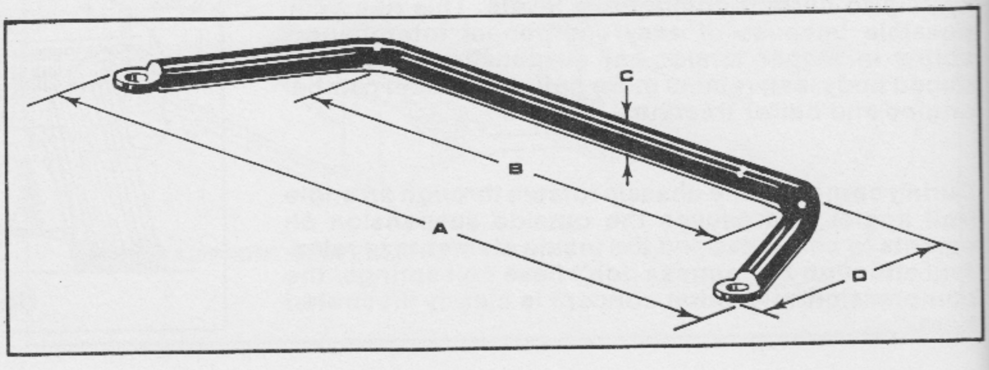
For some enthusiasts, roll rate is more appropriately measured in degrees of body roll, not in inches of suspension compression/relaxation. However, these measurements measure the same thing; they are just expressed in different terms. In this book we will use a pounds-per-inch standard for both wheel and roll rates.

SWAY BARS RATING

As we have found, measuring the static rate of rear leaf springs is not too difficult; but determining the rating for bars, both torsion and sway, involves knowing or measuring many variables. Some of these "data points" are the "flexibility of the steel"—often called the *torsional modulus*—bar diameter, length, and—for anti-sway bars—the arm length and attachment points on the lower control arm. All of this data is "crunched" in another formula, and the result is the roll rating.

Thankfully, Chrysler performance literature (and the appendix in this book) lists the diameter and roll

While anti-sway bars are not the "cure-all" they were once considered to be, they make an important contribution to handling when combined with other correctly selected suspension components. But even on their own, sway bars are an inexpensive addition that can improve the handling of virtually any car.



Sway bars are a relatively simple device. While dimension A is only important in ensuring that the bar will fit within a particular chassis, the key dimensions are the working diameter (C), the working length (B), and the lever-arm length (D). Adding adjustable connectors (to vary length D) will aid chassis tuning and reduce the need for a number of different sway bars (see drawing on page 44).

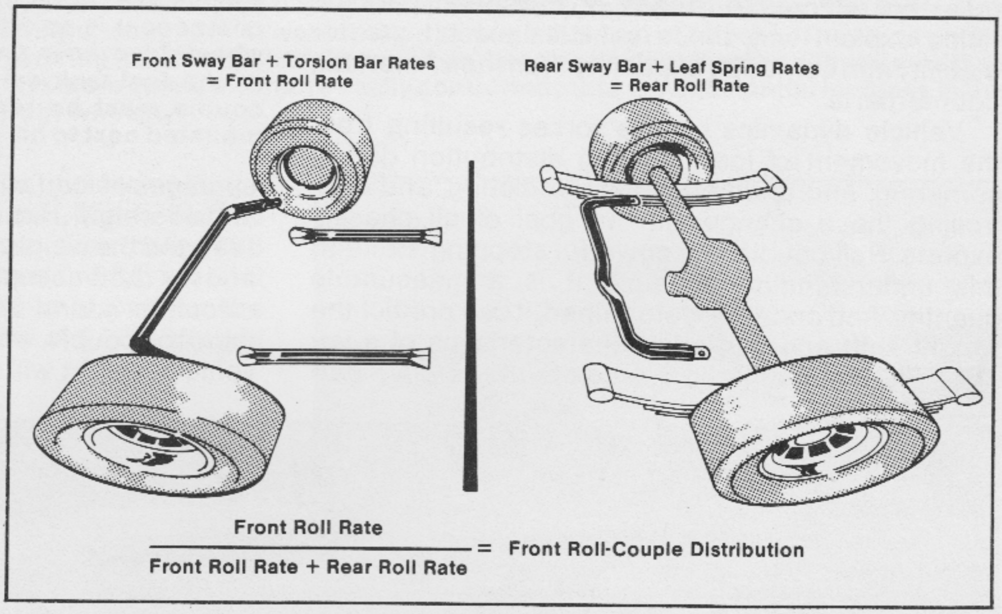
rate for various torsion and anti-sway bars. If you have an "unknown" bar, you can make an educated guess of the roll rate by comparing the diameter of your bar with those listed. Although this is no guarantee of accuracy, you will probably come close.

ROLL COUPLE AND ROLL-COUPLE DISTRIBUTION

Now that we have a better understanding of roll and wheel rates, we will further investigate the derivation and application of roll couple. *The sum of all the wheel rates (including those for the anti-sway bars) is the roll couple for the vehicle.* The most useful way to express roll couple, however, is to compare the front and rear values. By adding the rates of the front springs and sway bar, then dividing this by the total roll rate (of both the front and rear suspension, including all sway bars), the result is the front roll-couple percentage. The rear roll couple is found by simply subtracting the front percent from "100%." The comparison of the front-to-rear roll couple is called the roll-couple distribution.

For example: Suppose our Barracuda has a front wheel rate of 1) 227 lb/in for each torsion bar—producing a torsion-bar total of 454 lb/in and 2) a sway bar rate of 184 lb/in. The sum total is a suspension roll rate of 681 lbs/in. The rear wheel rate (as we computed in a previous section) is 151 lbs/in per spring—302 lbs/in total. Therefore the roll-rate percentage of the front is 69% (the front roll rate of 681 divided by the total roll rate for the vehicle of 983). If the front roll-rate percentage is 69%, the rear must be 31% (100% - 69% = 31%). Therefore, the roll-couple distribution for the Cuda is 69/31.

Roll-couple distribution is often referred to by the front suspension percentage. Most race cars, whether set up for oval track or road racing, have a front roll couple in the range of 77% to 93%. The larger the front roll-couple percentage, the greater the tendency toward understeer. And the lower the percentage of front roll couple, the greater the tendency to oversteer—more on this phenomenon later. The front suspension requires a higher percentage of roll couple to handle the high inside-to-outside weight transfer that occurs while cornering (predicted by the forward-sloping roll-axis line). However, a lower rear roll-couple percentage is preferred, because reduced left/right weight transfer will improve trac-



Determing the front roll-couple distribution, by adding the front roll rates and rear roll rates and performing a little math—as illustrated here, can be a tremendous aid in chassis tuning. Knowing roll-couple distribution—and having some experience—one can determine whether a particular chassis combination will overstear and require a higher precentage of front roll couple, or understeer and require less.

Photo by Toni Cortes

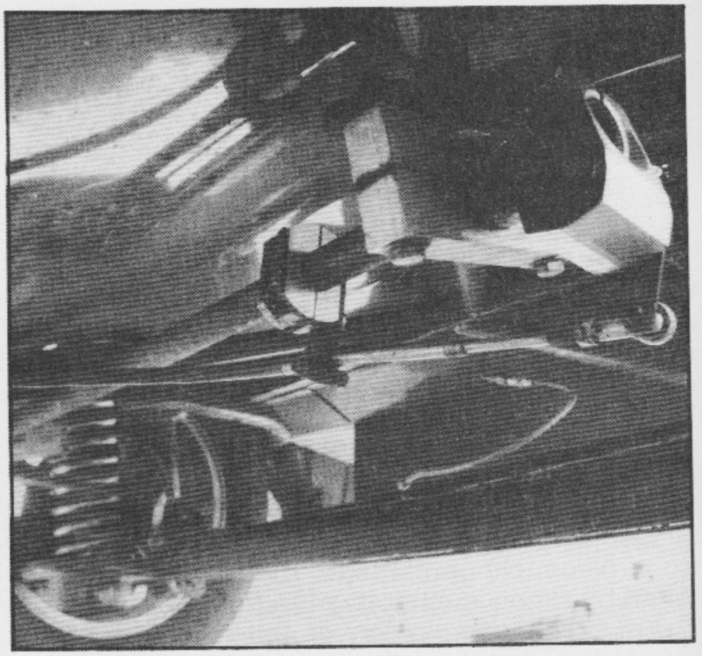
Careful attention to coachwork and suspension design produced this very clean Valiant. Owned by Steve Cooke of Chap racing, it is very competitive in Solo-I and Solo-II in the SE Division of SCCA. Roll-couple distribution and weight distribution were thoroughly analyzed and perfected; the result is a winner of several division championships.

tion and acceleration. The "natural tendency" to high front roll couple—which proves beneficial in front-engine, rear-drive vehicles—poses a severe disadvantage in front-drive vehicles, where the front tires must not only handle weight transfer and steering but also accelerate the vehicle out of turns. Although front-drive cars do have some advantages, high-speed handling is not one of them.

ROLL-COUPLE REALITY

As you brandish roll-couple distribution numbers about, keep in mind that *the same roll-couple distribution figures can be achieved using entirely different spring-rate ranges*. You can build a 77/23 roll-couple distribution with either very light springs or with *proportionately stiffer* springs—although the overall roll couple will not vary. As the speed of a vehicle and the stability requirements increase, so do the spring rate and roll-couple requirements. Since many domestic cars are equipped with "rubberband" suspensions, you can rightly conclude that they weren't designed for cornering and stability at speeds much faster than 55 mph. The higher spring-rate and roll-couple ranges of European GT cars helps explain why these vehicles exhibit greater stability and cornering capability than their American counterparts.

Vehicle dynamics involve forces resulting from the movement of load (weight) distribution during cornering; and understanding, predicting, and controlling these changes is the goal of all chassis experts. Roll couple is a powerful stepping stone to this understanding because it is a measurable quantity. And once it is determined, it can predict the weight shift and cornering characteristics of a vehicle. But roll couple just predicts the *relative percentages* of load shift. Is there a way to calculate the actual weight reduction (in pounds) on the inside tires and the weight increase on outside rubber? Yes, and it's rather simple. Using our new-found ability to calculate lateral acceleration, CGH, and to determine roll couple, we can predict how many pounds of vehicle weight will be transferred in a turn.



The rear-roll couple requirements are determined by many factors, including tire type and size, CGH, road surface conditions, but the most important factor is vehicle weight and weight distribution. This strange component is an *adjustable* rear sway bar on a front-wheel-drive racer. Overall vehicle weight is so low that as the fuel tank empties during a race, the rear roll couple must be reduced (the driver moves a lever mounted next to his seat) to maintain stable handling.

Photo by Toni Cortes

Here is a classic case of oversteer; the rear end has lost traction and is spinning out of the turn. Oversteering tendencies are more hazardous—and usually require greater driver correction—than understeer. A car that easily oversteers (because of too much rear roll couple) can spin out with even the slightest over-correction. Because a neutral chassis does not exhibit a strong tendency to either over or understeer, it is often the safest and fastest on the track and the easiest on the driver.

This comparison illustrates that even mild suspension tuning can substantially improve handling. The Barracuda on the left is a stock 1965 model without sway bars. It was clocked through this turn at 65 mph. The Cuda on the right has been mildly prepared, by adding good shocks and a sway bar. Despite the fact that the slightly modified vehicle is *traveling 10 mph faster*, it is more stable and more easily controlled (note the substantial increase in the tire contact patch on the inside front wheel).

Again using the Barracuda as an example, we determined that in a static position, the front tires each carry 832 pounds, the rear tires carry 768 pounds, the roll-couple distribution is 69/31, and it took 12.5 seconds to circle the skid pad on the fastest lap. This gives a "G" figure of 0.785. Weight transfer is given by the formula:

FORMULA 20

$$\text{Weight Transfer} = \frac{\text{WTtot X Lateral Acceleration X CGH}}{\text{Track of Vehicle}}$$

Plugging in the values:

WTtot—3200 pounds
Lateral Acceleration—0.785
CGH—31.8 inches
TRACK—55.6 inches

The data reveals that the Cuda is transferring 804 pounds in a turn. Since we know that the roll couple distribution is 69/31, this means that the weight transfer is 555 pounds for the front (69%) and 249 pounds for the rear (31%). Using the static wheel weights (measured with scales), we can determine that for a left-hand turn with a lateral G of 0.785, the actual weight on the wheels would be: 277 pounds on the left front (the original weight of 832 pounds less the weight transfer of 555 pounds); 1387 pounds on the right front (832 plus 555); 491 pounds on the left rear (768 less 277); 1045 pounds on the right rear (768 plus 277).

Most imported cars have been designed for the European market, which is a segment more interested in handling than smooth ride. Until recently, domestic suspension design was directed towards a pillow-soft ride with only secondary consideration given to handling performance.

USING ROLL COUPLE

Applying suspension dynamics theory to the "real world" is our ultimate goal. And the world doesn't get much more "real" than during the experience of slamming into a turn at a race track or autocross. If you find the rear end coming around, the car is

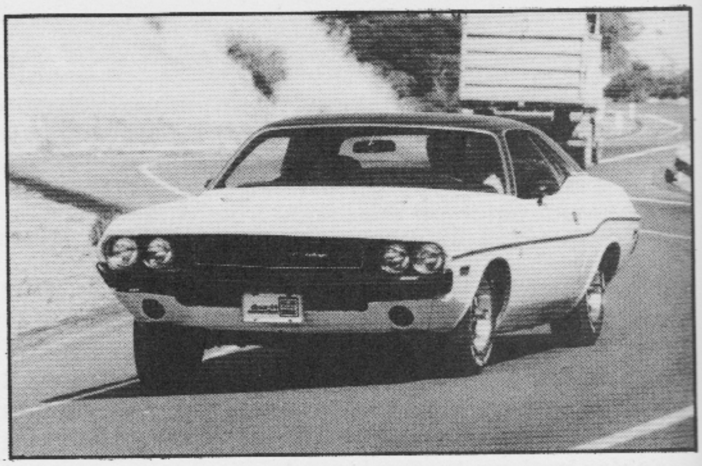
Oversteering caused by rear-brake lockup when overtaking a slower car too quickly. Note the use of a shock absorber as a traction device just in front of the Dart's rear wheel; correct spring rate and proper geometry is a much better method of controlling rear suspension movement during acceleration.

Photo by Toni Cortes

oversteering ("loose"); therefore, theory states that a higher percentage of front roll couple is required. This will transfer weight from the rear wheels to the front, reducing the load-carrying requirements of the rear tires and improving rear traction. However, if the car is "pushing" or the front end slides first, then the car is understeering and a lower front roll couple is required. Lowering the roll-couple percentage will reduce the vertical loads on the front tires, improve traction, and reduce "push." The theory predicts that the front roll-couple percentage may be increased by using a larger front anti-sway bar, or by lowering the rate of (or removing) the rear anti-sway bar. Decreasing the front roll couple is often accomplished by installing a smaller front anti-sway bar or by adding or increasing the size of a rear anti-sway bar.

At first you may find these corrective measures to be backwards; perhaps you are saying, "Shouldn't you increase the weight on a tire that is sliding?" On a pickup truck with an unloaded bed, the answer is yes!

This 340-powered T/A Challenger has less frontal weight and stiffer spring rates than the big-block Challenger pictured above; the result is flatter and more stable cornering. A big-block Challenger can be made to corner this well, but the extra up-front weight will always offer a slight disadvantage. An A-engine Mopar—like this T/A—can pull nearly .9 G on a skid pad and really open up your eyes on a twisty mountain road.

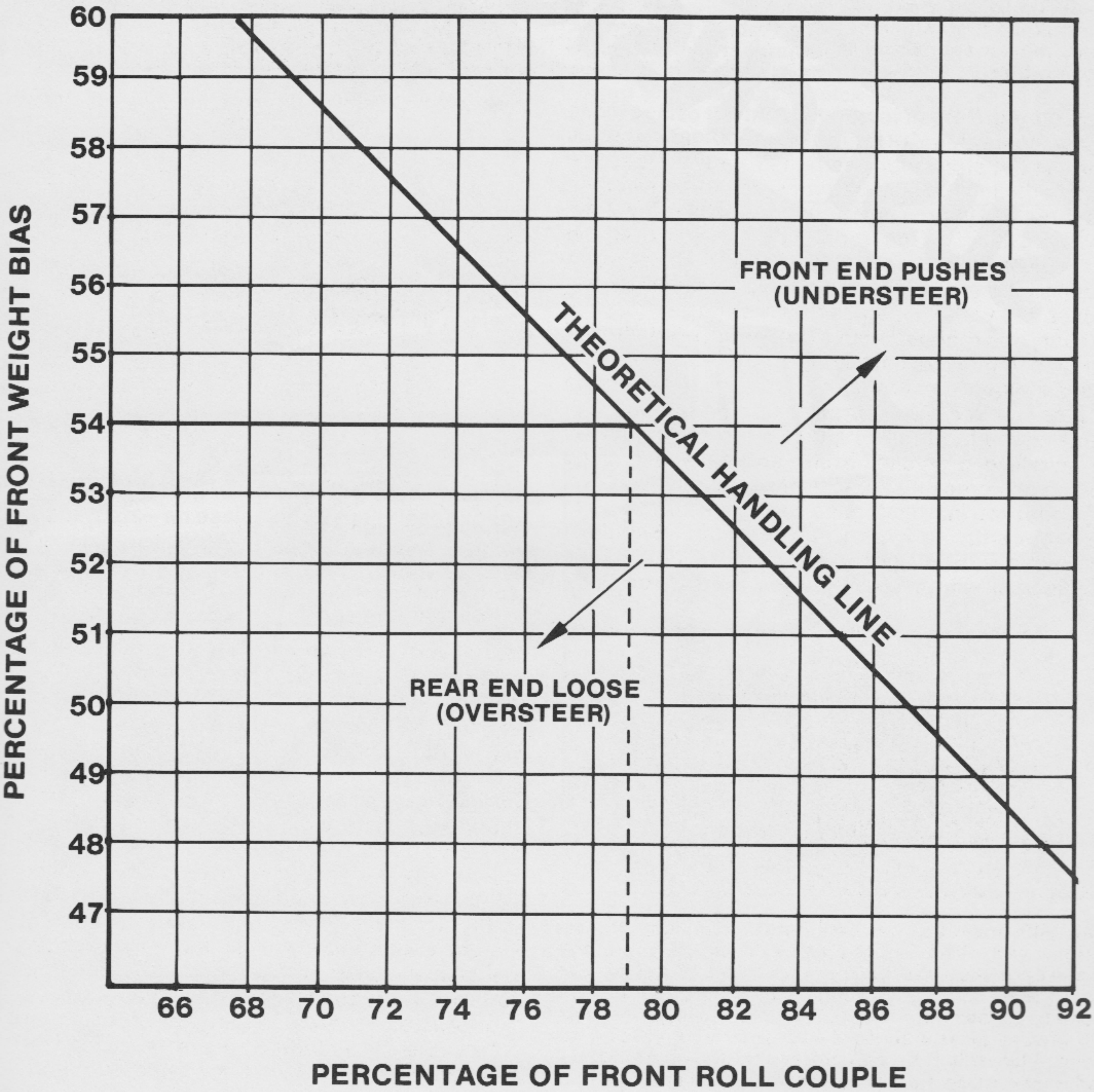


This 383cid Challenger is typical of B-engine Mopars; stock weight distribution and roll couple produce handling characteristics that are suitable for most car owners. However, when the vehicle is pushed into a turn, as shown here, handling gets a little "ragged." Notice the body roll and reduced road contact of the inside front tire.

But we are discussing the handling/dynamic characteristics of a vehicle designed to corner at high speeds. At these speeds, tires are highly loaded and exhibit non-linear characteristics. In other words, at low vertical loads, cornering traction improves as wheel weight increases; but during high-speed cornering, tire stress from both vertical and lateral loads reduce traction, which can only be improved by *reducing* these stress loads; i.e., by shifting weight to other, less-stressed tires. Almost any performance vehicle will easily reach a point where tire overstress is a factor, rather than sliding due to extremely poor weight distribution. And in these cases, our suspension dynamics equations apply: understeer is reduced by reducing the front roll-couple percentage, and oversteer is reduced by reducing the rear roll-couple percentage. (If you insist on road racing a front-end-heavy pickup truck with an empty bed, you're on your own!)

THEORETICAL HANDLING LINE

This graph is often referred to as the "theoretical handling line" chart. It establishes guidelines for determining the optimum front roll-couple percentage, based on the amount (percentage) of weight on the front wheels. The vertical axis plots front weight bias (from 47% to 60%); the horizontal axis contains the front roll-couple percentage (from 66% to 92%). The diagonal line indicates the optimum roll couple for any percentage of front-end weight listed on the graph. For example, if the front end of a particular Barracuda carries 54% of the weight, the front-roll couple requirements will be about 79% to maintain neutral handling. If the roll couple was actually 74%, the vehicle would tend toward oversteer, and if the calculated roll couple was 86%, the Cuda would probably suffer from understeer. Because there are so many variations that can alter the optimum position of the handling line, i.e., track surface, tire type and size, fuel load, etc., this chart should only be used as a guide. But if all testing is carefully done on the same track surface, with the same fuel load, etc., this chart can be very helpful in locating that "magic combination."



CHAPTER 3
SUSPENSION DESIGN

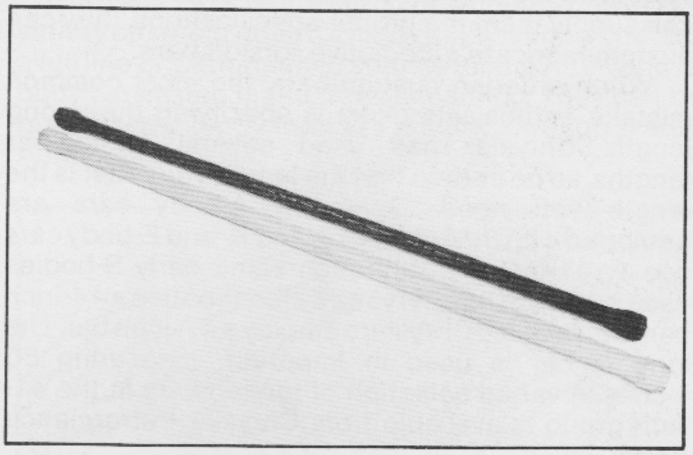
DESIGN AND FUNCTION OF SUSPENSION COMPONENTS

TORSION BARS

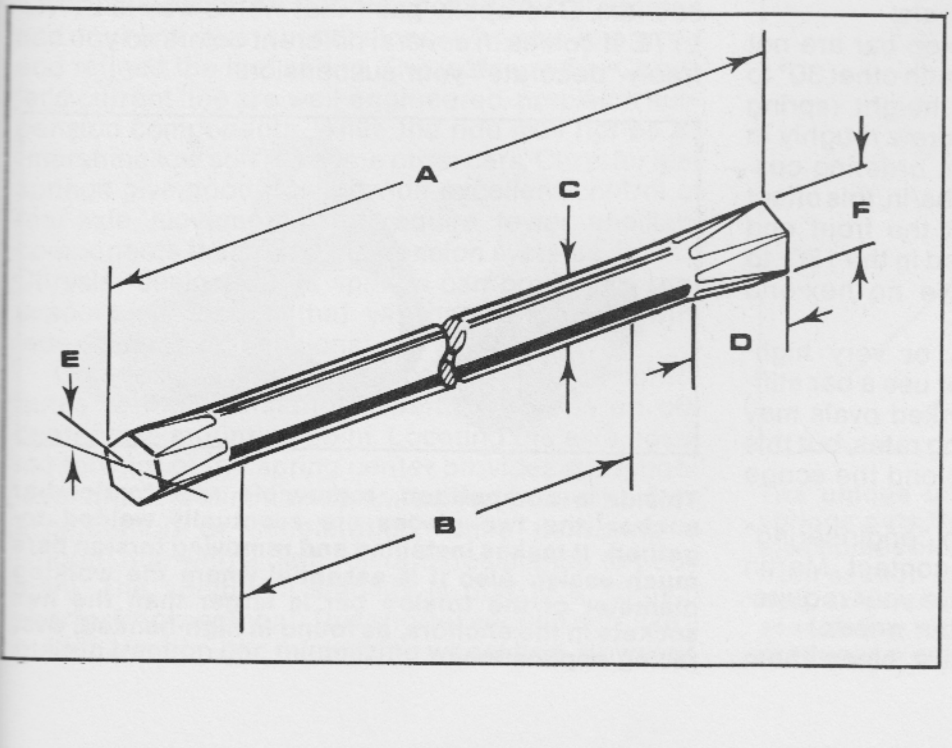
Torsion bars are often described as straight coil springs. This description is quite apropos, even without considering that they are both active suspension elements. In fact, a close examination of a coil spring undergoing compression reveals that it twists similarly to a torsion bar. But despite the similarities, each type of "spring" offers its own advantages and disadvantages. For example, while a coil spring is compact, cheap to make, and readily available, it is also more dangerous to work with, adds to the unsprung weight, and costs no less than a torsion bar. By comparison, the torsion bar requires a more involved manufacturing process, needs a stiffer chassis, and must be custom fabricated in competition rates. However, torsion bars are not part of the unsprung weight of the vehicle (more on unsprung weight later), they are safer and easier to remove/install, and by varying the adjustable tensioner employed in torsion-bar suspensions, vehicle height is readily adjustable.

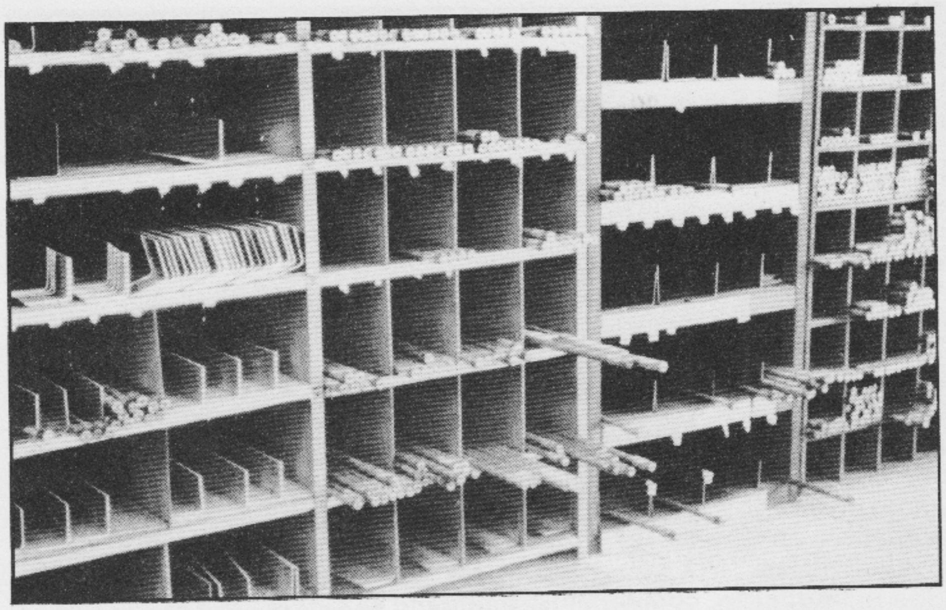
The spring rate of a torsion bar is dependent on its effective length and diameter. A long, thin bar will have a low spring rate. A short, thick bar will have a high rate. Since torsion-bar mounting sockets (the frame-mounted, hex-socket locators) used on Mopars are fixed for a specific bar length, the easiest way to increase the spring rate is to replace the bar with one of greater diameter. Some enthusiasts, however, have fabricated new rear mount sockets, permitting the use of longer bars from different cars; but this modification involves some cutting, welding, and fabrication. Most applications, however, can obtain suitable spring rates with the stock mounts and custom bars—an easier (and usually less expensive) method.

A long-time leader in sprint and midget-car racing components, Halibrand can claim many innovations, such as magnesium wheel (mags) for automotive racing and the "QC" (quick change) rear end. Halibrand is also the industry leader in torsion-bar design and manufacturing.



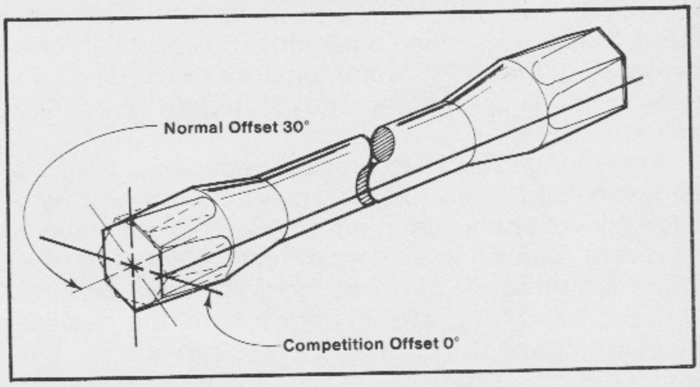
A stock Formula "S" high-performance torsion bar (top) compared to a Martin Automotive street/GT bar. The Formula "S" bar, rated at 108 lb/in, is designed for some performance handling while still retaining a soft ride. The Martin bar, rated at 216 lb/in, provides suspension control and stability for high-performance handling, with "driver comfort" a secondary consideration.





A torsion-bar aficionado's dream. This is one of the storage areas at Halibrand Engineering. It contains several hundred bars in many different rates and lengths, most of which are for sprint and midget cars. (Note the splined, rather than hex, ends.)

The 30-degree hex-end offset in stock torsion bars develop a built-in pre-load that compensates for their relatively low rate. This pre-load produces sufficient spring rate to provide the required stock front-end ride height. However, with competition-rate bars, hex-end offset is not required because 1) the bar rate is high enough to provide adequate ride height without pre-load, and 2) competition cars generally use a lower front-end height than stock passanger cars.



CUSTOM TORSION BARS

A well-known manufacturer of custom torsion bars is Halibrand Engineering in El Cajon, California. They produce bars for sprint cars, midgets, and if you can supply them the proper specifications, they can custom fabricate automotive torsion bars.

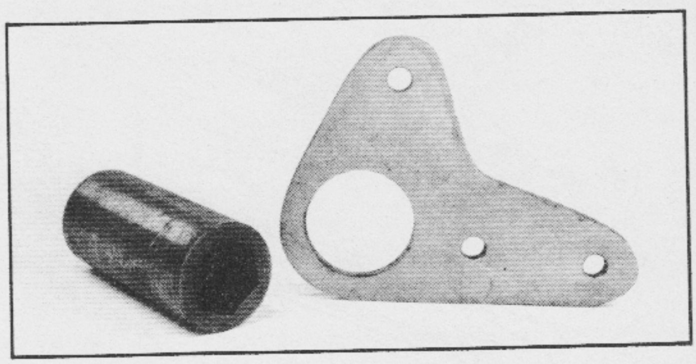
When ordering custom bars, the most common mistake enthusiasts make is specifying the wrong length. Chrysler has used several torsion-bar lengths, so be certain that the length you order is the length you need. Generally, A-body cars are equipped with 35.8-inch bars and B- and E-body cars use a 41-inch bar, although some early B-bodies used a 37-inch bar. Furys and Monacos use a 44-inch bar, and full-size Chryslers employ a 47-inch bar. The longest bar is used in Imperials, measuring 50 inches. A varied selection of torsion bars in the 41-inch group is available from Chrysler Performance Parts.

Two additional notes on custom bars:

1) The hex ends of a stock torsion bar are not aligned; they are offset from each other 30° to provide adequate front-end height (spring preload) with the adjusting screw roughly in the middle of its travel. When ordering custom bars rated at or over 150 lbs/in, this offset must be reduced to prevent the front end from sitting too high. Bars rated in the 180- to 200-lbs/in range should have no hex-end mount offset at all.
2) For road racing, autocross, or very high-performance street use, never use a bar stiffer than 250 lbs/in. (High banked ovals may require the use of higher spring rates, but this specialized application is beyond the scope of this book.)

(If you don't want to go through "engineering" your own high-rate torsion bars, contact Martin Automotive Design. They can evaluate your requirements and fabricate bars to meet your needs.)

Before installing new custom bars, clean them thoroughly with a soft rag and solvent. Then paint them with several coats of epoxy rubber paint. This is important to the survival of your torsion bars because a rock or other sharp object striking the bar can cause surface stress riser, later resulting in a fatigue crack and bar failure (the same process causes connecting rod failure from a nick in the beam section). One epoxy paint that works well is *ZYNO-LYTE*. It comes in several different colors so you can freely "decorate" your suspension.



This is a competition, removable rear torsion-bar anchor (the two pieces are eventually welded together). It makes installing and removing torsion bars much easier. Also it is essential where the working diameter of the torsion bar is larger than the hex sockets in the anchors, as found in high-banked, oval racing applications.

Custom torsion bars represent a sizable investment and should be protected from the elements by a protective coating. Zynolite epoxy paint can protect bars from nicks and dings (and the fatigue cracks that sometimes develop from these surface imperfections) by adding a rugged surface coating. A single can is enough to coat two torsion bars and one front anti-sway bar. It is available in numerous colors, even metal flake.

LEAF SPRINGS

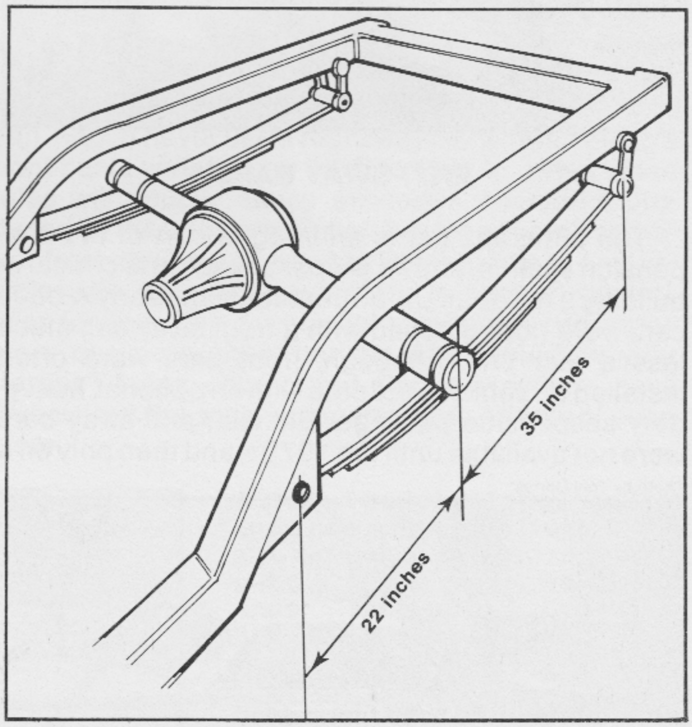
I can confidently say that Chrysler has the best leaf springs designed for passenger-car use. While other manufacturers have switched to coil springs and four-link rear suspensions, Chrysler improved and refined the leaf spring. The leafs used in Chrysler's current line are well-engineered, precision suspension components. While the ride may not be as "marshmallow soft" as some other cars, Chrysler leaf springs give good ride, provide excellent control of rear-axle movement, and require fewer ancillary components than other suspension systems. In fact, Chrysler-designed leaf springs can produce a rear suspension system that will handle on par with independent suspensions.

One design element of Mopar springs that contributes to their outstanding performance is an off-center axle mounting point. Locating the axle housing forward of the spring center provides a stronger lever arm and minimizes axle "tramp" (wheel hop) on acceleration or deceleration, while delivering a smooth ride with a high roll rate (resistance to body lean). The additional stability is due to a 15-inch shorter front segment. This shorter section acts like a built-in traction bar, minimizing wheel hop. However, the rear of the spring is longer and less stiff, giving smooth "seat-of-the-pants" ride.

While on the subject of mounting, there is an added benefit in typical leaf-spring mounts: they help prevent body lean. How? As the body rolls, the springs are twisted in their mounts (somewhat like a torsion bar). Naturally, the leaf springs resist this twisting and transfer a complementary force into the chassis, counteracting body roll. Rear leaf springs are able to provide a ride as comfortable as coil springs while much more effectively resisting body lean.

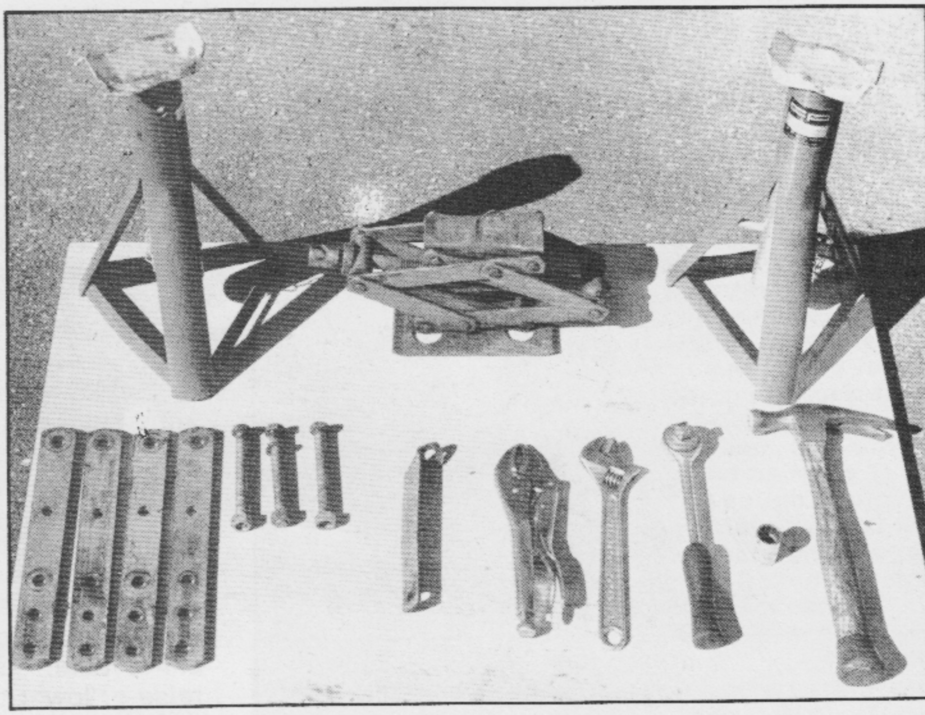
These desirable features (and some others, like anti-squat control, which we will cover later) are optimized by the spring-attachment points. In other words, *do not use long rear shackles.* Lowering the rear mounting (with long shackles) will de-arch the springs and affect chassis dynamics—almost always adversely. This negative effect is caused by a progressive increase in the spring rate as the spring arch is reduced. This means that the spring will get stiffer faster—usually causing instability. If you must raise or lower the rear of your car, there are better ways to do it—lowering blocks, different springs, taller or shorter tires/wheels are all better choices.

Chrysler sells regular- and offset-leaf springs in a variety of spring rates for many applications. There is a full line of performance springs for the A-, B-, and E-body cars listed in the Kit-Car catalog. They can be purchased directly from Chrysler or through Martin Automotive.

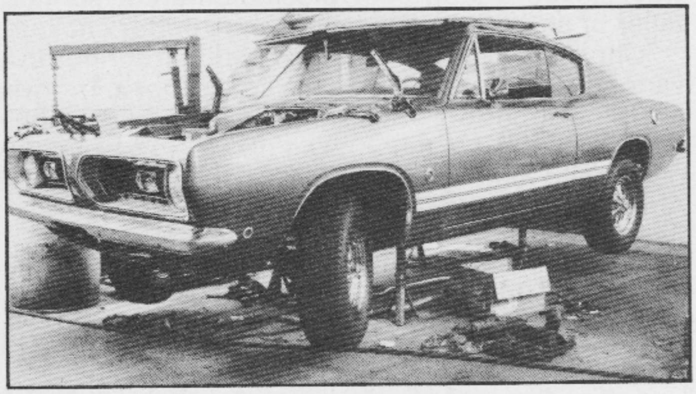


The unique unequal-length design of Chrysler leaf springs puts them in a class by themselves; nothing else comes close in both handling and ride. The short front section acts as a traction bar, preventing wheel hop, axle tramp, and nearly all roll-steer that is usually associated with leaf springs. The long rear section produces a smooth, pleasant ride.

These are the tools required to remove/install A-, B-, and E-body torsion bars. Once the torsioner has been loosened and the retaining clip removed, most torsion bars will easily slide out of the hex mounts. But every once in a while a bar may stick; for these stubborn bars use a clamp made out of long spring shackles (it's the only thing they are good for)—see photos on page 10.



This 1968 Barracuda is undergoing a complete rebuild. The owner chose to use Martin Automotive components to improve handling and a "freshened-up" 340 with a 3-2bbl intake for power. This fine piece of reconstruction is now on the highway; perhaps you'll see it on some twisty back road.



ANTI-SWAY BARS

The anti-sway bar is an integral part of the suspension system, and its proper selection is critical in building a road car. Unfortunately, many early A-body cars were not equipped with a front sway bar, much less a rear one. Although, front bars were often installed on vehicles ordered with an optional, heavy-duty suspension package. But rear anti-sway bars were not available until the 1970s, and then only on a few cars. Now, however, many Mopars are supplied with front—and sometimes rear—bars, either as standard or optional equipment. If you are considering a new car purchase, order the best handling package available on the model you select; you will buy an excellent foundation to build upon.

The anti-sway bar controls body lean by connecting the left and right sides of the suspension together. This connection restricts unequal up-and-down movement in the suspension control arms. As the body of the car tends to lean, the sway bar is twisted, producing resistance to this motion. A heavier (thicker) bar will provide greater resistance to body roll. However, an anti-sway bar is only effective if it is securely anchored to the chassis, and there are two common mounting techniques: 1) To reduce road vibration felt by the driver, street installations normally locate the anti-sway bar in rubber bushings. But the slight "give" in rubber bushings permits additional roll, so; 2) competition applications often use aluminum "pillow" block mounts, or polyeurethane (plastic) bushings bolted directly to the frame.

Sway bars are rated in the force required to

Photo by Toni Cortes

An early A-body Dart rounds a corner in a long-gone street race. The early days of road racing saw more enthusiasm than sophisticated suspension. "State of the art" at the time, the Dart's excessive body lean would relegate the car noncompetitive by today's standards. Stiffer springs and anti-sway bars would substantially improve handling.

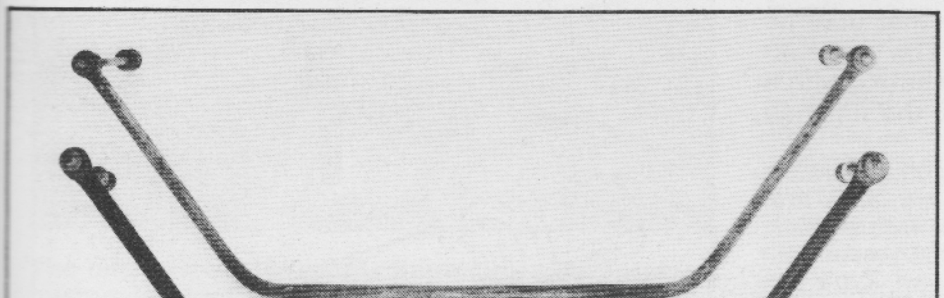
Any amount of oversteer is too much; even slight oversteer combined with a slippery patch of track road can result in loss of control. Oversteer is usually sensed more easily by the driver—steering response begins to suffer—than oversteer, which can "creep up" on even an experienced racer.

deflect them a measured amount. The most common rating method is pounds per degree of deflection—a standard for aftermarket bars. However, the Chrysler Kit Car catalog rates bars in pounds per inch of deflection; so to compare the Chrysler bars to the aftermarket pieces, you must convert the degrees-of-deflection figure into inches of deflection. This can be done by applying the method we developed for calculating transverse-torsion bar rate (see Transverse Torsion Bars in the previous chapter).

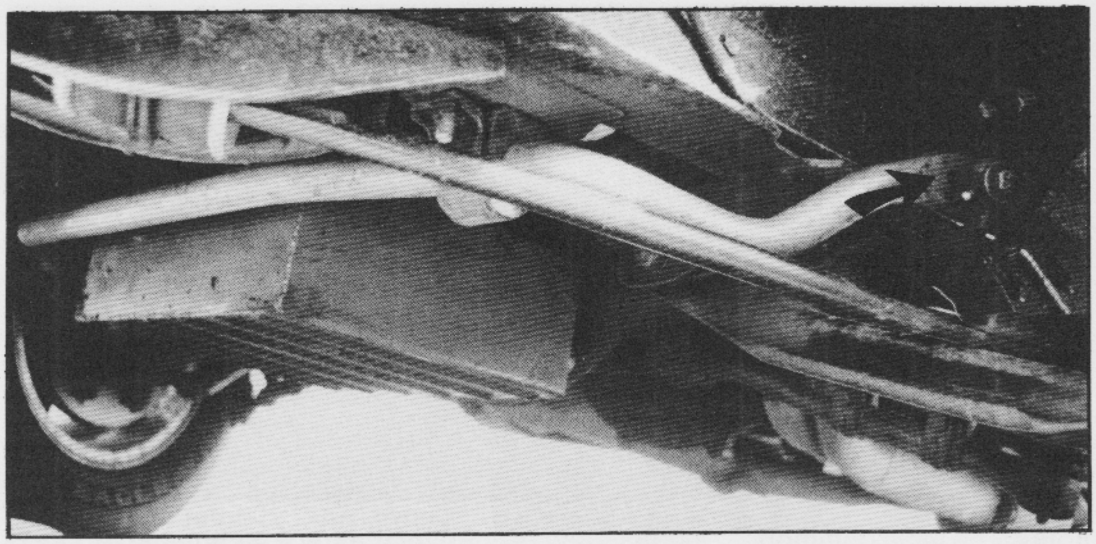
Another important fact to consider in evaluating anti-sway bars is their point of attachment to the suspension system. Most bars are not connected to the control arms at the ball joints, but are located inward. To determine the effective anti-roll capability for this type of mounting, you must determine the wheel rate (Kw) as we did for coil springs in the last chapter. Knowing the static rate of the bar and measuring the control-arm connecting point (relative to the lower-control arm inner mount) will give you the numbers needed for formulas 17 and 19 in the previous chapter. The result will be the true wheel rate for the anti-sway bar.

But what about rear anti-sway bars? One popular manufacturer produces a bar for the A-Body chassis of 0.75-inch diameter, rated at 200 lbs/ in. But the bar is only as wide as the leaf-spring mounts on the rear end. Remember we determined that leaf springs also have unique static and wheel rates that are determined by the motion ratio (spring perch-mounting distance divided by the track width) and formula 19. The difference between wheel and static rates also applies to rear anti-sway bars that mount inward from the wheels. First determine the motion ratio by dividing the distance between the rear end sway-bar mounts (in this example the distance is same as the spring-perch width). Then use formula 19 to convert the bar rating (200 lbs/in for this example) to the wheel rate. If we assume that the leaf-spring mounts and the track width are the same as those in the example in the last chapter, we can calculate the motion ratio (43-inch perch width divided by 55.6-inch track width equals 0.773 MR), then the true wheel rate for the rear sway bar is 119.5-lbs/in (200 x 0.773 x 0.773).

There are two related "rules" for the proper use of anti-sway bars that you should never violate: 1) The front bar should always be more resistant to movement than the rear bar (a higher rating for the front bar). And 2) never use a rear sway bar without using a front bar (a variation on the first rule). Ignoring these rules will usually cause excessive oversteer, produced by the large amount of rear roll couple generated from the heavy rear-bar configuration. Any vehicle with excessive oversteer is hard to control, even for an expert driver; slight understeer is a much more desirable characteristic and should be your "target." Many once-proud Porsche owners have discovered the scary, uncontrollable effects of oversteer.



A stock A-body front sway bar (top) and a competition bar illustrate the size difference required to be competitive. The lower bar would generate substantial understeer unless it was complemented by a rear bar installation. The stock front bar (used alone or with a lightweight rear bar) can improve handling on the many Mopars that were not equipped with any anti-sway bars.



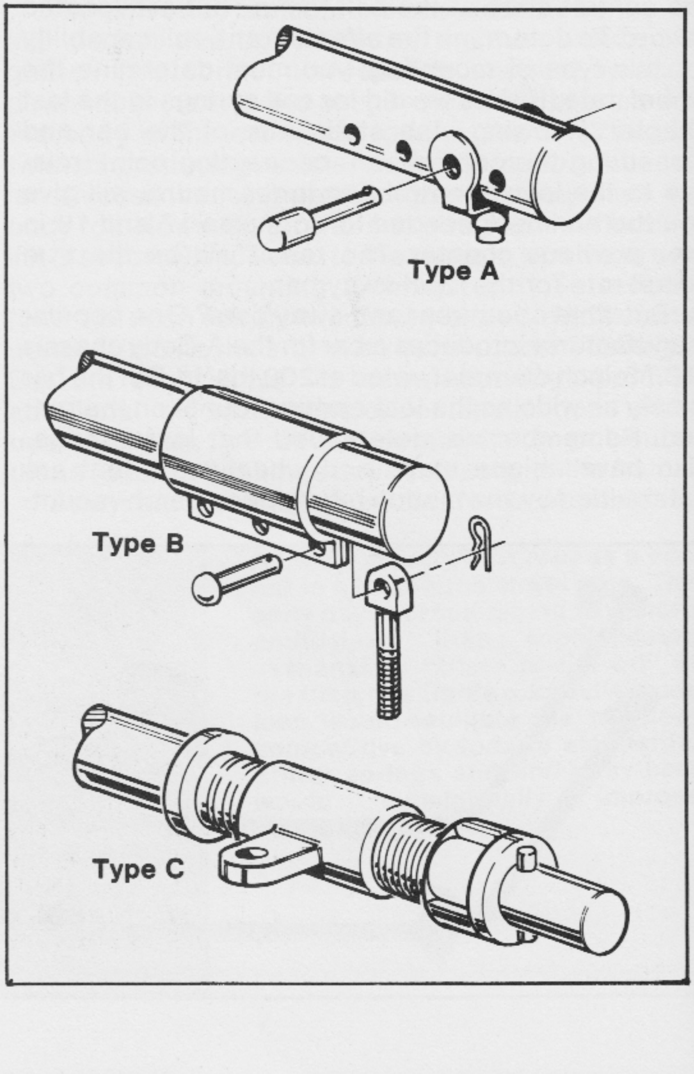
The front sway bar on this Mustang has two holes in the arms for adjusting wheel rate. Even the small distance between these holes can make a considerable difference in both bar rate and handling characteristics. The hole closer to the front of the car (effectively shortening the bar-arm length) will increases the anti-sway rate.

"BUILDING" THE BEST FRONT SWAY BAR

As we have mentioned, anti-sway bars are rated in their resistance to flexing. This factor is dependent on the diameter, effective length, and length of the lever arm (for the standard transverse-mounted sway bars). Most applications require additional roll stiffness in the front end. But bolting on a heavier bar is not always the best answer. Modifying the stock bar can be another solution; done properly, this can increase roll stiffness without adding front-end weight. Modification is often the best choice with A-body cars, because it is almost impossible to locate a sway bar that has a rating of more than 110 lbs/in.

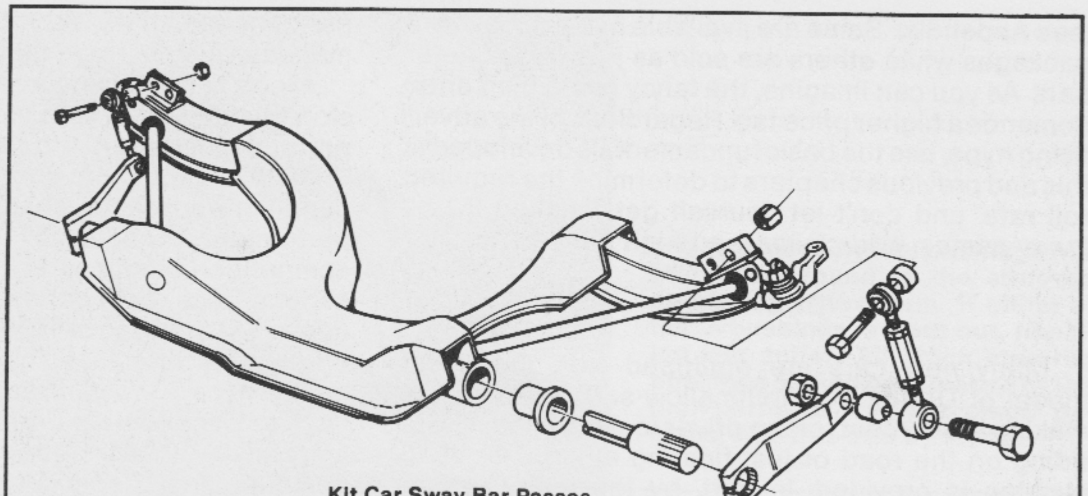
There are only a few practical ways that the stock sway-bar rating can be increased. Reducing the length of the bar will increase the stiffness rate, but this modification is well outside the capabilities of most enthusiasts. The only remaining choice is shortening the length of the sway-bar lever arms. This can be accomplished by moving (modifying) the control-arm link attachment points closer to the sway-bar centerline. When this is done, the entire sway bar is moved closer to the control arms to keep the connecting links in a vertical position (see accompanying photos). The sway-bar roll stiffness will be doubled if the lever arms are reduced to half their original length. The most practical installation employs a movable link connection on the sway-bar lever arm, producing a variable rate that may be adjusted to suit track/road conditions. The sway-bar arms may have to be bent inward (after heating with a torch) to provide wheel clearance; but the finished product is a practical, inexpensive, and lightweight solution to the roll-stiffness problem.

If this modification does not produce the desired roll stiffness, it is sometimes possible to remount the bar even closer to the lower control arms—through a clearance hole in the K-member. This technique often more than doubles the stock roll stiffness. The fabrication, however, is tricky and involves building adjustable brackets that snugly attach to the sway bar, bending the sway-bar ends to provide sufficient tire clearance, and much measuring and patience.



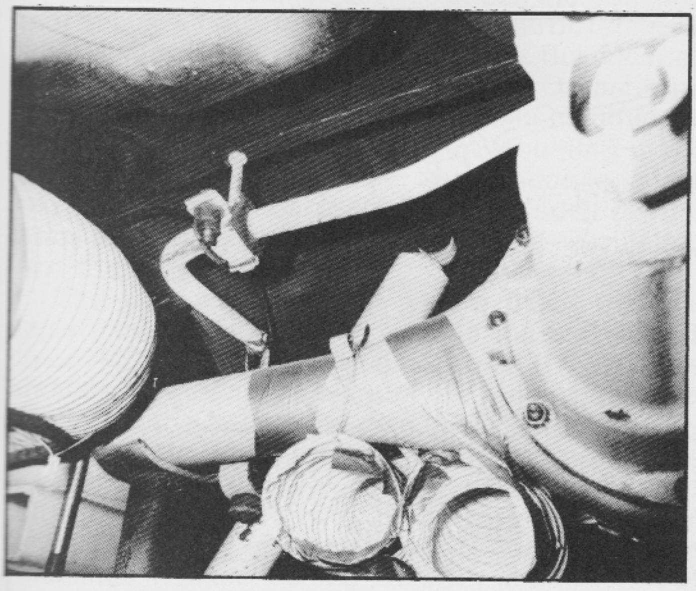
Sway-bar mountings can be modified to incorporate a roll-rate adjustment. Three common methods are: Type A—a simple series of through holes with a quick-release pin, Type B—a clamp that can be moved closer or further from the end of the bar, and Type C—threading the bar (or adding a threaded collar over the bar end). Type C is the most sophisticated and the the most difficult to fabricate. A note of warning: any holes drilled in the bar must be accurate (correct net-fit size); holes that are too large or spaced too closely together will severely weaken the sway bar.

Though expensive and difficult to install, the Kit-Car bar is the most professional sway-bar system. The solid end connectors ensure instantaneous response; however, for street use, solid-link ends should be supplemented with eurethane bushings.



Kit Car Sway Bar Passes Through K-Member

The very best method to obtain adequate sway-bar stiffness and wide range of adjustment is to install the Kit Car sway-bar package. This system uses a tube installed through the K-member in which the sway bar (in this case, a straight "torsion-like" bar) is located by metal bushings at each end. The bar ends are splined, and a fancy lever-arm and adjustable-link system connects the bar to modified lower control arms. The splined lever arms are easily removed, greatly facilitating sway-bar replacement. These sway-bar kits are expensive; but when you have to have the best, nothing else comes close. They are available in roll rates from just under 200 lbs/in, with a bar diameter of 0.800-inch, to over 700 lbs/in, with a bar diameter of 1.125-inches (WOW)!



The rear suspension of this Barracuda incorporates a custom-fabricated, chassis-mount sway bar. The chassis mounting design minimizes unsprung weight and provides superior adjustment control. To provide sufficient axle movement, the sway bar was formed (note the bend in bar above differential) to clear the rear-end housing.

If all this "modification stuff" is a little more than you had in mind, interchanging sway bars is possible. You can usually swap bars between B- and E-body cars; but since the factory made several "unpredictable" design changes, there are no guarantees for fit. A-body bars cannot be directly interchanged with bars from a non-A-body, although you can swap within A-bodies from 1960 to 1969 models. 1970 and later A-body bars may be interchanged, but several changes made to the front suspension in 1970 preclude bar swapping with earlier models.

REAR SWAY BARS

Installing or improving rear anti-sway bars is often easier than similar modifications to the front end. In fact, after adequate clearance is provided for the exhaust system, brake lines, shocks, and differential, the rest of the job is simple. But there are some choices to make. One of the most basic is the method of mounting the bar.

Rear sway bars are mounted in two ways: 1) on the frame with connecting links to the rear axle, or 2) on the rear axle with connecting links to the frame. While both mounting designs function similarly, frame mounting reduces unsprung weight and often provides more clearance for brake lines. However, frame mounts can interfere with (and require the repositioning of) exhaust pipes that are routed over the rear-end housing. If your Mopar is already equipped with a rear bar, it is probably mounted on the rear axle. Although frame mounting can result in a slight performance improvement, it is often not worth the relocation effort. But if you are installing a custom rear bar (or replacing the factory unit) and you are not "put off" by some fabrication work, you should consider frame mounting. Both rear-axle and frame-mounted rear bars can be modified to use adjustable links (similar to the front-bar modifications discussed in the previous section) permitting easy chassis tuning with minimum cost.

Rear sway bars are available from many sources (see Appendix). Some are available in fancy boxes or packages while others are sold as just "plain Jane" bars. As you can imagine, the fancy packaging often demands a higher price tag. Regardless of the advertising hype, use the basic fundamentals developed in this and previous chapters to determine the required roll rate, and don't let yourself get "sucked in" by fancy names, wild claims, and high prices.

BUSHINGS

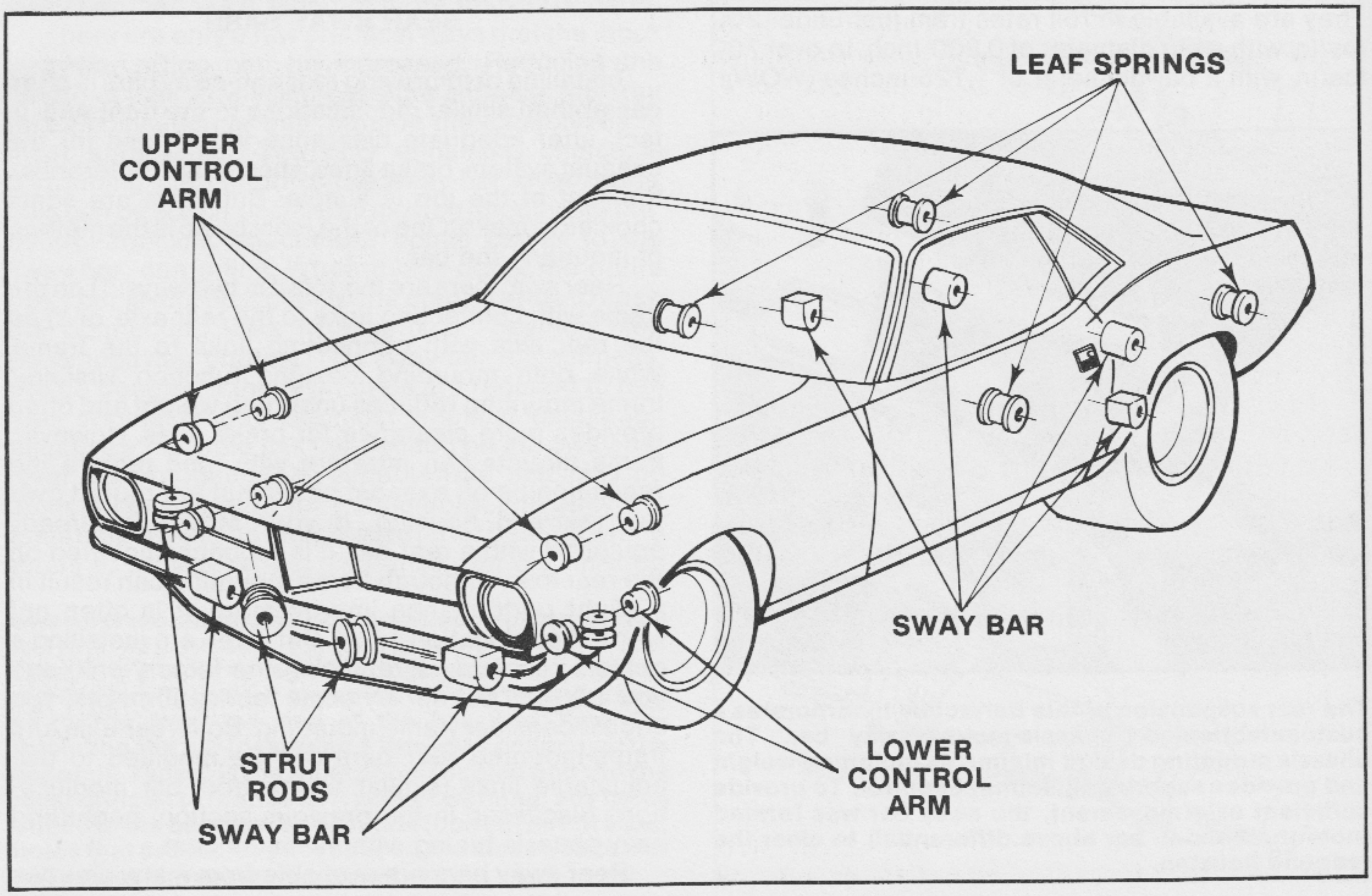
Many new cars are equipped with the latest model of Detroit's "marshmallow-soft" suspension, making it a contest for the driver to determine if he is riding on the road or just floating along. This road cushion is provided, in part, by the many rubber bushings that separate the body from the chassis and the chassis from the suspension. This insulation may be required to please the average driver; but with all perception of "road-feel" absorbed by rubber, many enthusiasts feel that the sensation is equivalent to driving with one eye closed.

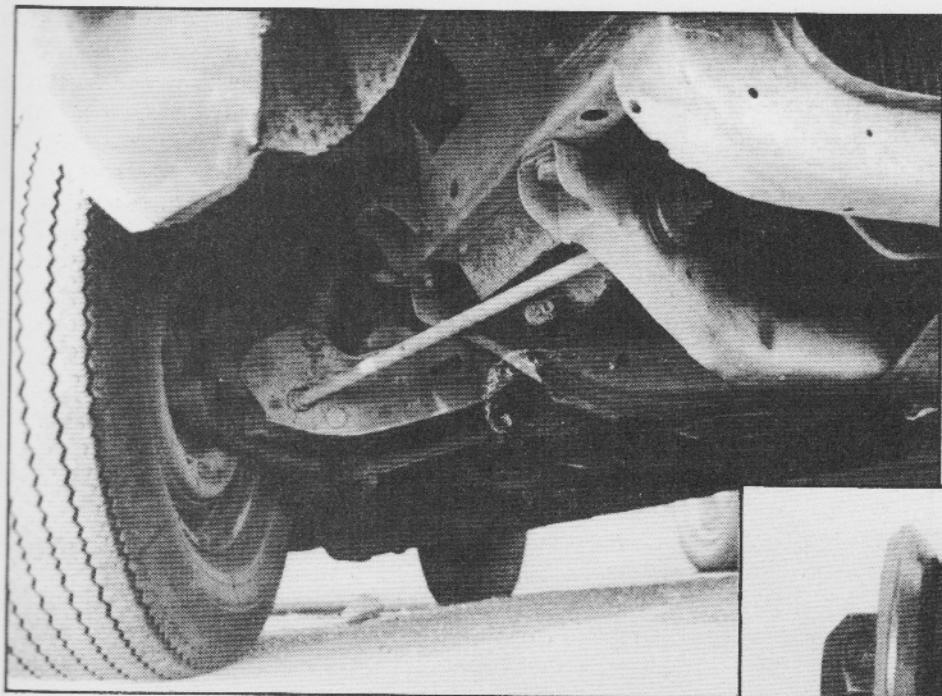
In addition to controlling road vibration, several critical suspension bushings must retain optimum geometry as components move. The quality and firmness of these bushings are very important, because the maintenance of suspension alignment will not only improve handling but also reduce tire wear. The most important bushings used in Mopar suspensions are the spring-eye, control-arm, and strut-rod bushings. Replacing all of these bushings with high-performance parts should be considered virtually mandatory to improve handling and road feel.

Moog and TRW make an excellent line of suspension bushings for Mopars. They offer parts for high-performance street and off-road use, plus a line for "all-out" competition. For typical high-performance handling applications, use Moog parts rated "high performance," especially for strut rod bushings. The competition bushings are too stiff for anything but serious race applications. If your local supplier doesn't carry Moog or TRW, or if they try to sell you an off-the-shelf item that isn't rated "high performance" in the Moog cross-reference chart, shop around awhile. Compromise parts give compromise handling and ride. (Martin Automotive Design can supply a complete line of specially selected Moog and TRW suspension parts for Mopars.)

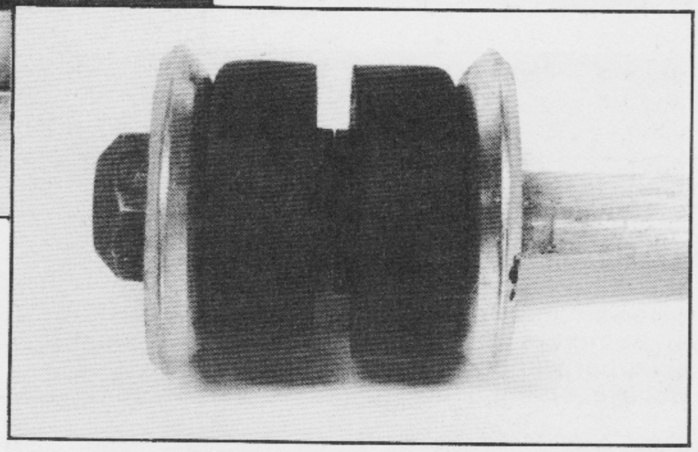
There are a few custom bushing makers that can fabricate bushings to your specifications out of *Delrin,* nylon or very hard rubber. Most often, bushings of this type are custom made only for leaf-spring eyes, because upper and lower control arm

Rubber bushings clearly play an important role in both front- and rear-suspension design. Only the major bushings used in Mopar suspension are illustrated here; others include shock, idler-arm, TV-bar arm, engine mount, etc. The condition of these rubber bushings directly affects handling, ride, and safety.





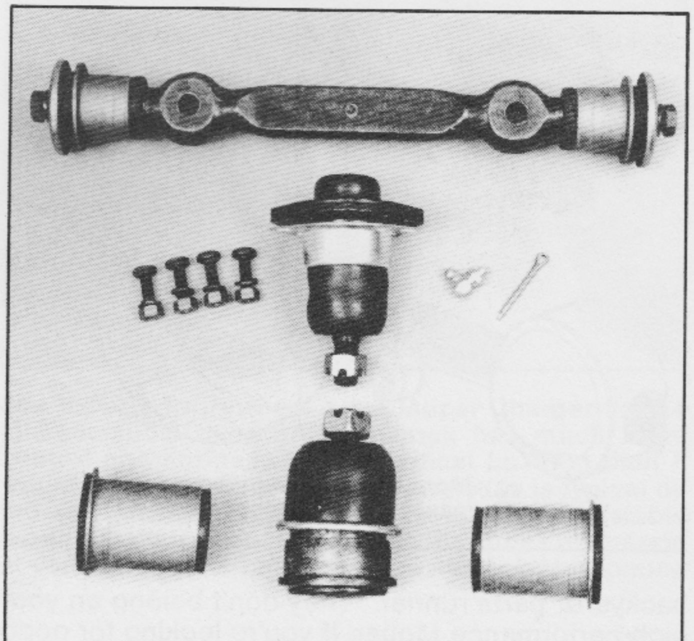
The two most critical bushings in a Mopar are found in the strut-rod and lower control arm. If either of these bushings is worn out, handling will suffer. All stock strut-rod bushings should be replaced by the high-performance, two-piece designs, like this one by Moog.



bushings are specially shaped and much more difficult to manufacture. If you decide to go the custom approach, be sure to provide a new (and correct) replacement bushing as a sample, to ensure that the dimensions on the final piece are exact. Remember that aluminum-alloy replacement bushings are a race-only part. They transfer a very harsh ride to the driver (and to the car body, evidenced by all the sheet-metal screws that will shake loose). And note that the rubber bushings in the upper and lower controls arm also act as bearings for arm movement and *should not be replaced with solid metal parts.*

The rubber bushings used to attach the front and rear anti-sway bars to the frame can also be replaced with high-performance or custom parts. But because of the limited road vibration that is transferred through sway bars, steel spherical-joint mounting is practical. However, spherical-joint mounting is expensive and requires special frame mods to locate the threaded section of the joints. In addition, the sway bar must be modified to permit the articulating end to grasp the bar.

The least critical bushings are those used to isolate the body of the car from the chassis. Mopars used this type of cushioning only on 1974 and later models. The same rules apply in modifying these bushings: the harder the compound, the better the handling (and the more likely your teeth will rattle loose). However, never completely remove these bushings without replacing them with dimensionally-identical rubber or aluminum pieces. Without the bushings in place, bolting the body in position may induce chassis stress that will cause suspension misalignment.



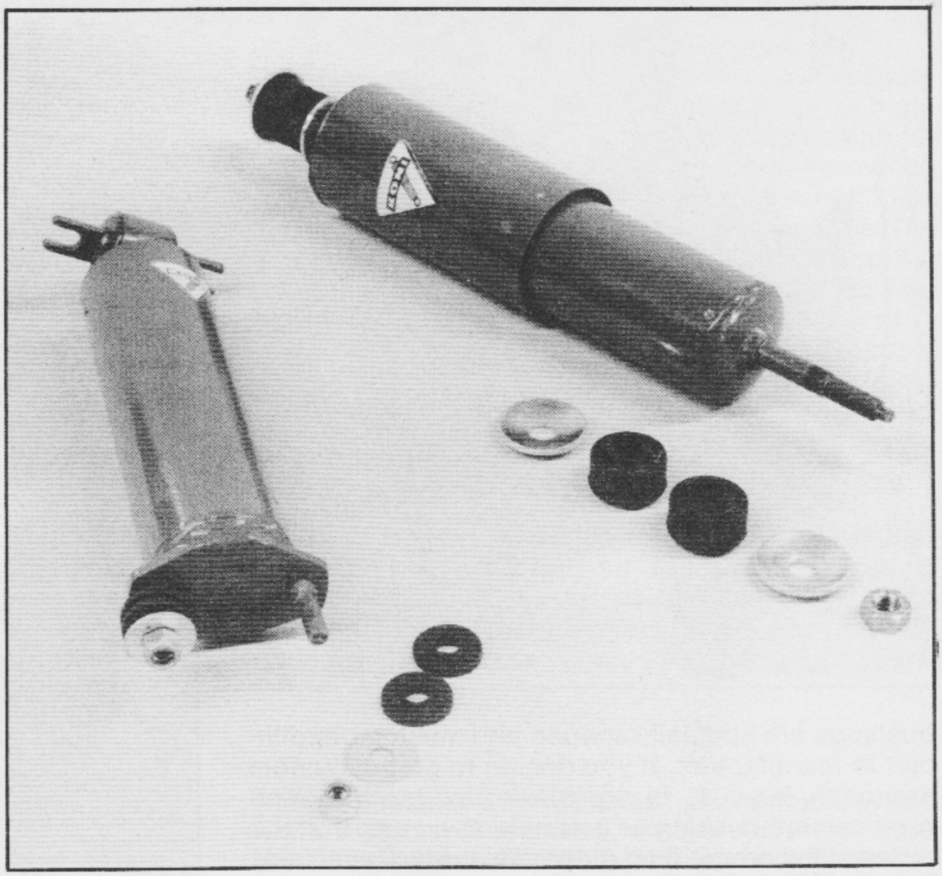
Late model Mopars use these types of bushings and ball joints (similar to Ford and GM), while earlier (pre-1973) models used a slightly different design. But regardless of design, if all of these components are in top shape, your car's handling will be too.

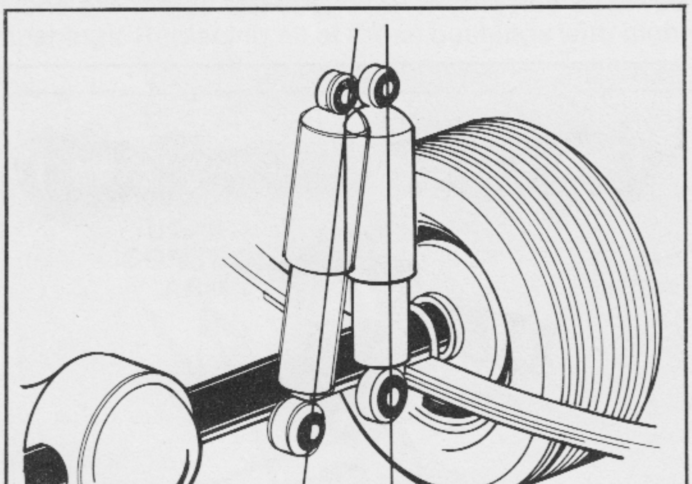
SHOCK ABSORBERS

If you buy $4.00 shocks at a department-store sale, make sure that you install them on your old

The Koni is probably the best shock you can get for your Mopar, and they are available for all models. While the initial price may seem high, you could buy five sets of bargain shocks before you wear out one set of Konis. They incorporate an adjustment that allows you to match shock valving to spring rates and even permits compensation as the shock wears.

The location and mounting angle of shock absorbers have a major effect on vehicle stability and handling during jounce and rebound. When mounted closer to the wheel and/or more vertical, the shock absorber will provide greater control over spring action.





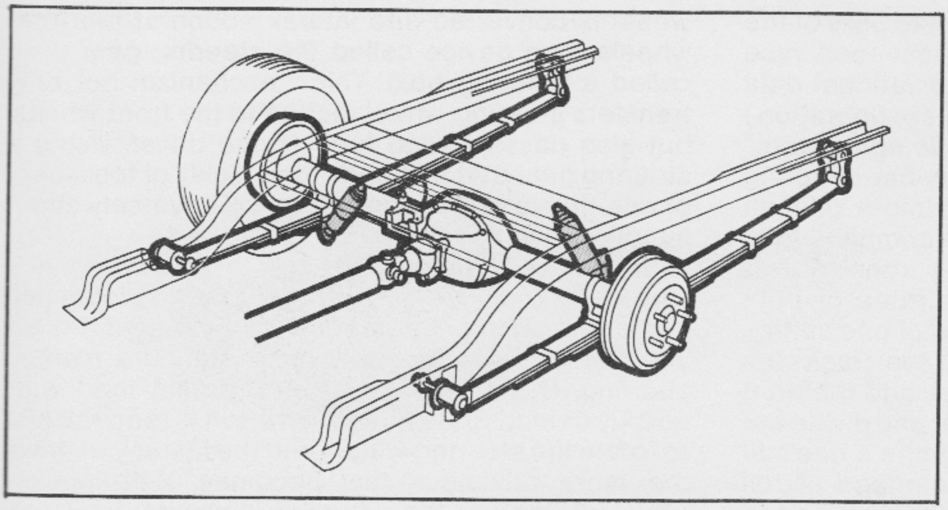
More Vertical Mounting Increases Shock Effectiveness

backyard "parts runner." They don't belong on your high-performance Mopar. If you're looking for good handling, you have to use good shocks. It's that simple! And good shocks are made by KONI or Carrera. You can even have Carrera make a set of shocks to *your specifications* (more on shock selection later). Brand-name performance shocks have a "spec sheet" attached to the shock or carton. This vital information contains the shock ratings that are essential in chassis tuning. Don't buy shocks that are not supplied with this information; you can't figure out where you should be if you don't know where you are.

Shocks are rated by their ability to resist motion both in jounce (compression) and rebound (extension). When this resistance is the same in both directions, the *proportion* rating is 50/50. However, other common proportion values are 45/55, 70/30, etc. For most handling applications, a 50/50 shock is preferred, because it will give equal control in jounce and rebound, reducing the possibility of "strange" oscillations that can occur in hard cornering. Drag-race cars, however, require a very different shock; here a 90/10 design is often used to encourage the front of the car to rise up—and stay up—during acceleration for improved weight transfer. This is definitely *not* the tact to take for cornering, because the reduced roll control in the front suspension will permit excessive weight transfer to the outside wheel, inducing generally "squirrelly" handling.

But there's more to it than just rating. Shocks differ in their compressed and extended lengths. For a car of stock height, you can normally use the stock-length shock. However, if the car has been lowered for better handling, you may need a shorter shock. Make absolutely sure that the shock will not bottom out when the suspension is compressed. You can ruin a new set of shocks very quickly if the fully compressed length is too long.

Shock absorbers are a "mass-market" item and share the advantages (low price) and disadvantages (advertising verbiage) of this type of product. (A recent ad touts "computer-matched shocks." This tricky advertising lingo really means that the shocks have been matched to the rate and frequency of the



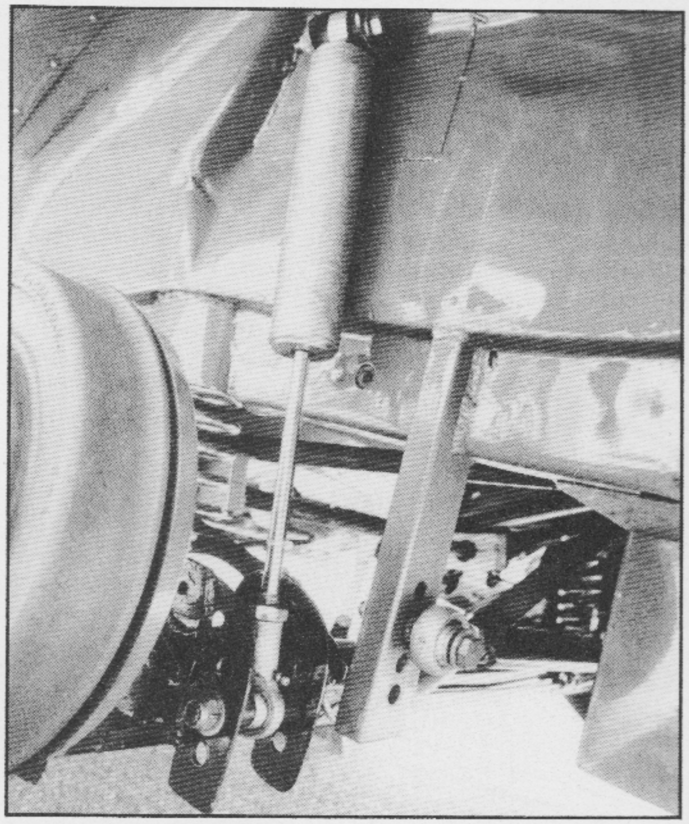
Rear spring and shocks work together to provide stability over bumps and during acceleration, deceleration and cornering. If very stiff shocks are used with weak springs, the shocks will overheat and soon fail. Select a shock that will control twice the static rate of the spring (see text), with this rating it will proportionally share the load with the spring and both optimum shock life and handling will result (for the particular spring rate being used).

springs they are designed to work with.)

DETERMINING SHOCK REQUIREMENTS

So now that you are baffled by all the advertising claims, it's time to reveal two basic shock-absorber facts that you can live by. 1) Select shocks by the load-absorbing rate that closely matches the load-carrying ability of the spring. This means that stiffening the ride on a weakly-sprung car by using very-firm shocks, rather than *improving the rates of both the springs and shocks,* will result in poor handling and rapid shock wear. Using light shocks to compensate for stiff springs is just as undesirable because spring motion will not be adequately damped, once again causing poor handling and premature shock wear. The suspension can only function properly when the shocks are able to control and share the load with the springs. Which brings us to fact two: 2) the closer the shock is mounted to the wheel and the more vertical the mounting, the more the shock will control the spring. To control the same wheel force, a shock absorber that is installed at a steep angle must be much stiffer than its vertically-mounted counterpart. Fortunately, stock Mopar shock installations are nearly optimum and require little modification, especially on the front end where the greatest control is required.

Since optimum suspension control is achieved by matching the load-absorbing potential of the shock with the load-carrying ability of the spring, let's look at some hard facts. A good rule of thumb for shock selection is to locate a shock that can control twice the static spring rate in pounds. For example: with 180 lbs/in rear springs, the shock rate should be 360 pounds (doubling 180). This can be considered an approximation of the force (on each wheel) that the shock will have to control to proportionally share the load with the spring. This same technique is applicable to front shocks: for torsion bars with a static rate of 200 lbs/in, shocks of a 400-pound load-absorbing capacity should be selected. Perhaps you are wondering how this method was developed and if it is always applicable. The first answer is "experimentation"; and the second is "not really." Read on.



Joe Varde's front-wheel drive Mopar Charger uses a custom shock mount. The shock has much more control and stiffness in this vertical position than it would if it were mounted at an angle (as is typical on the rear of most stock cars). Note: The adjustable mounting on the rear axle allows the shock to operate at various chassis-height settings without bottoming out.

CHRYSLER'S EMPIRICAL METHOD

Someone once said, "Just use what works!" Not bad advice, but how do you determine what works? Well, if you've been racing cars for many years, you eventually find out what works by trial and error. And Chrysler used their experience gained from circuit-car racing to develop a method of approximating shock-absorber requirements. However, this empirical method applies only when the front spring and anti-sway bar each control about 45% to 55% of the front roll couple. (This applies to most road race applications. Chrysler has little observational data for suspension systems outside this configuration.) In order to apply the "twice the static spring rate" rule, we must first determine the sway-bar-to-spring roll-couple distribution. This sounds more difficult than it actually is; basically, it's just a comparison of rates. First add the rate of the two front springs together and divide by two. (If the rates of both springs are identical, just use the rate of one spring. This method was designed to include oval track cars that use staggered spring rates.) Now add the front anti-sway bar and spring rate together and divide the total into the sway-bar rate. The result is a decimal value. To convert this into the percentage of roll couple absorbed by the sway bar, multiply the decimal value by 100.

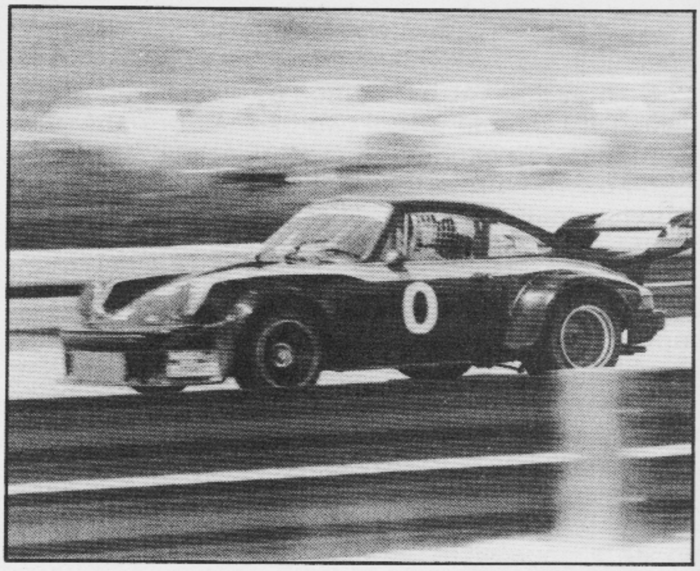
For example, a 1968 Dart GTS with heavy-duty suspension uses torsion bars rated at 108 lbs/in and a sway bar rated at 110 lbs/in. Adding the two rates together gives 218 lbs/in. Dividing the total into the sway-bar rate of 110 lbs/in produces a decimal value (0.505), which converts to 50.5% when multiplied by 100. This percentage falls within the requirements of 45% to 55% for applying Chrysler's empirical method, and the "twice the spring rate" rule for shock selection can be used. Therefore, the Dart needs shocks capable of controlling 216 lbs/in per wheel for the front (twice the 108 lbs/in front spring rate) and 240 lbs/in per wheel for the rear (twice the 120 lbs/in rear spring rate).

STEERING GEAR

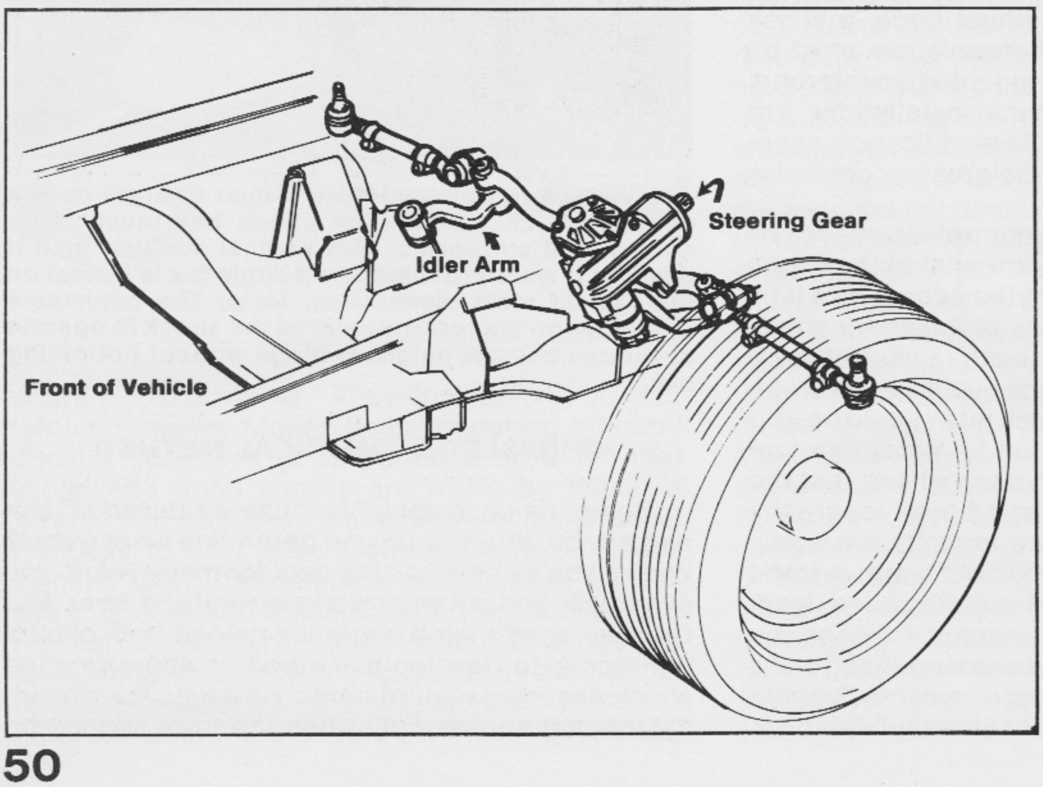
MANUAL

The driver-input rotary motion of the steering wheel is converted into lateral motion at the front wheels by a device called the steering gear (also called a steering box). This mechanism not only transfers steering-wheel motion to the front wheels but also passes "road feel" to the driver. Using a steering gear that responds too quickly or too slowly or one that is worn or defective will adversely affect handling, and can be dangerous.

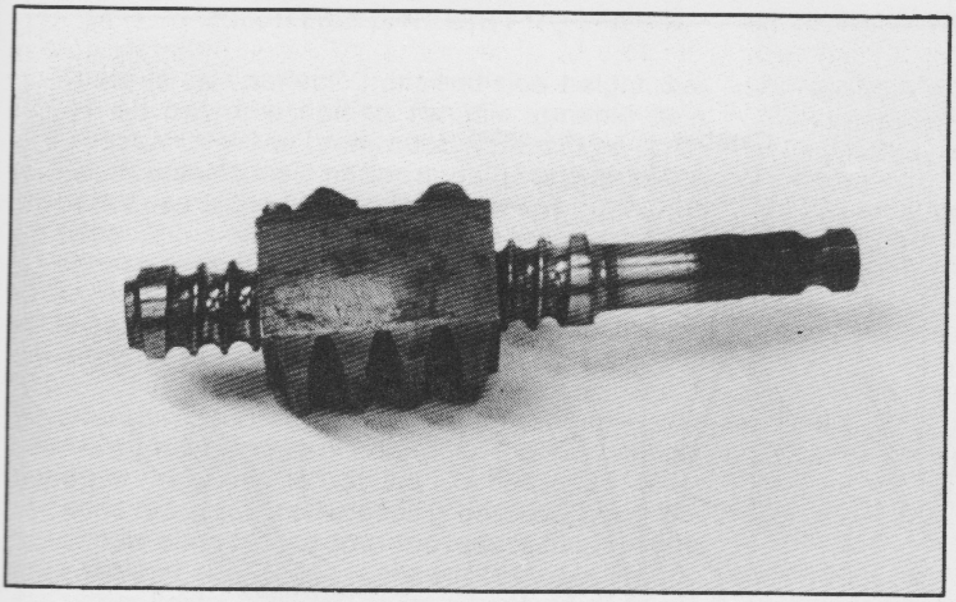
In racing situations, using a high-performance steering box will not only improve steering response but also reduce driver fatigue. You can test this for yourself in a car equipped with standard manual steering. Driving on a twisty mountain road will quickly demonstrate how much effort is required just to rotate the steering wheel; and the faster you drive, the more ridiculous this becomes. With power-assisted steering the situation is similar; however, instead of fatigued arms, you'll surely notice the



Although many sports cars use rack-and-pinion steering, the conventional steering used on Mopars performs very well, even in competition. Steering response can be substantially improved, however, by using a fast-ratio steering gear (from Chrysler P-parts), new rod ends, idler arm, and *a precision front-end alignment by an experienced performance mechanic.*

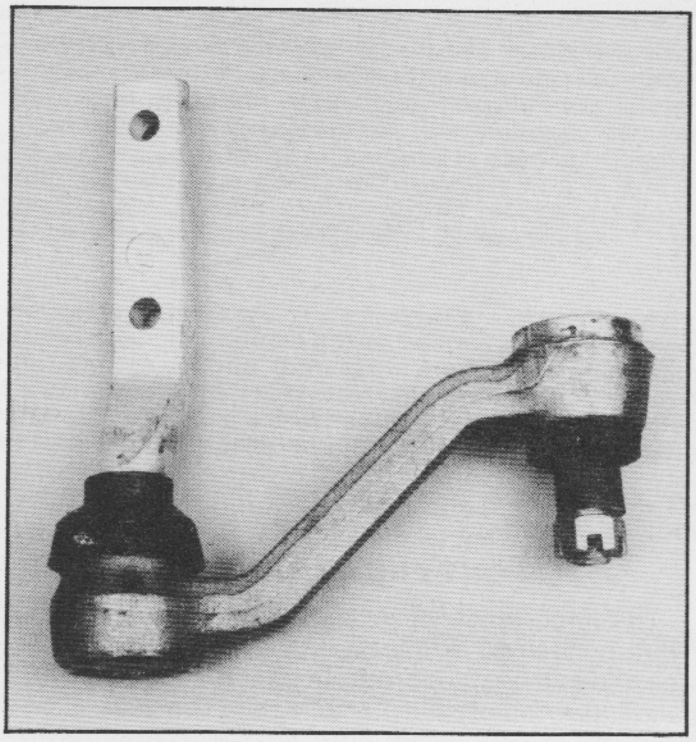


The steering box, linkage, idler arm, and rod ends make up the steering system and—to a major extent—control directional stability. Loose, worn, improperly installed, or damaged parts will quickly diminish this control.



The steering-box internals are together known as the "steering chuck." Technically, these parts form a worm-and-ball-nut assembly (using recirculating ball-bearings in the nut section) that determines the steering-box ratio. This design provides for long life, easy rebuilding, and the use of a variable-ratio worm section.

"Sealed-for-life" idler arms do not have lubrication fittings (or provisions to install any). Some suspension components, although not supplied with grease fittings, do include the holes (with small screws in place) for their installation.



"floating" feeling in the front end—due to excessive response and the tendency for the power unit to "over assist." Effortless steering might be fine for boulevard travelers, but high-performance driving necessitates high-performance steering.

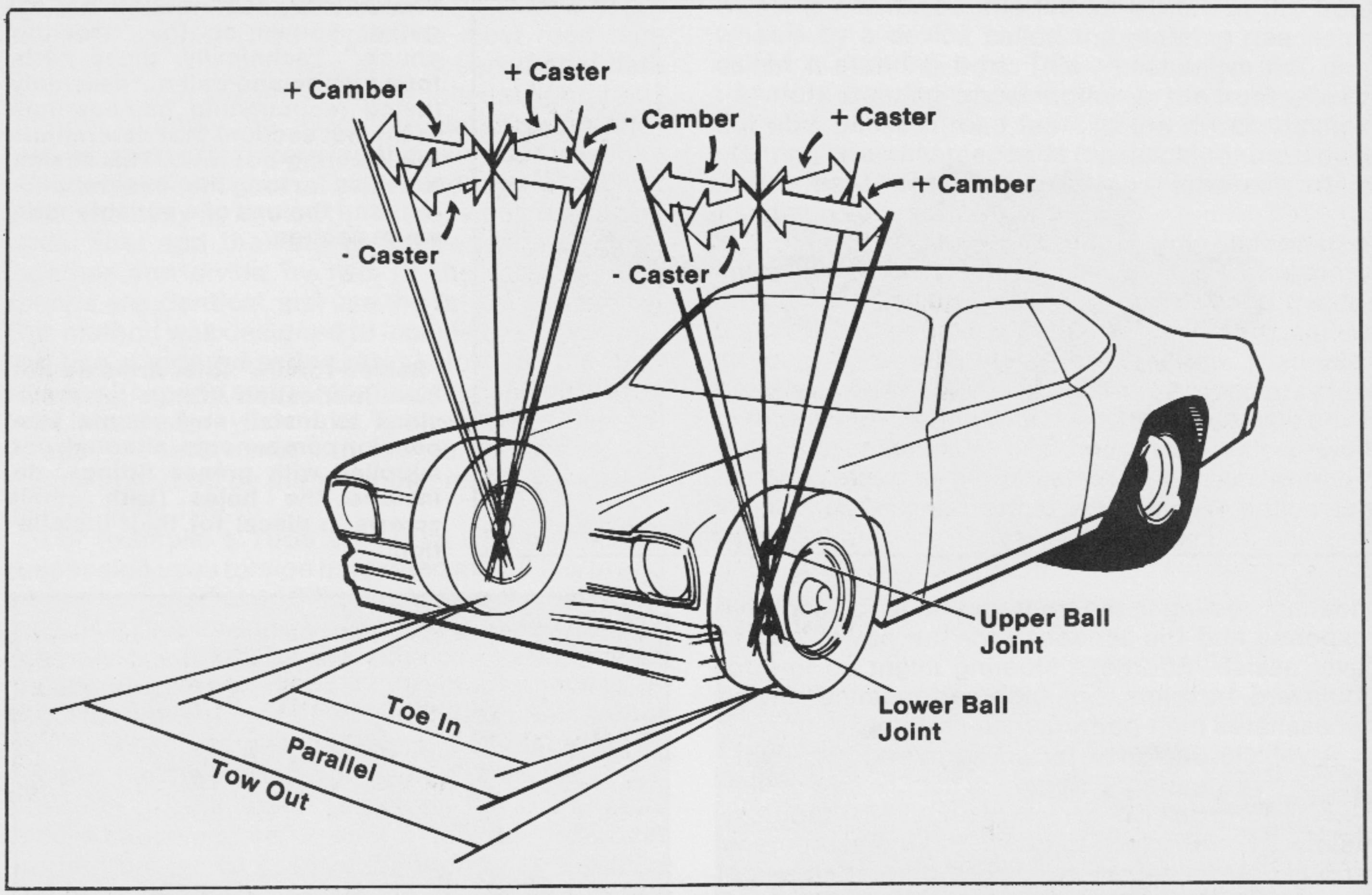
A vehicle designed for cornering needs "tight," responsive steering. This means using a new or like-new steering box with a ratio of about 16:1 (approximately 3.5 turns lock-to-lock). This measurement/ratio is determined by twisting the steering wheel from its full left-turn to its full right-turn position and counting the number of rotations. Many "family-sedan" manual boxes have a turns ratio of 5 or more. Chrysler Performance Parts at one time offered fast ratio manual boxes and quick steering gear parts to convert one of these "sloppy" steering assemblies into a responsive road-race piece. Unfortunately, the 16:1 units are no longer available, but you can still purchase a 20:1 steering box through P-Parts. Remember to realign the front end after changing any steering-related parts. Steering box (and internal gear) part numbers are listed in the Appendix.

POWER STEERING

In the stock configuration, the pre-1979 Chrysler power-assisted steering is not well suited to handling. It is over assisted (excessive turning motion is generated when the steering wheel is rotated), and it transmits too little road feel to the driver. The late-model power-steering units are much improved over the older, pre-1979 designs; but if you have one of the early "slush boxes," its worth the effort to have the valving recalibrated for an increase in steering effort. This modification, however, requires the expertise of a specialized shop—one that prepares race cars or off-road racers. (You might want to take a test drive in a late-model Firebird Trans-Am to see how *good* power-assisted steering feels.) A more straightforward solution is scrapping the power-assisted steering gear/pump unit and installing a quick-ratio manual box.

STEERING LINKAGE

Another vital part of the steering system is the connecting linkage. These appendages include the Pitman arm, idler arm, drag link, and adjustable end links. All steering components should have tight, wear-free ball/socket connections, because any "give" here will reduce steering control. Replace all worn or loose parts with models rated for high-performance use. Moog and TRW have a complete line of suspension components, some of which are designed exclusively for competition/off-road use. The Moog performance line is used and recommended by many racers. Finally, make sure that ball/socket components are manufactured with integral lube fittings. Stay away from the "sealed for



Here the angles formed by the camber, caster, steering-axis inclination (the seemingly vertical line from which caster and camber are measured), and toe settings illustrate the major geometry points that establish front-end handling characteristics. Note that the caster forms a front-to-back angle, with the rearward position referred to as positive, and the camber forms a side-to-side angle, with the outside position as positive; the toe in/out measures the deviation from parallel in front-wheel tracking.

life" variety; their usable life may not be as long as you had hoped.

FRONT-END ALIGNMENT

The importance of proper alignment cannot be overemphasized. A misaligned car will not handle well and can be dangerous. In addition to optimizing handling, predictability, and safety, correct alignment will prevent premature tire wear. Poor alignment can "chew up" tires faster than your girl friend can reach the credit limit on your Visa-Card. Optimum alignment (as opposed to necessarily using factory alignment specifications) often makes the difference between "normal" tire life of 20,000 miles and driving up to 40,000 miles per set. Some tires can last 70,000 miles when the alignment and suspension are complementary; that is, when both the suspension and alignment are designed for one purpose—optimum handling. In this sense, optimum handling requires precision front-end geometry, low rolling friction, proper spring and torsion-bar rates—all resulting in minimum tire wear.

Worn ball joints will ruin the best alignment and suspension set up. Not only will they adversely affect handling, they often cause accelerated tire wear.

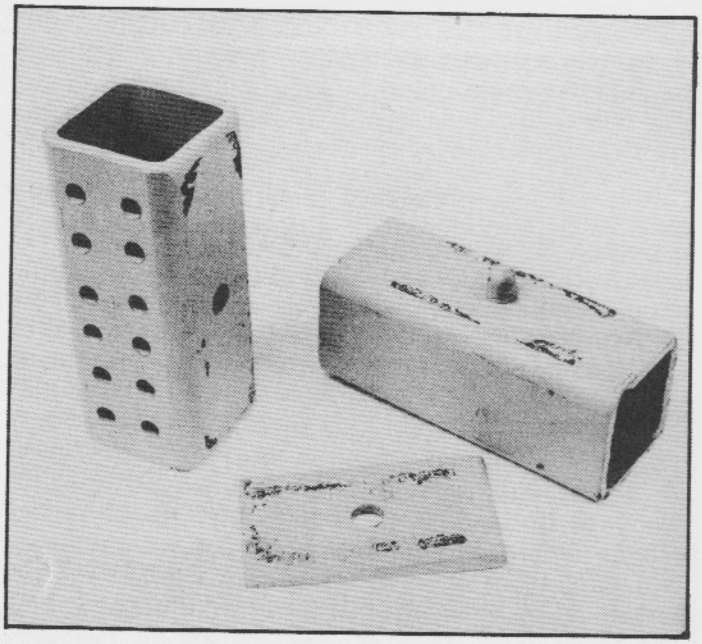
Proper alignment can only be achieved after other modifications have been completed. For example, all new suspension components should be installed, the front and rear ride heights should be set, and the vehicle should be equipped with the tires and rims that will be used under actual driving conditions. If additional changes are made that require front end disassembly or readjustment, *realign the suspension.* This is particularly important for changes in ride height. Remember this rule-of-thumb for Mopar suspensions: altering the ride height *will affect* alignment. And since ride-height modification is often poorly understood (and must be set *before* the front-end is aligned), a short discussion on this subject is in order.

FRONT-END HEIGHT

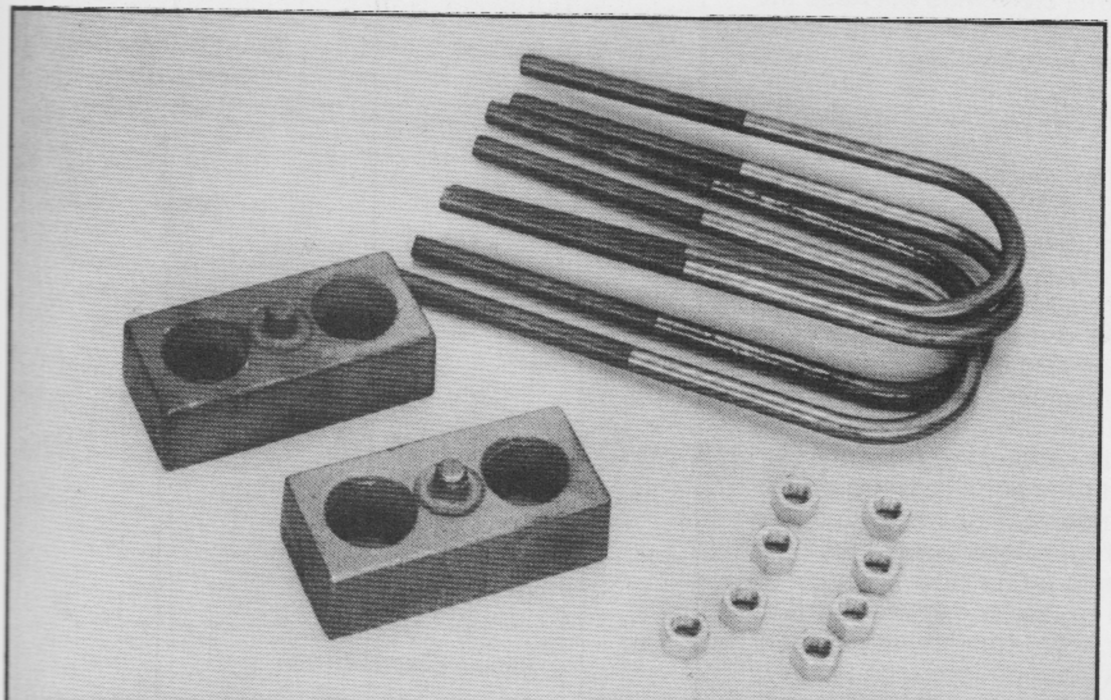
There is no "perfect" front-end ride height, because an optimum setting for this variable is a combination of driver preference and trial-and-error. However, some consideration should be given to the intended *use* of the vehicle. If the car has been built for competition only, as little as 3.5 inches clearance between the ground and the bottom of the K-member is preferred. But if street driving is part of your plan, more ground clearance will be required; 5 to 7 inches will prevent bottoming on dips, rail-road tracks, etc. However, a front-end height of 7.0 inches or more will have an adverse effect on handling. Front-end height should always be slightly lower than the rear, with an optimum "rake angle" of 1.5° to minimize aerodynamic drag. Either increasing or decreasing the rake angle will increase drag and reduce gas mileage (and top-end speed, if you are interested in testing the upper limits). More information on aerodynamic drag will be provided later.

Chassis height can be altered in several ways; some are acceptable, some are "no-no's." A super "no-no" is heating (de-arching) rear leaf springs (or, heaven forbid, torsion bars) to lower a car. Put this modification on your list with Russian Roulette because it can cause stress fractures and loss of control. If you need to lower the rear height of a Mopar, use low-profile tires. Changing from a 70 to a 50 series tire will lower the car by about two inches.

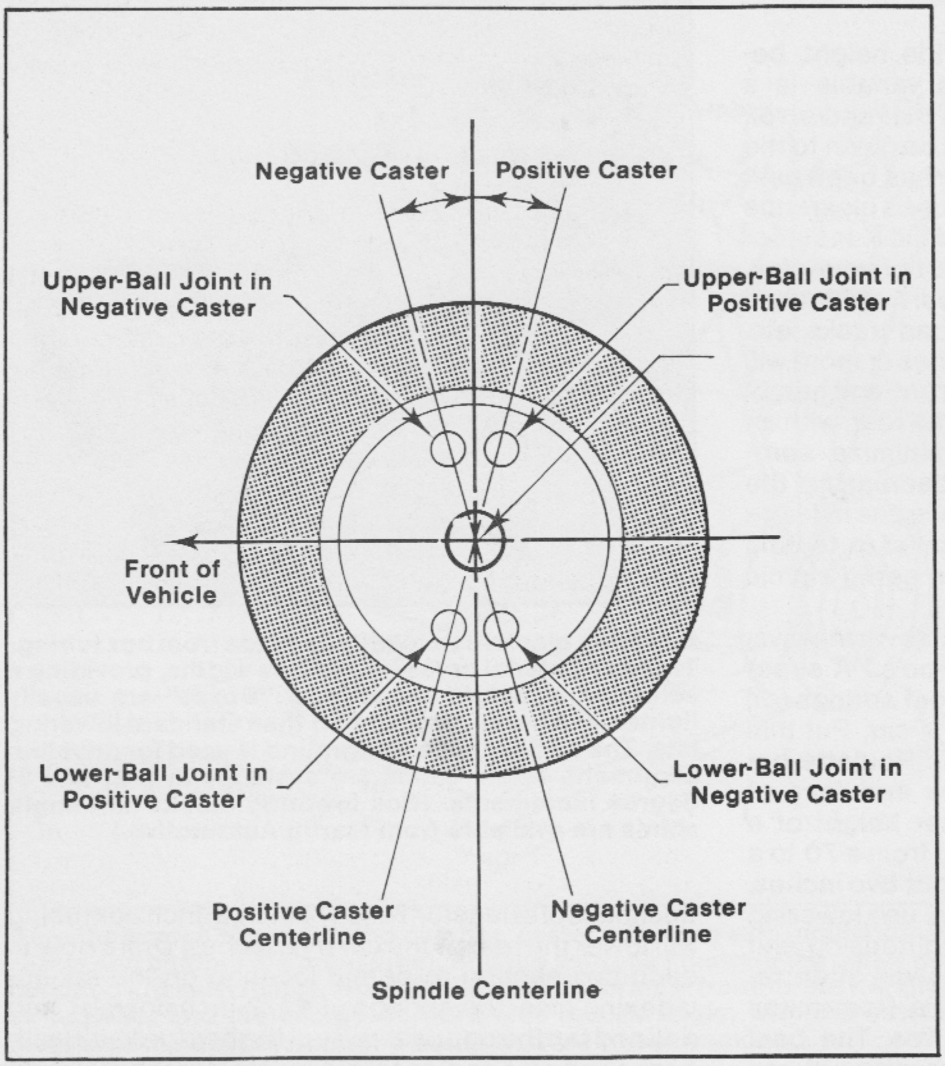
If more height reduction is required, use lowering blocks installed between the rear-end housing and the leaf-spring pads. Installing blocks will often require using longer "U-bolts," available from many spring manufacturers for a nominal fee. The best lowering blocks are fabricated from sections of box tubing. This material is available in various dimensions; but always use box tubing that has a wall thickness of at least 3/16 to 1/4 inch. 2-inch box tubing will lower the rear of the car by 2 inches. Drill a hole in each box section to permit locating on the spring-indexing pins (the springs use 1/2-inch dowels), and drill and tap the opposite side of the box for a bolt (with a rounded-off head) to form a locator for the rear-axle housing. Remember: *only fabricate blocks from steel stock with a wall thickness of at least 3/16 inch.*



Lowering blocks can readily be made from box tubing. The box material comes in various widths, providing a selection of height reductions. "Boxes" are usually lighter (and certainly stronger) than standard lowering kits. The wedge in the foreground is used for driveline alignment. Angle shims are available in l, 2, and 3-degree increments. (Box lowering blocks and angle shims are available from Martin Automotive.)

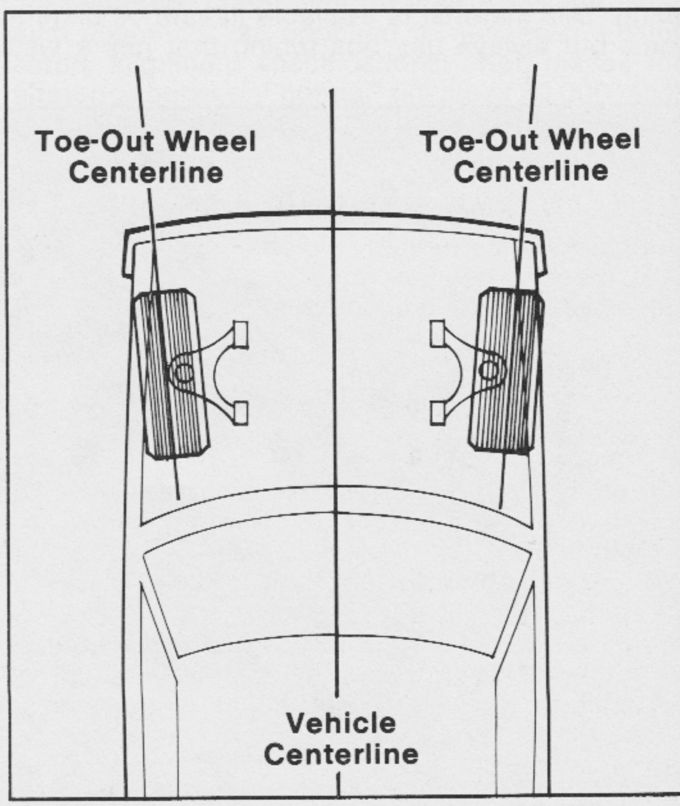


This is the standard lowering kit available at most auto parts stores and speed shops. The blocks (and shims—not illustrated) come in varying thicknesses, from 1/4 inch to 3 inches. They provide a simple and effective way to reduce the rear ride height.



These are the relative positions of the upper and lower ball joints when positive or negative caster is adjusted in the front suspension. The caster inclination lines connect the centerlines of the ball joints and are measured relative to a vertical line drawn through the center of the spindle. Positive caster will increase high-speed steering stability. Negative caster has the opposite effect and it also reduces steering effort (an attribute that accounts for negative caster angles on many production cars).

This top view illustrates toe-out, a required adjustment for optimum handling in autocross and competition. However, street cars should be adjusted for toe-in to maintain straight-line stability. See the text for recommended values.



Adjusting the front-end height of a Mopar is even easier. Just rotate the torsion-bar height-adjustment bolt in the lower control arm; clockwise will raise the car, counter-clockwise will lower the front-end height until the bottoming rubber is contacted.

TOE-IN/TOE-OUT

Once the proper front- and rear-end heights have been established, front-end alignment is the final step. This involves setting three important variables: toe-in/toe-out, camber, and caster. Now another road racer's rule-of-thumb: A well-handling Mopar will have toe-out on turns, positive caster (for both power or manual steering), and negative camber.

Toe-out and Toe-in describes the "aim" of the front tires. Toe-in occurs when the front tires point inward—towards each other, albiet very slightly; toe-out occurs when the front tires point outwards. If the toe-in/out settings are incorrect or excessive, tire wear will be excessive, regardless of how well the suspension system is designed or how many miles the tires are guaranteed to last. Most street-driven vehicles are adjusted for slight toe-in when stationary. This practice ensures straight-line stability and is fine for most street and drag-racing cars. But *opti-*

mum cornering requires a static toe-out setting. Although this may cause slight "drift" when driving in a straight line (requiring correction at the steering wheel), the improvement in cornering ability is quite noticeable. The new toe-out setting should not exceed the original toe-in specification; i.e., if the factory manual calls for a setting of 3/32-inch total toe-in, then the target toe-out value should not exceed 3/32-inch. Since the toe-in/out setting can vary during the adjustment of camber and caster, this adjustment should be performed last—or at least rechecked after the alignment is complete. For high-performance street cars, use the factory toe-in setting; and for competition road racing or autocrossing, use the toe-out setting.

CASTER

Caster is the measurement of the steering angle from vertical, as viewed from the side of the car. As the front wheels turn left or right, the spindles—to which the wheels are attached—pivot within the upper and lower ball joints. If the upper ball joint is positioned rearward of the lower ball joint, the top of the spindle inclines towards the rear of the car; the caster is then said to have a positive angle. When the upper ball joint is adjusted to a position that places it forward of the lower ball joint, the caster is said to be negative.

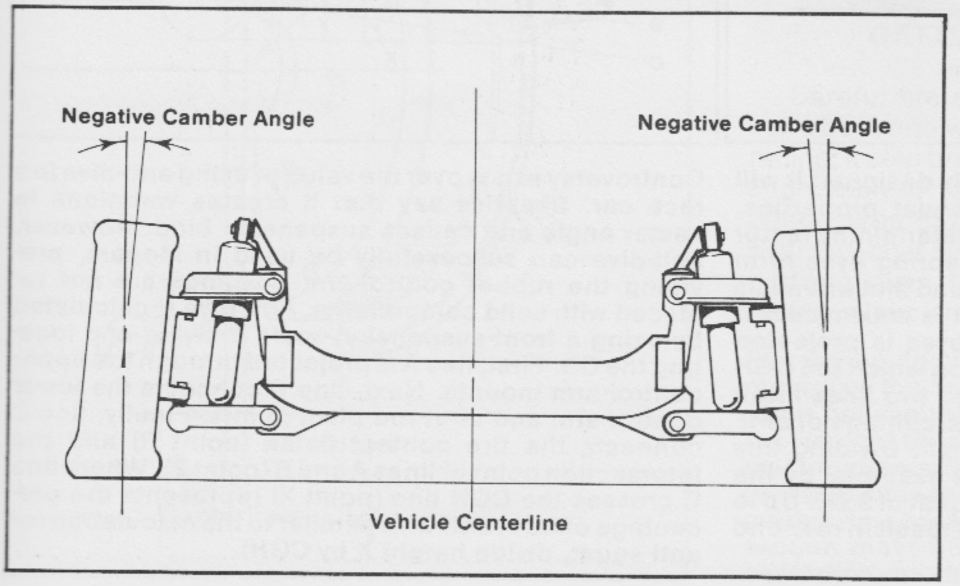
Many factory manuals indicate different caster settings for the same model car. These irregularities are due to different driver requirements with power or manual steering. Typically, the caster specifications will be positive for power-assisted steering and negative for manual steering. Why? With negative caster, the steering effort is substantially reduced and the driver has to put less "muscle" into driving—a nice advantage for "Joe Average" with his box-stock, manual-steering grocery getter. But there is a trade-off: handling and high-speed stability suffer with negative caster. *For optimum handling the caster angle should always be positive, regardless of the type of steering box.* Most road racers have found the best caster angle for cars weighing over 2600 pounds is 3° to 5° *POSITIVE.* Do not exceed 5° positive caster because steering effort will become uncontrollably high.

CAMBER

Camber is a measurement (similar to caster) of the front tire angle but from a different viewing position—from the front of the car rather than the side. From this head-on perspective, the tires appear to either "lean in" at the top towards each other (closer at the top is negative camber) or they lean out (further apart at the top is positive camber). *For good handling, the correct camber setting should be "NEGATIVE,"* with a value between 1/4° and 1° for street-driven cars. For road racing, up to 1-1/4° negative camber may be used. Negative camber tends to hold the outside tire vertical in a turn, improving traction and reducing tire wear. But remember, *do not* exceed 1-1/4° of negative camber because uneven tire wear can result.

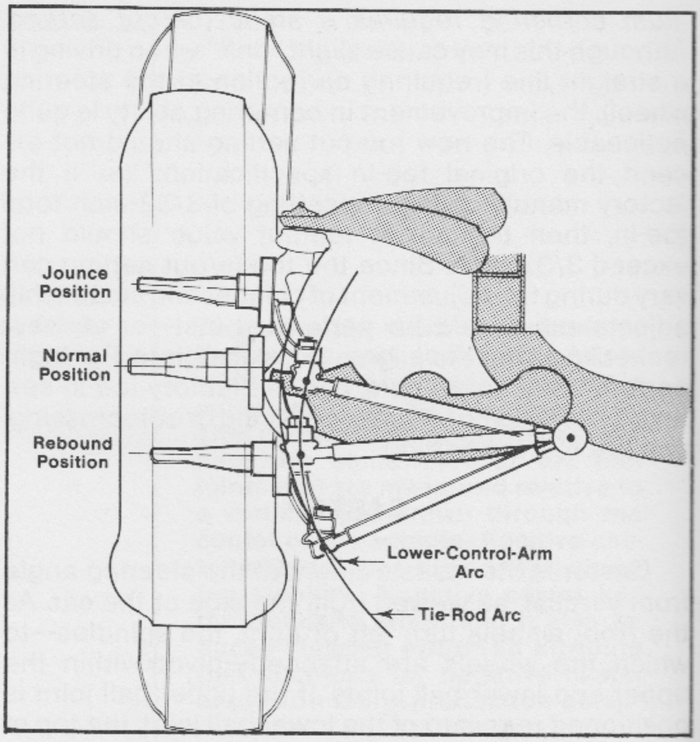
BUMP STEER

You may have noticed the tendency for the front end of some cars to "squirm" in a tight turn, particularly when the road surface is undulated and bumpy. The overall feeling is some loss of control, and the steering wheel will wiggle from right to left. These conditions are due to the ill effects of *bump steer,* a malady of the front suspension that generally occurs when the tie rods are not properly aligned with the lower control arms. Some suspension systems may have tie rods that are too long or too short, or they may be mounted too high or too low (relative to the lower control arm), or any combination of these factors. When this occurs, the tie rods do not follow the same arc taken by the lower control arms; they do not remain geometrically parallel. Therefore, as the suspension compresses and relaxes, the wheels are pushed in and out (in effect, steering the car) from this misalignment.



Negative camber tends to hold the outside tire vertical in a turn, improving traction and reducing tire wear on both street and competition cars. For street applications, a slight degree of negative camber (about 1/4 to 1 degree) is adequate, however, competition cars should use 1-1/4 degrees (maximum) of negative camber.

Fortunately, Mopar suspension systems suffer little from bump steer. However, if you suspect this may be a problem with your car, or if you plan on competing in autocross or road racing, it may be worthwhile to have a *professional race-preparation shop* check the front suspension. Most local "front-end" shops do not understand the concept of bump steer and won't know how to look for it. But if you won't be happy until *you* measure the bump steer on your Mopar and you have access to front-end alignment plates (the discs on an alignment rack onto which the front wheels are driven—they are calibrated to measure left and right turns), the test is relatively straightforward. With a floor jack supporting the front of the car, remove all tension on the torsion bars. Then lower the front end until the full-jounce position is reached. Now set the pointers on the alignment plates to a "zero" reference, and raise the front end—one inch at a time—observing any movement in the pointers until the front suspension is fully extended. The pointers should remain steady on their zero marks. Movement from zero is the result of bump steer, and *any amount of bump steer is too much.* For additional information on bump steer, refer to Chrysler Performance Bulletin 15—*Front Suspension Racing.*

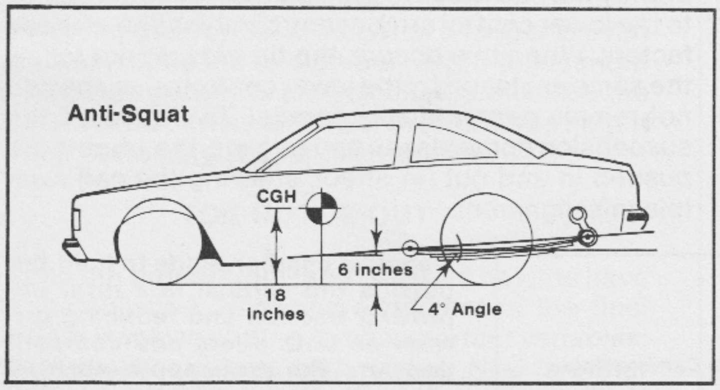


As the tire and lower control-arm move through jounce and rebound, the steering linkage must control directional stability. However, a poorly designed tie-rod system (arcs made by the tie-rod and the lower control-arm are not the same) will induce "bump steer"—forcing the front wheels to wiggle or "steer" when striking a bump.

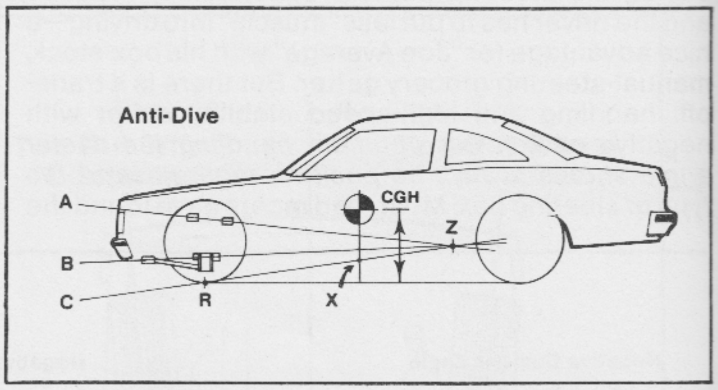
ANTI-DIVE AND ANTI-SQUAT

Anti-dive and anti-squat accurately describe a design feature of the front and rear suspension, respectively. *Anti-dive* is the tendency for the front suspension to resist compression when the brakes are firmly applied. *Anti-squat* is a rear-suspension feature that resists similar compression during hard acceleration. They both help maintain vehicle stability and limit excessive weight transfer.

Anti-dive is designed into Mopar front suspensions by the raised mounting position on the front



When the rear suspension is properly designed it will resist "squatting," i.e., have anti-squat properties, during hard acceleration. The main determining factor for anti-squat is the angle the leaf spring eyes form with a horizonal plane. It has been found that when this angle is set at 4 degrees, anti-squat is maximized. If the line that connects the spring eyes is projected forward it will cross the vertical line on which the CGH is positioned. The point where these two lines intersect can be used to determine the percentage of anti-squat the rear suspension will exhibit. Dividing this intersection height (6 inches in this example) by the CGH (18 inches) produces an anti-squat of 33%. Up to 50% can be used; more than this will result in rear-end instability.



Controversy exists over the value of using anti-dive in a race car. Skeptics say that it creates variations in caster angle and causes suspension bind. However, anti-dive can successfully be used in Mopars, providing the rubber control-arm bushings are not replaced with solid components. Anti-dive is calculated by using a front-suspension scale drawing and locating the CG. First, line A is projected through the upper control-arm mounts. Next, line B connects the lower control-arm and strut-rod pivot points. Finally, line C connects the tire contact patch (point R) and the intersection point of lines A and B (point Z). Where line C crosses the CGH line (point X) represents the percentage of front anti-dive (similar to the calculation for anti-squat, divide height X by CGH).

pivots of the upper-control arms. This subtle feature forces the CGH to work through a lever arm during braking that, in effect, causes a resistance to front-end movement. The result is reduced dive. Of course, shock-absorber design (and other factors) contribute to anti-dive; but most racers find the stock anti-dive features of Mopar suspensions to be adequate for all types of handling requirements.

Anti-squat is a very favorable design element of Mopar leaf springs. Simply stated, the angle generated by a line through the spring mounting bolts, relative to horizontal, will predict rear anti-squat (see illustration). Optimum anti-squat is obtained when the angle generated is 4.0 degrees. Do not exceed 5.0° or drop below 3.0° because rear-suspension dynamics will noticeably suffer. Keep this in mind when you are considering lowering the rear-end height. Using lowering blocks, instead of de-arching the springs, is not only safer but the vehicle will retain more anti-squat tendency. The same applies to extended rear shackles; they, in effect, de-arch the springs, reducing the mounting-bolt angle, and consequently, anti-squat.

TIRES AND WHEELS

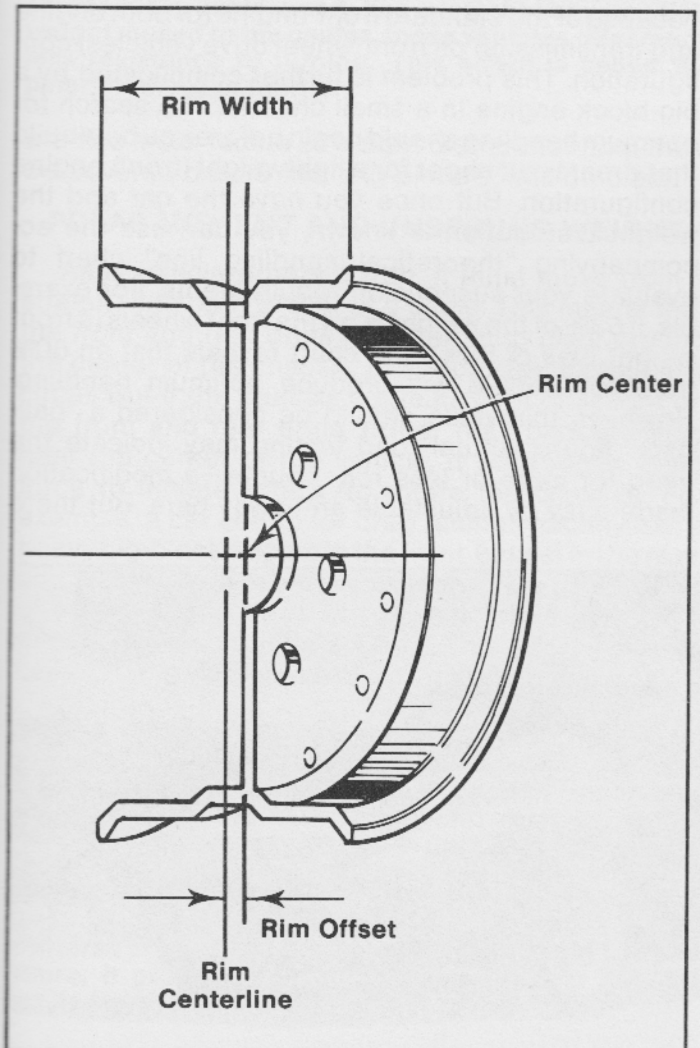
The best handling is achieved with wide, high-performance, low-profile tires. But wide tires will often interfere with the body, because stock Mopar wheel-well clearance is slim at best. Since raising the car is out of the question, tire selection is primarily dependent on wheel-well size; and wheel-well size is entirely dependent on how much modification is done to the body/chassis. The best—and the most costly—way to solve this problem is to replace the entire wheel-well with specially fabricated sheet metal and to incorporate external fender flares. These add-ons, when hand made and skillfully attached, can cost more than $2000. However, many racers are able to run reasonably wide tires by only installing fender flares and performing a minimum modification to the inside of the stock wheel-wells. The cost for this "next-best" solution is often under $1000.

Regardless of how much wheel-well clearance your chassis has, you should choose rims with the appropriate offset to place the tire in the middle of the available wheel-well space. Optimum wheel selection will usually mean the difference between "goin' racing" and "burning the midnight oil" pounding a desperately-needed 1/4-inch space into wheel-well sheet metal.

Most Mopar wheel installations require 3/4-inch or less offset (either positive or negative) for proper tire fit. This is all for the good because wheels of greater offset will place excessive loads on wheel bearings. If you can pick and choose, try to select wheels that will provide a positive offset of not more than 3/4-inch, since this increases the track of the car and (slightly) improves handling. Wheels can be of the traditional steel-rim variety, which are inexpensive, easily obtained, and often are no heaver than custom wheels. Or you can select a "mag-type" wheel, but be sure that it is SEMA-approved (indicated by the "SEMA" identifier cast into the wheel). You can also use the new modular wheels (if you can afford them). These wheels are very popular with road racers since they offer some offset adjustment. They are even showing up on many street-driven cars, despite the over $500 price tag.

GETTING THE RIGHT RUBBER

Careful tire selection not only makes the difference between winning or being an "also ran" in road racing; it also brings a noticeable "seat-of-the-pants" improvement to street driving. High-quality bias-belted tires offer superior handling characteristics over low-cost bias-ply tires, however, radial tires outperform even the belted designs. Although road racing slicks are usually of a bias-ply construction



Rim terminology often causes confusion, particularly regarding the amount and direction of offset. Some users believe that positive offsetting means *moving the rim (not the center section) outwards*, while most manufacturers define positive offsetting as *moving the center section (not the rim itself) outwards*. Most after-market rims have 3/4- to 1-1/2-inch offset—i.e., center section moved 3/4 to 1-1/2-inch outboard from rim centerline, as shown at left.

(because of the prohibitive cost in manufacturing belted or radial slicks), radial tires are *the* choice for street use. They offer not only improved handling but also longer life and better gas mileage. In addition, some radial tires are "speed rated" in "S," "H," and "V" grades. This designation indicates the ability of the tire to withstand sustained high speed. An "S" tire is rated for steady speeds up to 100 mph, an "H" rating means the tire will withstand 120 mph, and a "V" rating is given to tires that will remain "safe" at speeds in excess of 120 mph. The rating is usually incorporated as part of the tire designation: a *165/70**S**R13* is an "S" rated tire; *205/60**HR**14* and *225/50**VR**15* tires are respectively "H" and "V" rated. But not all radials are speed rated, particularly those that are American-made. This doesn't mean that imported radials are *better*; they're just "easier to figure out" and probably safer since there is less chance for misapplication.

TREAD VS. HANDLING

Ironically, the more tread on your new high-buck radials, the worse they will handle. Sound strange? It's all due to (are you ready for another technical term?) "tread squirm." Tire tread tends to move and vibrate at the ground contact point, disturbing traction and inducing a perceptible "sponginess" in the steering wheel. But as the tread wears down, the tire becomes more stable and handling and steering-wheel "feel" become just a bit better. Since this fact is known to most road racers in showroom stock and the radial challenge series, they will have their new radials shaved down on a special machine (anything to win races). But we are not suggesting you need to do the same for high-performance street driving. In exchange for tread life of 50,000 to 60,000 miles, most enthusiasts are willing to put with some initial tread squirm.

High-traction tires are usually manufactured of a soft rubber compound and will generate more tread squirm when new, while tires with a harder compound will have a more stable tread. Additionally, tires designed for "wet" driving (a tread incorporating wide and deep groves) will exhibit more squirm. Make your choices carefully!

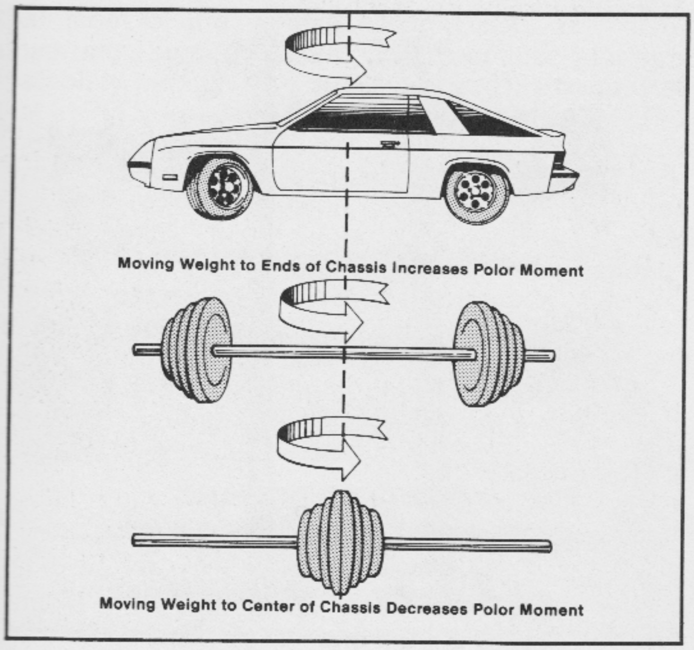
WEIGHT DISTRIBUTION

In the last chapter, we looked at methods of measuring chassis weight on each wheel and the proportion of the total vehicle weight on the front or rear end. This proportion, or distribution, of weight plays an important role in handling. How to optimize this element of chassis preparation for superior handling is the subject of this section.

It is not surprising to learn that most chassis configurations will produce optimum handling when the weight on the front wheels equals the weight on the rear wheels—a 50/50 distribution. After all, with uniform weight distribution, there will be less tendency for understeer or oversteer since the lateral forces will be nearly the same on both the front and rear ends. Building a competition chassis with 50/50 distribution is relatively easy because once the vehicle is completely stripped to the bare essentials, parts can be installed where the builder wants them—ballast may even be added to the passenger side to compensate for the driver's weight. But modifying a street machine for the same uniform weight distribution takes some real planning and is not always practical.

Many American-made cars are front-end heavy because of the standard front-engine (or both engine and transmission on front-wheel drive vehicles) configuration. This problem is further complicated by a big-block engine in a small chassis. The search for optimum handling should begin before you buy/build that dream car; shoot for a lightweight (front) engine configuration. But once you *have* the car and the weight distribution is known, you can use the accompanying "theoretical handling line" chart to evaluate your suspension requirements. For example, if 53% of the weight is on the front wheels (a front weight bias of 53%), the chart reveals that an 80% front roll couple will produce optimum handling. However, this value should be considered a "ballpark" figure. Actual road testing may indicate the need for more or less roll couple—a modification made easy by adjustable anti-sway bars. But there

This Shelby sports car is an exciting prototype for a mid-engine Chrysler car of the future. From performance cars (such as the Ferrari 308 and DeTomaso Pantera) to sports and commuter cars (such as the Fiat X-1/9 and Pontiac Fiero), the basic design is similar; most of the vehicle mass is concentrated towards the center of the car to generate a low polar moment of inertia and quick, responsive handling.

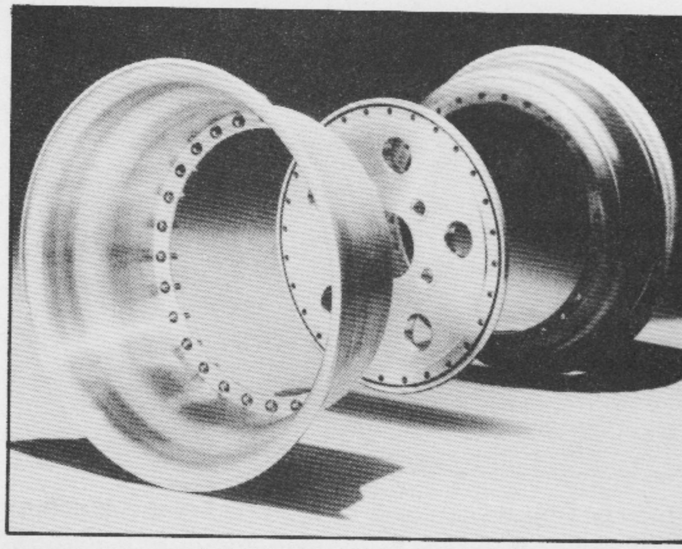


Many American cars have a heavy engine up front and a large rear end and gas tank at the rear, creating a weight distribution similar to a weight-lifters bar—very end-heavy. This distribution resists steering inputs and directional changes—just like a weight bar resists rotation. However, the nimble Porsche 914 and Fiat X-119 have a large proportion of thier mass located in the center of the vehicle and, like a weight bar with all its weight moved to the center, these vehicles offer much less resistance to turning. This effect is called the polar moment of inertia.

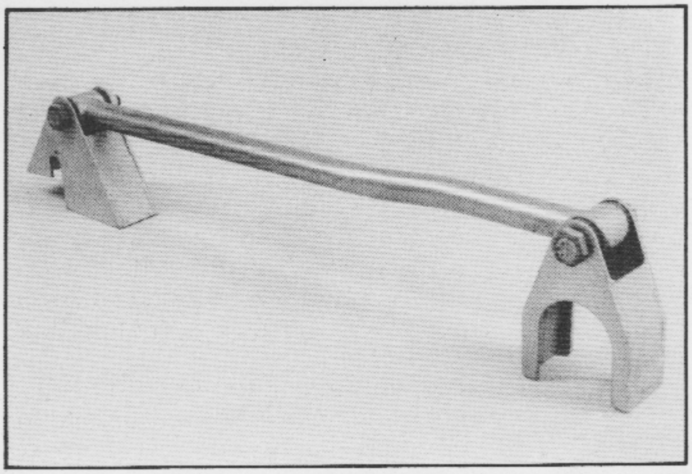
are two more facets of weight distribution that must be covered before the "big picture" will be clear.

POLAR MOMENT AND UNSPRUNG WEIGHT

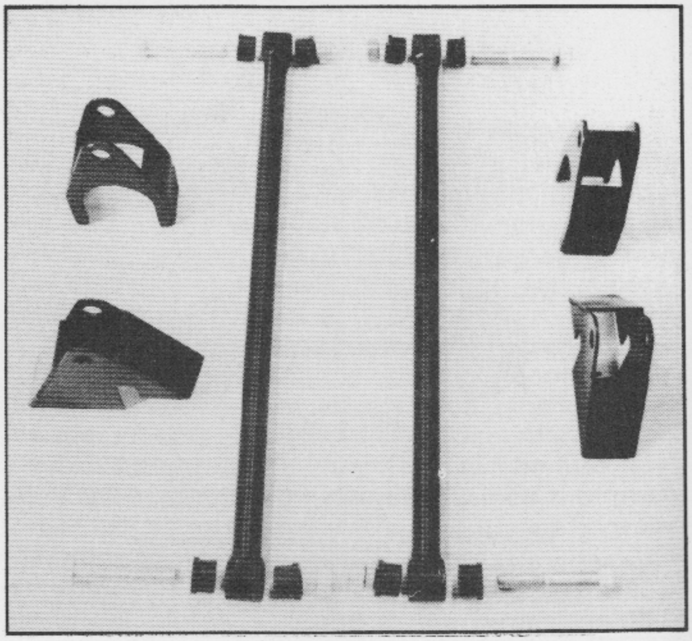
As a vehicle enters a turn, it must move from a straight-line path to that of a circle. This means that the front and rear ends must begin to pivot around the center of the chassis, in effect, start to spin in a circle. But if the chassis contains a lot of weight on the front and rear ends, rather than concentrated near the center of the vehicle, it will require more force to initiate a turn (to rotate the chassis). For example, a car set up for drag racing typically has a heavy big-block engine in the front, ballast in the rear, and a gutted driver's compartment. This distribution of weight to the ends of the vehicle increases the effort required to initiate a turn. This effort (force) is called the "polar moment." A high polar moment will



The modular rim is the latest in wheel technology. While more expensive than conventional wheels, they are far easier to adjust for rim width and offset; and they facilitate tire installation and removal. Top—Center Line modular wheels are precision stamped on a 500-ton press. Middle—Note how the tread has been shaved on this tire from Joe Varde's Charger (see page 105) to optimize handling. Bottom—All out racing requires a "slick"; when warmed to operating temperature, it provides outstanding traction and stability while being able to withstand 200 mph.



This traction device was developed for the Mustang road-racing program to prevent wheel hop that occurred from mounting the rear axle in the center of the leaf springs (rather than offset forward, as is the case on Mopars). These units, called over-riders or under-riders, eliminated wheel hop and—where necessary—are the only type of traction devices that should be used on a car set up for handling.



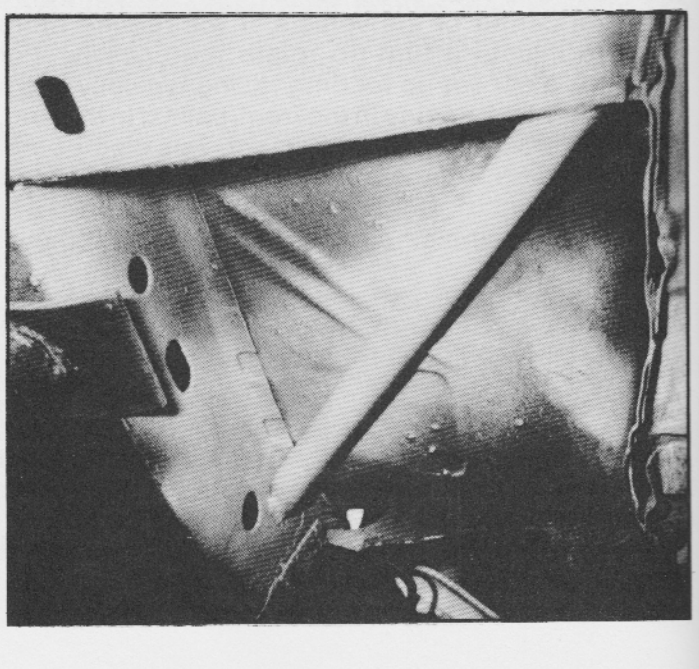
cause momentary hesitation after the abrupt steering is initiated. The solution, of course, is to move as much weight as possible to the center of the chassis. On street machines, this is not easy; but there are at least a couple of possibilities: relocate the gas tank closer to the rear axle or move the battery to the deck just behind (or under) the rear seat.

Now you know why mid-engine vehicles are renowned for their excellent steering response. With most of the weight concentrated around the CG, the vehicle has a very low polar moment (usually accompanied by a very high price tag).

Up to this point, we have considered two aspects of weight distribution: 1) whether there is more weight on the front or the rear axle, and 2) how weight distribution affects the polar moment. Another important aspect of weight distribution is the amount of total vehicle weight that is sprung and unsprung. *Unsprung weight* is that portion of the vehicle that is not supported by the springs: tires, wheels, differential, brake assemblies, front suspension arms, spindles, etc. Optimum handling characteristics can only be achieved when the unsprung weight is keep to a minimum. Excessive unsprung weight will force the springs to over-react to bumps and irregularities in the road surface, rebounding too far because of the excessive momentum stored in heavy suspension components. However, a lightweight unsprung mass will minimize spring rebound, reducing the loads on the springs and shocks and ensuring that the wheels stay on the ground as much as possible. The overall result is better handling and control with lower spring and shock rates.

You can minimize unsprung weight by using the lightest wheels and tires possible. (Trans Am cars will use magnesium wheels, small diameter tires with thin wall sections, trimmed suspension components, etc., all adding up to better than a 50 pound weight reduction per axle.) Big, thick, impressive-looking traction bars are absolutely out! If you have a wheel-hop problem with high-performance Chrysler springs, you can use a single (per side) connecting link mounted from the frame to the rear-axle assembly. This system is lightweight and, since one end is mounted to the frame, only a portion of its weight is unsprung.

The final consideration in unsprung weight is spring stiffness. It is possible to have springs so stiff



A simple diagonal brace, such as this one from the rocker panel to the front longitudinal section, can add structural stiffness to the chassis without adding much weight. The cost is low and installation is straightforward, although welding can be difficult because of limited space and an awkward welding position.

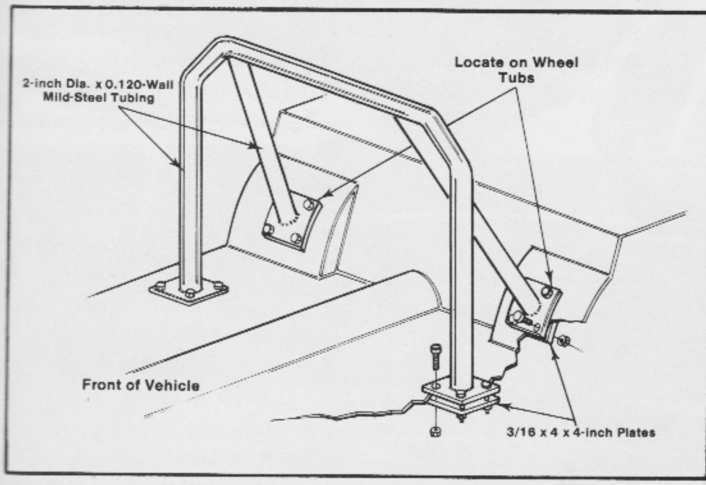
that most of the vehicle weight acts as if it is unsprung. Super-stiff springs will most likely induce instability, meaning the driver will have his hands full just trying to keep the thing moving in a straight line. There is an exception to this rule, however; it is found on cars racing on high-banked ovals at speeds of 200 mph or greater, or using the new ground-effect aerodynamics. Under these conditions—and often with the additional complication of aerodynamic "wings" that can add hundreds of pounds of down force to a vehicle at speed—the super-stiff suspension "no-no" rule doesn't apply. But for most road-race cars, and especially for street-driven machines, *the lowest spring rate that will ensure stability and control will provide the best handling.*

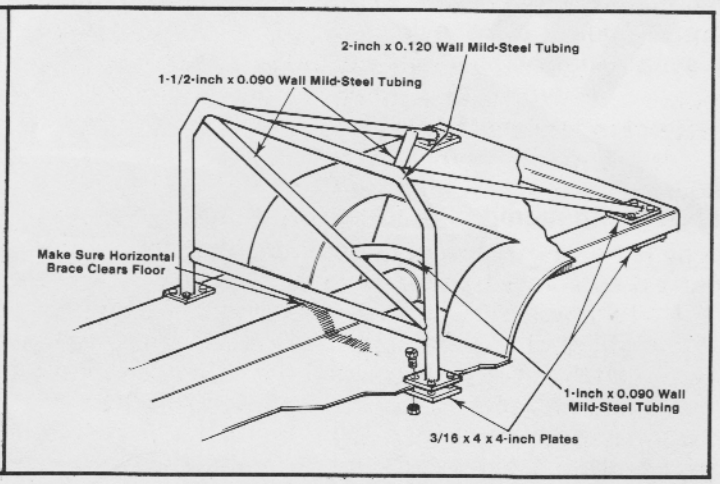
CHASSIS STIFFNESS

Proper springs, low unsprung weight, the correct roll rates, etc., all add up to produce a well-handling vehicle with solid stability and control, right? Yes, if the chassis doesn't flex and induce "strange" characteristics into the system. A "flexy" chassis will act like an oversize torsion bar, twisting and relaxing, altering suspension alignment, steering stability, and generally giving the car a "squirrelly" feeling. But there are several things that can be done to improve stiffness, and some are quite inexpensive.

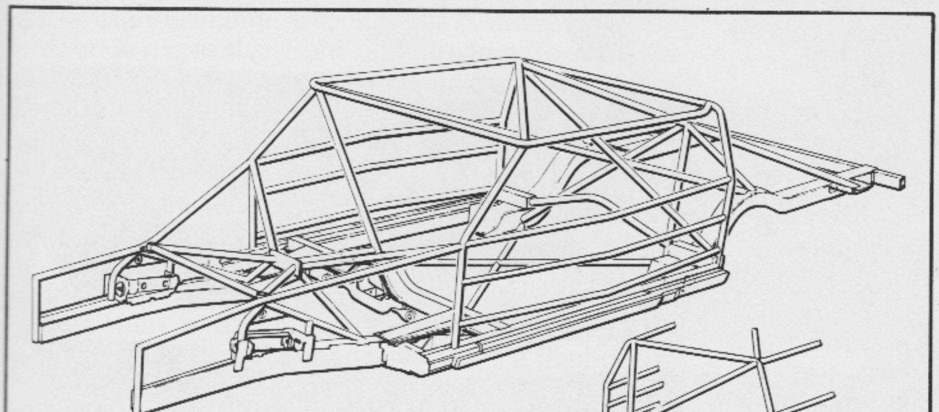
The best chassis stiffener is a roll cage, and it should be considered an essential item for competition use. You can fabricate your own cage from mild-steel tubing; the chrome-moly material should be left to a professional shop since it requires stress-relieving after welding. Use MIG (Metal Inert Gas) or TIG (Tungsten Inert Gas) welding techniques, and reserve regular arc welding for less critical jobs. Refer to the Chrysler roll-cage design pictured in this book; it is solid, stiff, and uses less tubing than conventional roll-cage designs.

A roll cage will certainly add stiffness to a chassis, but it is somewhat impractical for a street machine. Chassis flex can be reduced by less radical methods, like the addition of a simple but well-designed roll bar. Frame connectors and diagonal braces from the frame rails to body rails will also help. Frame connectors can be fabricated from thin-wall, mild-steel box tubing. But if building connectors from the "ground up" is too much for you to tackle, Moroso manufactures a frame-connector package for A-body cars. The instructions for this kit state that the braces need only be *bolted* in position. This may suffice for drag racing; but for road-racing applications, the connecting braces should be welded to the subframe.





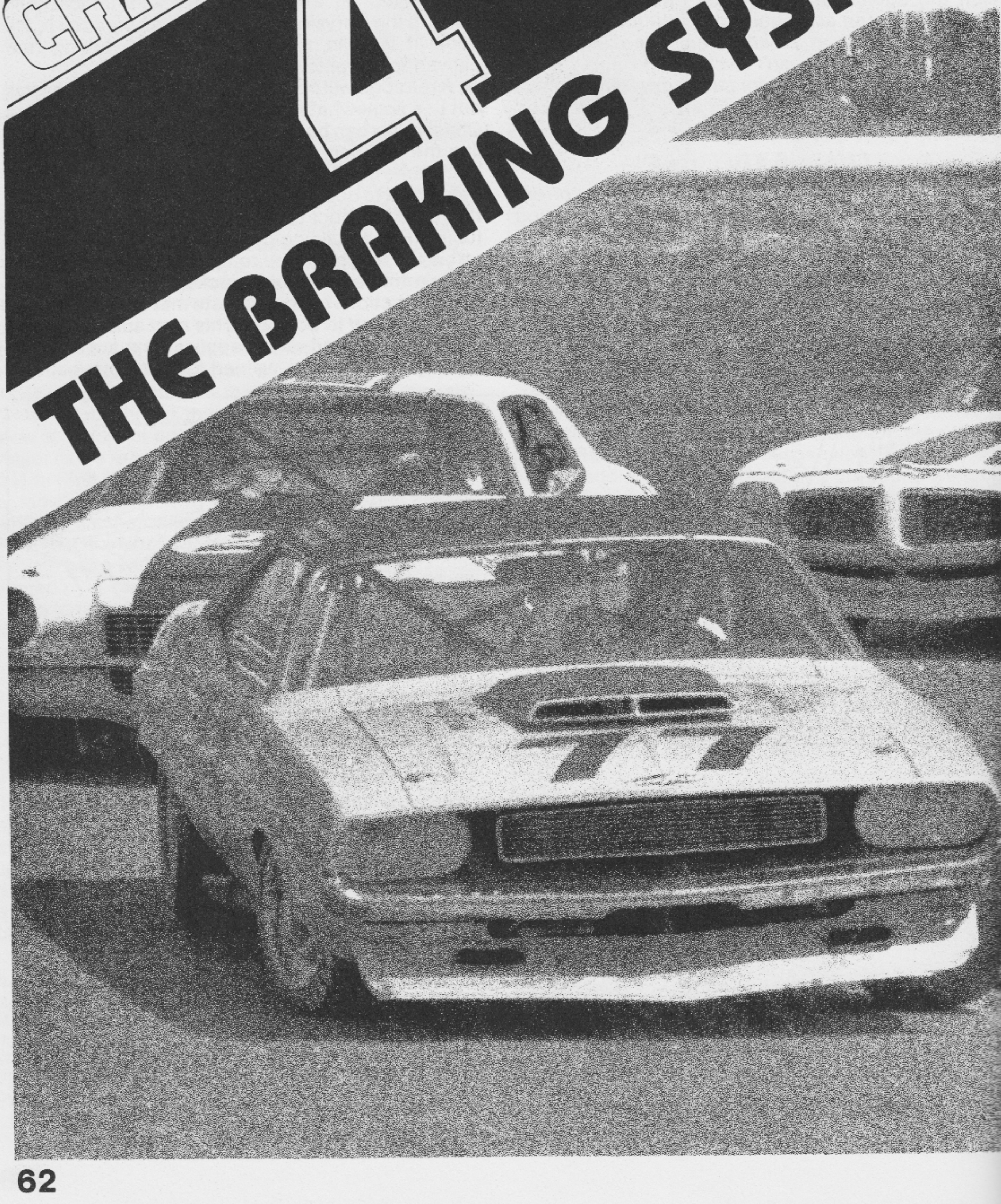
The basic roll-bar design for typical American sedans (left) may not be as strong as a diagonally-braced competition bar, but it does improve driver and passenger safety and torsional rigidity, sorely needed in coupes and convertibles. A well-braced competition roll bar (right) can nearly double torsional rigidity, substantially improving handling and steering accuracy.



The Chrysler-designed roll cage for circle-track racing works very well for road racing, too! Light, strong, and adaptable to all A-, B-, E-, and F-body cars, it includes the unique "X" bracing in the rear section. This design is much stronger, uses less tubing, and is lighter than standard longitudinal bracing.

CHAPTER 4

THE BRAKING SYSTEM



Notice how evenly balanced the Challenger is while braking into this sharp turn. Optimum braking is essential for good handling. When all four brakes are properly sized and functioning correctly, the car will settle gently on the suspension without upsetting stability.

Photo by Toni Cortes

BRAKES AND BRAKING

Incorporating properly working dynamics in the chassis is an important design element when building a superior road car. But good chassis design is only one facet; superior brakes are an absolute necessity for optimum handling.

Inadequate braking ability is always unsafe and often results in otherwise-avoidable accidents. In this section we will pinpoint the weak spots in typical factory "binders" and give specific recommendations that will help you build a safe, high-performance braking system.

FRONT/REAR PROPORTIONING

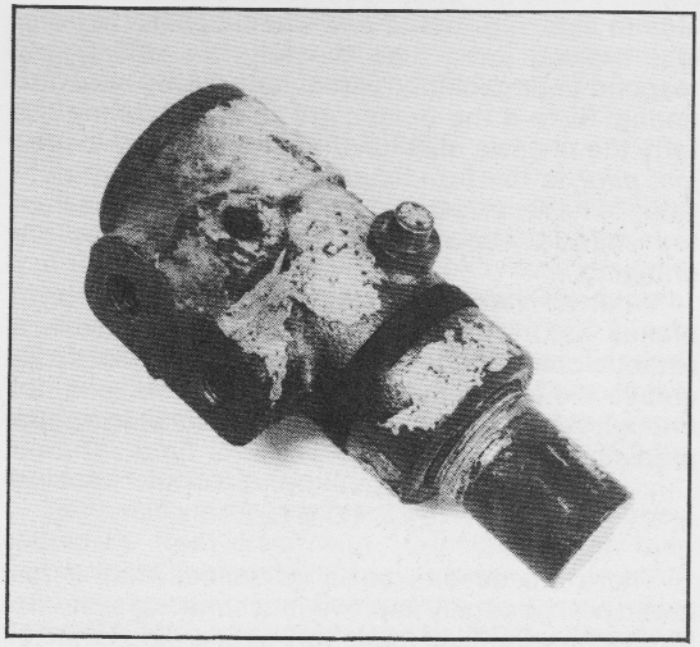
The greatest discrepancy in most stock braking systems is the improper proportioning of braking forces between the front and rear wheels. Too much rear braking force will cause rear-wheel lockup. (This is a common problem with pickup trucks particularly when the bed is empty because most truck brakes are designed to stop efficiently when carrying a heavy load.) Rear-wheel lockup often causes severe handling problems—i.e., a complete spin out—and poor overall braking. Also unsafe, although not as common is excessive front braking. In a front-wheel lockup, the driver loses at least some steering control and, as in rear-wheel lockup, overall braking performance is inferior.

A simple solution to these problems is to incorporate an *adjustable* valve in the braking system—a valve that will, in effect, "fine tune" the front and rear braking forces. In an all-drum system, adjustable proportioning is a far simpler balancing method than changing wheel-cylinder diameters. But regardless of whether an all-drum, a disc-drum, or an all-disc system is used, an adjustable proportioning valve will facilitate tuning the braking system and improve stopping, handling, and safety.

In nearly all situations, a properly adjusted proportioning valve allows the front brakes to lock up just before the rears. This means the front braking energy must be greater than the rear, since a decelerating vehicle will transfer a large proportion of its weight to the front end. A substantial degree of front-heavy braking is already designed into most vehicles (from the larger drums, linings, etc. used on front brakes); so proportioning valves normally make only slight adjustments in front/rear braking forces. However, these slight adjustments can substantially improve overall braking. But despite the benefits, most production cars are not equipped with an adjustable proportioning valve, even though it is a relatively inexpensive component (Detroit's "save a nickel here, a nickel there" philosophy).

SIDE-TO-SIDE BRAKING FORCES

Another important factor in optimum handling is



This is the standard Kelsey-Hayes disc-brake proportioning valve found on 1965 to 1970 A- and B-body cars. It is pre-set at the factory, and its "adjustability" is not readily apparent. However, twisting the clyindrical section on the lower right will vary the rear braking pressure.

Pushing the brake pedal without worrying about tire lockup or brake fade are the attributes of a well-designed braking system. Race-track speeds demand the ultimate in braking components; however, many braking improvements for street machines can be obtained through stock parts swapping.

Photos by Toni Cortes

If the driver of this car applied the brakes, he could make an already-unstable condition much worse, particularly if the braking system worked less than perfect. In this case, brake grab could be an "unsettling" and further destabilizing factor.

Many suppliers offer brake linings superior to original factory parts. Just switching to a harder, semi-metallic lining compound can make a substantial improvement in braking—both in stopping ability and fade resistance.

assuring that the braking forces are balanced from side to side. If a right-hand wheel applies more or less braking force than the left, the vehicle will respond unpredictably, particularly during hard cornering. Even if the brakes are released *during* the turn, the chassis may have been unequally loaded *just prior* to the turn by an overloaded left or right wheel. The result can be a "squirrely" response from the chassis, especially if any of the brakes are dragging.

When all is working well, the proper application of brakes before and during cornering will produce smooth, controllable operation. This will directly increase the handling ability of the vehicle, and not enough can be said for the pleasure of driving a solid, predictable chassis.

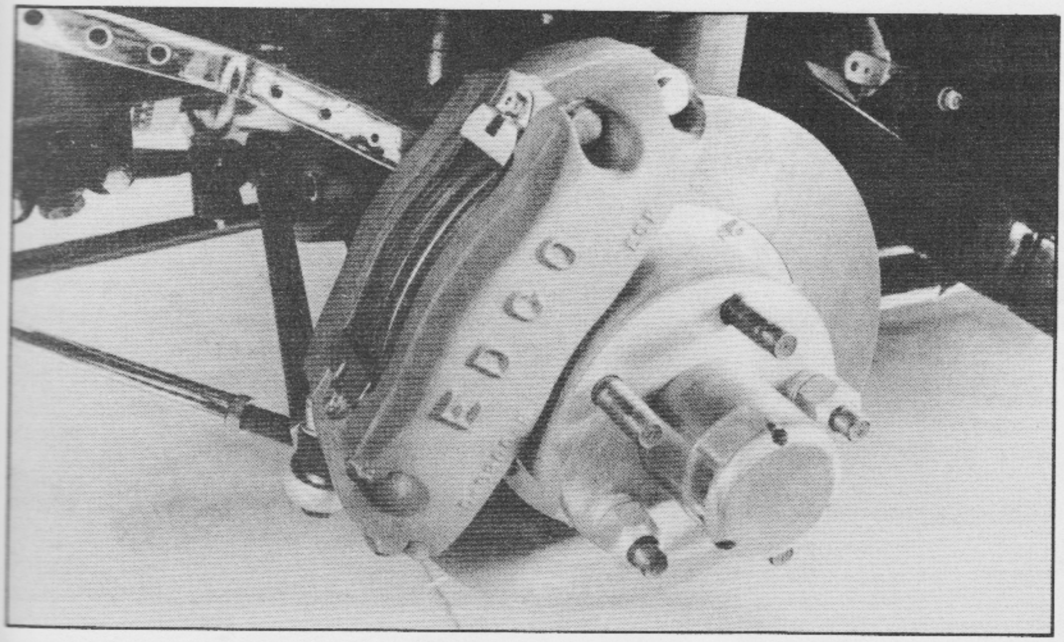
WHERE TO START

There are several basic guidelines that, if followed, will generally improve most braking systems. But because braking loads vary over a wide range of uses (street to competition), a more detailed analysis will be provided in upcoming sections.

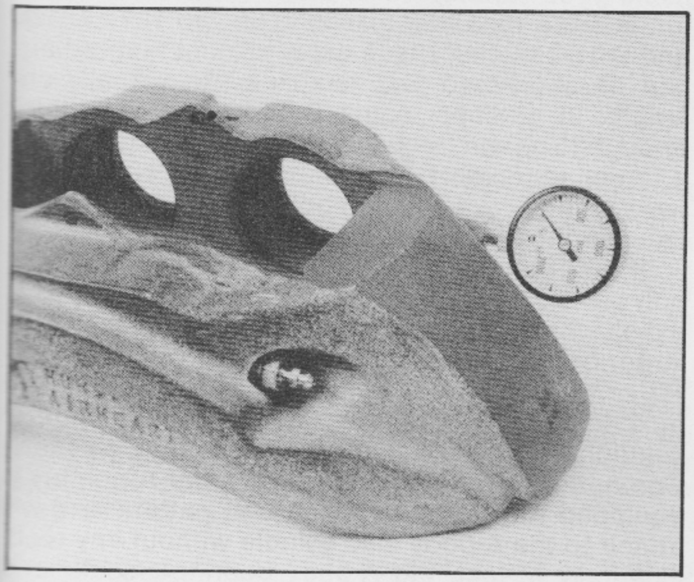
THE BIGGER THE BETTER?

The stock factory brakes are a sad story. Most A-, B- and E-body vehicles were designed to provide basic transportation, with minimum cost a major consideration. Unfortunately, the braking system, too, came under the "bean counter's" scrutiny. All six-cylinder cars, most 2-bbl V-8s equipped with automatics (and some manual shifts), and particularly A-body cars built prior to 1968 are all equipped with "minimum" brakes. The only exceptions to this were the Formula-S Barracudas and GTS Darts; both models used a disc/drum system. The *stock* 9- and 10-inch drum brakes are marginal for a six-cylinder car and completely unacceptable for a V-8 in any performance application.

However, if the *original* tire size and engine horsepower are maintained, the stock brake-system



This high-performance Edco caliper can provide comparable braking to similar AirHeart units. In addition to this 4-piston model, Edco also manufacturers a 6-piston caliper for even more demanding requirements. Notice the massive structure of the caliper designed to reduce flex. Caliper rigidity is a vital part of a high-performance brake system and is an essential requirement in road- or circle-track racing.



This one-piece AirHeart fixed caliper incorporates a line-pressure testing gauge, manufactured by Lamb Components, that aids brake-system testing and proportioning valve adjustment. The one-piece design of this caliper allows it to withstand tremendous pressure without leaking, deflecting, or failing.

components *can be used.* But top-quality brake pads (and/or shoes), discs, drums, and proper brake-force balancing (with a proportioning valve) are required modifications. However, if larger tires and additional horsepower have been added, more brake *swept area* is generally required. Unfortunately, adding bigger components to a Mopar is not always inexpensive. But, when true high-performance braking is required, replacing the stock components with heavy-duty pieces is mandatory.

B-BODY BRAKE BOOST

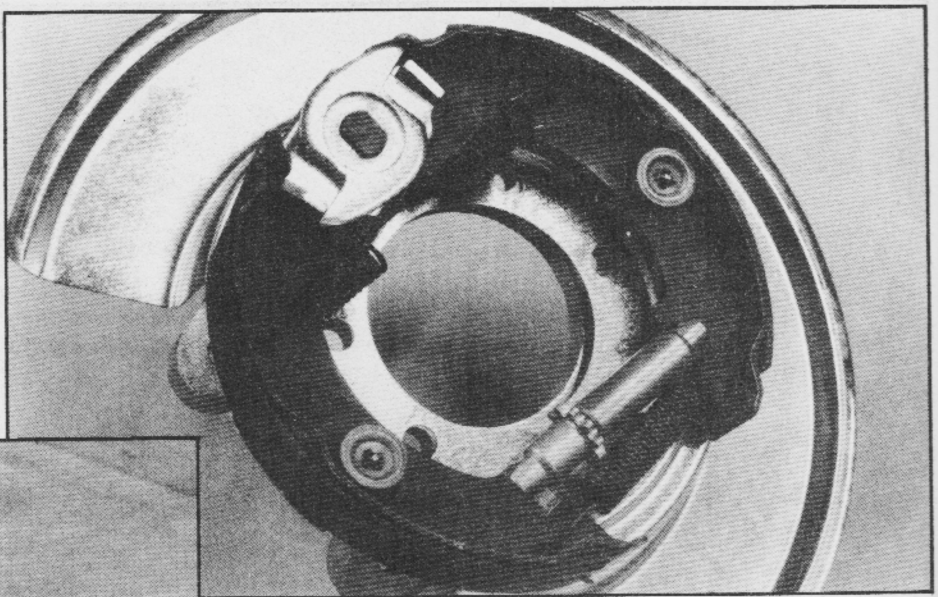
Modifying the braking system on B- and E-body cars is relatively easy because an excellent out-of-the-box brake system (the Kit Car package) will bolt on with only few modifications. All of the parts and pieces are available from any Direct Connection dealer. The Kit-Car system is an excellent choice for all but the most ambitious racing project, since it is reasonably priced (compared to custom components) and provides nearly the same braking capabilities as expensive, racing systems.

All-out competition with a B-body car, however, requires the absolute best in braking. This is available in the Hurst/Airheart competition brake package—an expensive, no-compromise, disc-brake system. But the Hurst/Airheart system has no provision for a parking brake and requires modifications to both the stock and Hurst pieces for Mopar installations. Only the experienced racer (or qualified shop) should attempt installing this system.

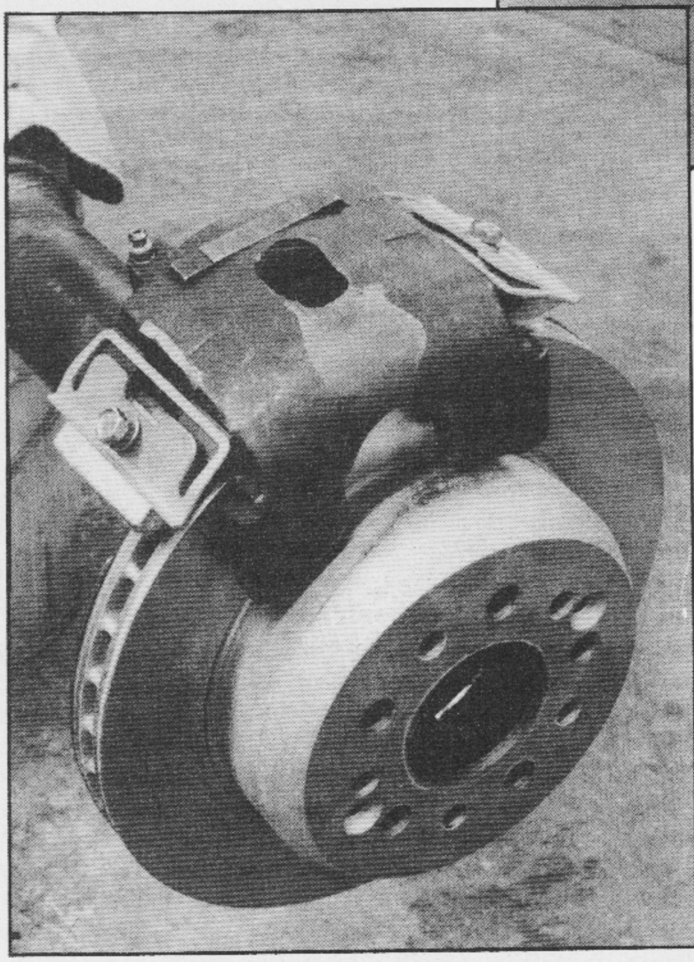
A-BODY BLUES

The A-body chassis poses a much more serious problem for the enthusiast. Early A-bodies used a small 4-inch bolt pattern, which greatly restricted the interchangeability of wheels with B- and E-body cars. In addition, the early A-body was supplied with 9-inch front brakes and a small (7-1/4-inch) Spicer rear end, both of which just barely handled the power of the slant-six engine. When connected to anything more powerful, both the brakes and rear axle MUST be replaced with better equipment!

The basic starting point for brakes on any A-body should be front discs with 10-inch-minimum rear drums. Installing front disc brakes on an early A-body requires replacing spindles and the master cylinder. Some B-body components can be fitted with only a few modifications (see next section). A 10-inch rear drum system was offered on the 7-1/4-inch Spicer, but these are now very rare. A much better way to increase rear-brake area is to install the ubiquitous 8-3/4-inch Mopar rear axle or the 8-1/4-inch Spicer axle (used on the 1974 and later A- and F-body cars).



The Chrysler sliding caliper is an excellent choice for a rear-disc application, but there is often no provision for a parking brake. This system is an exception; adapted to the 8-3/4-inch rear end, this Imperial rear-disc assembly uses a unique drum-brake assembly as a parking brake. The "hat" section of the rotor acts as the drum, while a special backing plate retains the small brake shoes. There are several rear-disc systems that can be adapted to other cars, but they all require special machine work and fabrication.



The larger brakes and axles supplied on this rear end are better able to withstand the rigors of high horsepower and cornering loads.

The only drawback with the 8-3/4-inch rear end is the 4-inch bolt pattern used on most A-body axles. There were a few, optional, 8-3/4-inch A-body axles that were equipped with the 4-1/2-inch bolt pattern, but they are no longer available from Chrysler. Axles from B-bodies that used the 8-3/4-inch rear end all have the 4-1/2-inch bolt pattern, but they must be shortened for use in the narrower A-body housing.

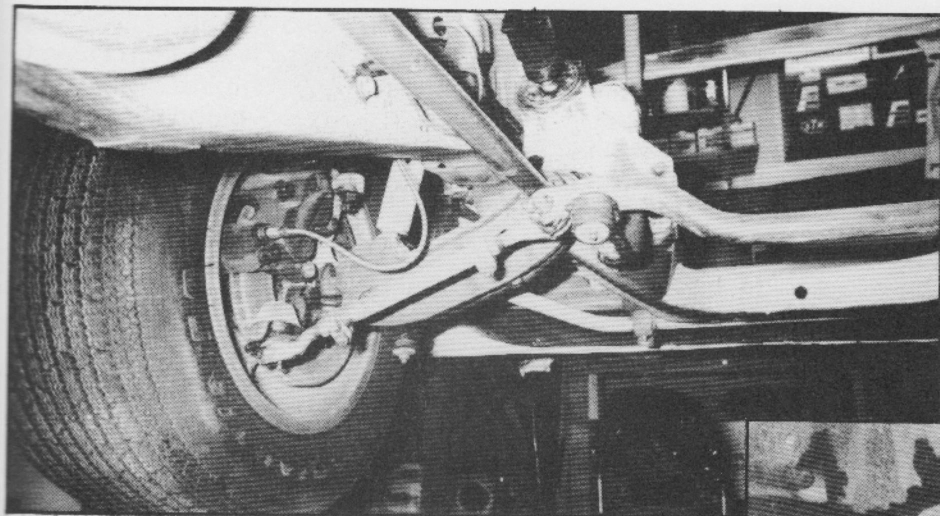
If you decide to shorten the large bolt-pattern axles, find a high-quality machine shop (one with experience in axle work), and be sure the axles are shortened and resplined, not cut and re-welded. *Axles modified by welding will fail and are not recommended for any application.* Axle shafts that make good candidates for shortening can be found in the 8-3/4-inch rear axles in 1968 to 1972 D-100 and D-150 trucks. These axles are not tapered and can be shortened and re-splined to virtually any length. In addition, some B-body axles are thick enough to permit re-splining to certain lengths. With a larger rear-axle bolt circle, you can choose between the 10- or 11-inch B-body drum brakes, or you can use the 8-3/4-inch Kit Car disc brakes.

If the rear axles are modified for the 4-1/2-inch bolt pattern, it is good practice (and it greatly simplifies wheel interchangeability) to install the same larger bolt circle on the front hubs. If you install the 10- or 11-inch B-body front brakes (discussed in the next section), your problem will be solved. Another solution is to use the 1965-1968 B-body (Belvedere, Coronet) disc rotor. This hub-and-rotor assembly is equipped with the 4-1/2-inch bolt pattern and will attach to the A-body disc spindle without any modifications. In fact, the A-body bearings, seals, and calipers can be used with this assembly.

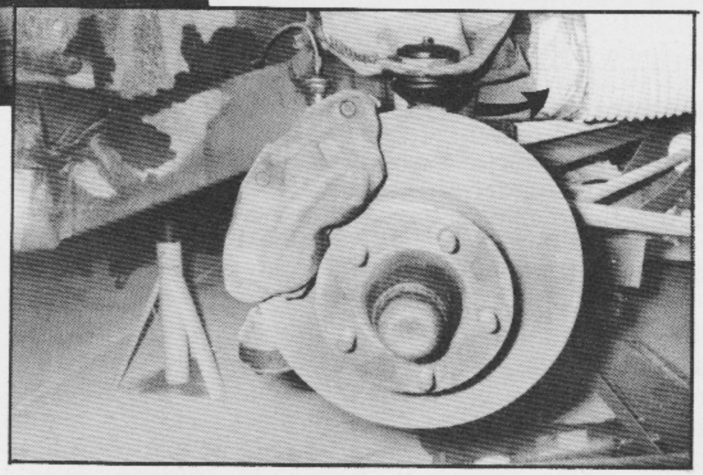
FRONT B-BRAKES FOR THE A-BODY

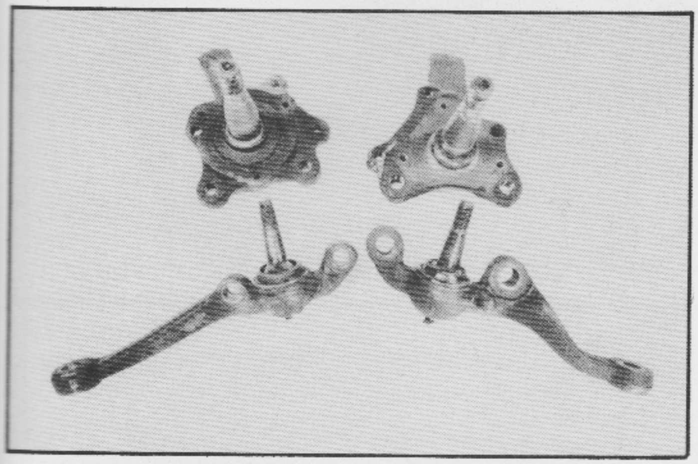
The A-body will accept various components from the B-body that will improve braking. First of all, the totally inadequate, 9-inch A-body brakes and spindles MUST be replaced with either 10-inch A-body pieces or, if the upper control arm is modified (to accept the larger ball joint used on the B-Bodies), the B-body 10-inch, 11-inch, or disc spindles can be installed. In either case, the B-body backing plates and shoes will easily bolt on.

If the B-body spindles (disc or drum) are installed, the lower control arm does *not* have to be modified to accept the larger B-body lower ball joint—the A-body tie-rod ends will also work without modification. The upper control arm can be modified to use the large disc brakes from the Road Runner and Charger R/T. However, if you prefer to use the 11-inch B-body drum brakes, the backing plates for these brakes will not bolt onto the 10-inch spindles. In this case, the



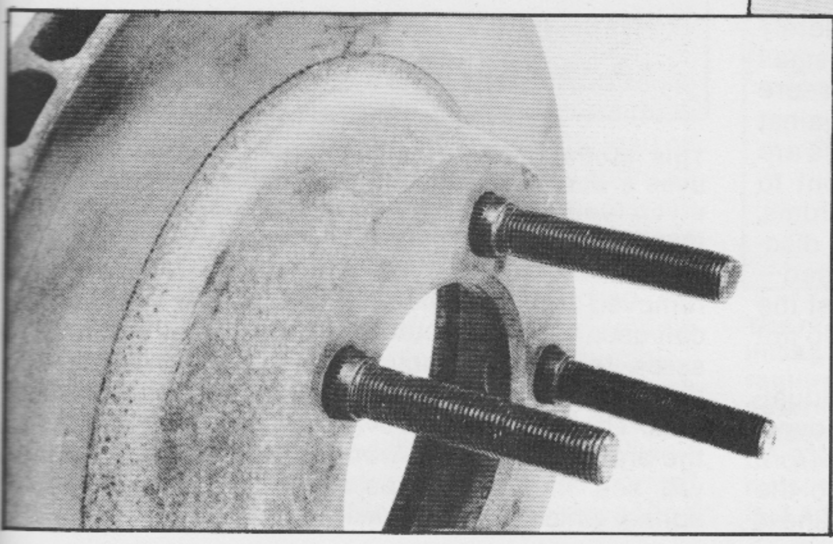
This 1968 Bendix B-body disc-brake assembly has been installed on an early A-body. The massive caliper and 12-inch rotor (with a 4-1/2-inch bolt circle) make an enormous difference in braking, handling, and safety. Cool air, ducted from under the front bumper (arrow, below), is directed at the caliper to aid cooling while road racing.

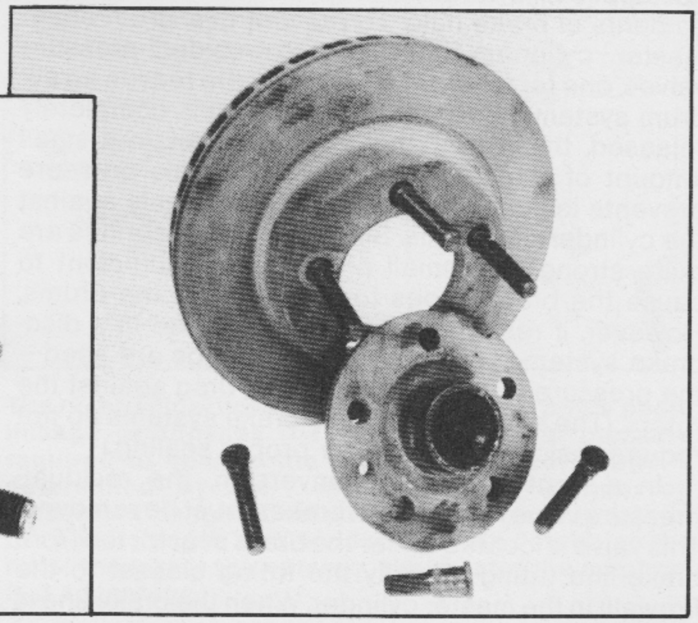




The A-body spindle and lower ball joint (left) are smaller than the B-body pieces. Because of these differences, both spindle and ball joint must be changed as an assembly. And since the caliper mounts are also different, the B-body calipers must be used with the larger components.

The standard rotor and hub assembly used on 1965 to 1969 A- and B-body cars is the best choice for most disc-brake applications. The long lugs may interfere with some lug-nut lock sets, but they add an extra measure of insurance for high-performance and competition use.

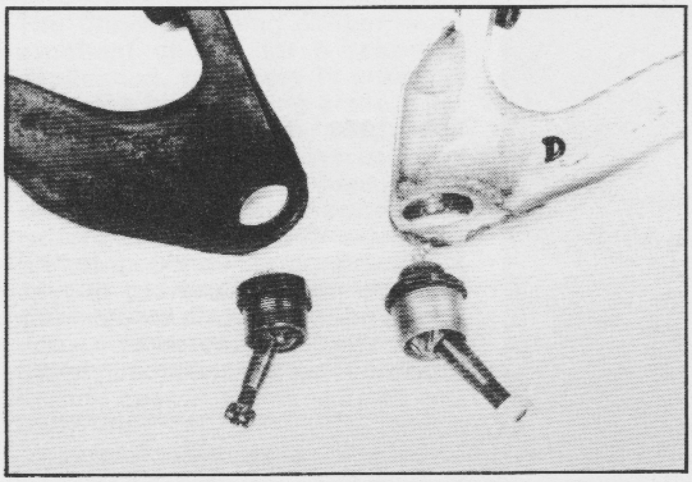




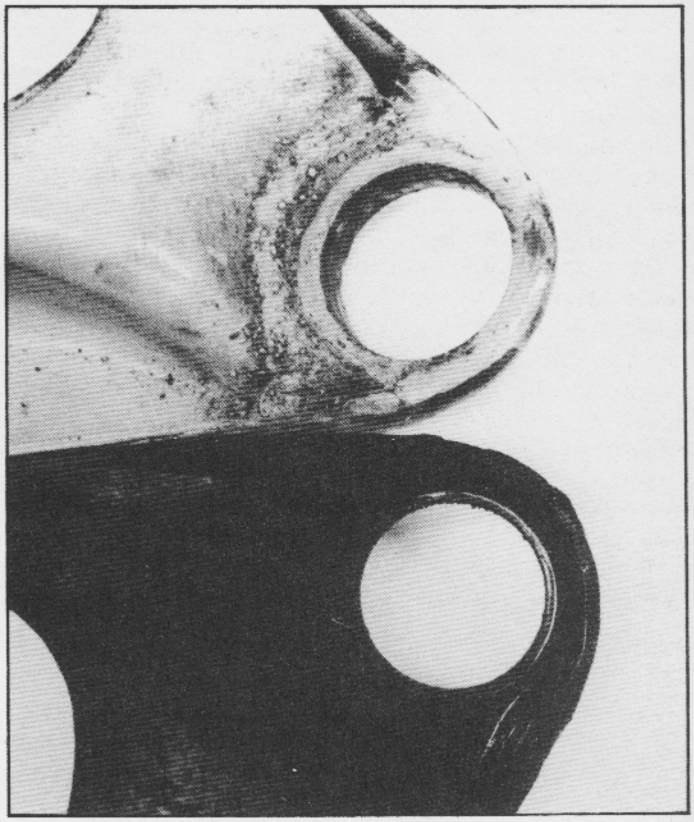
11-inch B-body spindles must be installed along with a modified upper control arm.

Brakes even larger than the 11-inch passenger-car brakes (such as those used on the C-body Fury or Monaco police cars) can be installed if the socket in the *lower control arm* is machined to accept the larger C-body lower ball joint.

Any modifications to upper or lower control arms, ball joints, or sockets require precision machining. The upper control-arm bearing retainer must be cut out to install the larger ball joint and a new retainer carefully welded in and machined to size. When the



The stock A-body upper control arm (dark-colored unit) must be modified to accept the larger B-body spindle and ball joint when installing the Charger R/T and Road Runner front brakes. Note how the modified control-arm ball-joint socket has been built up and relocated inward. The slight additional inward mounting permits more initial caster and camber.

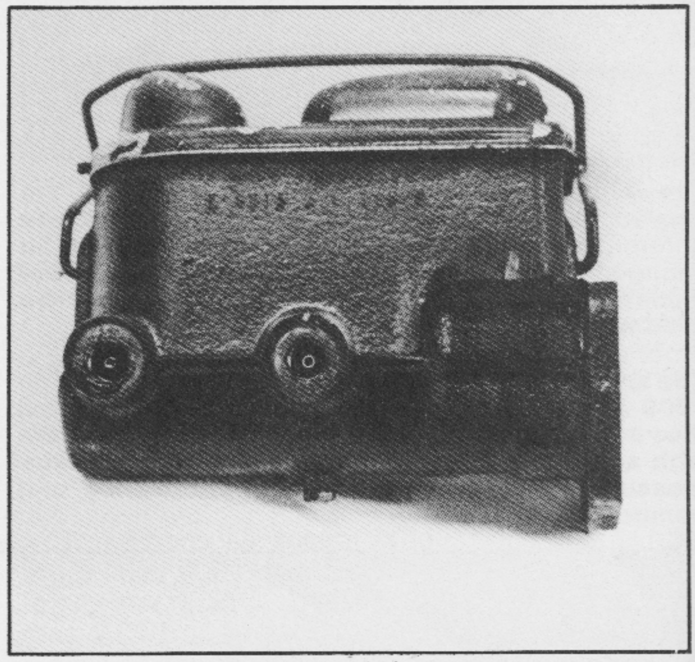


lower control-arm socket requires reaming to accept a larger lower ball joint, the same precision machine requirements apply. These specialized modifications should be left to a machine shop with stock-car or road-racing experience. (You can also contact Martin Automotive for complete assemblies, modifications to your assemblies, and replacement parts.)

DISC-BRAKE MASTER-CYLINDER CONSIDERATIONS

Converting from drum brakes to a disc-drum or all-disc system requires replacing the stock master cylinder. Drum brakes use rubber "cups" in the wheel cylinders that seal by fluid pressure, forcing the rubber "lips" against the cylinder walls. If all pressure in the system is released (i.e., when the brakes are not applied), the rubber seals tend to ooze small amounts of brake fluid. To prevent this, drum-brake master cylinders contain two residual-pressure valves, one for the front and one for the rear (in an all-drum system). When the brake pedal is completely released, these rubber "flap" valves retain a small amount of pressure in the system. This pressure prevents leaks by holding the rubber seals against the cylinder bore walls. Since the return springs are quite strong, this small pressure is insufficient to cause the brake shoes to drag against the drums. However, if residual pressure is retained in a disc-brake system—where no return springs are used—the pressure will force the pads to drag against the rotors. (The seals used in disc-brake systems do not require residual pressure for proper sealing.)

In a front-disc-brake conversion, the residual-pressure valve for the front brakes must be removed. This valve is located under the brass seat for the front brake-line fitting (usually the fitting closest to the firewall) in the master cylinder. When the brake line is



This MoPar heavy-duty disc-brake master cylinder uses a dual-reservoir design. The smaller front reservoir feeds the rear-drum brakes, and the larger rear reservoir supplies the front discs.

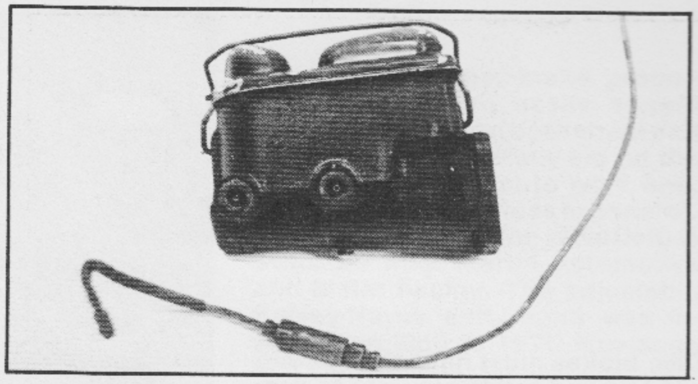
removed from the master-cylinder, the brass seat can usually be extracted by threading a sheet-metal screw (sometimes included in master cylinder rebuilding kits) into the small hole in the center of the seat. Then use two screw drivers and carefully pry the brass seat loose. When the seat is removed, you will see the rubber residual-pressure valve and spring, which you can easily withdraw with needlenose pliers. Finally, clean off any brass chips and reinstall the seat. It will be firmly reseated when the brake line is retightened to the master cylinder. This procedure should be performed for each section (front/rear) of the master cylinder that supplies fluid to disc brakes. Note: On early *disc/drum* applications, a master cylinder was used that does not contain a residual-pressure valve. It can be identified by its single fluid-reservoir design. The pressure/check valve for this system is located in the rear brake line (it looks like a small brass cylinder). No modifications are required when using this master cylinder in disc-brake applications.

Rather than modifying the master cylinder, purchasing one designed especially for disc-brake use is often a better choice. Aside from using no residual-pressure valves, the disc-brake master cylinder may use a piston diameter more suitable to disc-caliper requirements. Further, try to select a master cylinder (for disc, disc/drum, or disc/disc systems) that uses a large fluid reservoir; this facilitates brake bleeding and adds an additional margin of safety to the overall system.

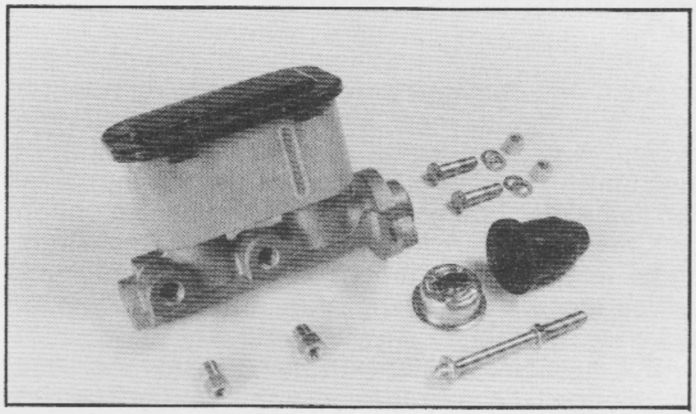
The mechanical advantage at the brake pedal (the distance the brake pedal moves in relation to the distance the master-cylinder piston moves) determines how much fluid volume and pressure can be applied to the calipers or wheel cylinders (see illustration). As the piston-rod mounting point is moved closer to the pedal pivot (higher up adds to the mechanical advantage), less pedal pressure will be required, but *more pedal travel will be required.* If the piston rod is attached lower on the pedal arm (reducing the mechanical advantage), the pedal travel will be reduced, but *the pedal pressure requirements will be increased.* The optimum balance is obtained when the piston-rod attachment point is located such that full pedal travel produces full master-cylinder piston travel. This ensures that a minimum pedal pressure will be required to deliver the maximum fluid volume. This is of particular importance (and maximizes the safety margin) during braking from high speed with hot calipers, rotors, or drums.

DISC-BRAKE SPINDLE-NUT CONSIDERATIONS

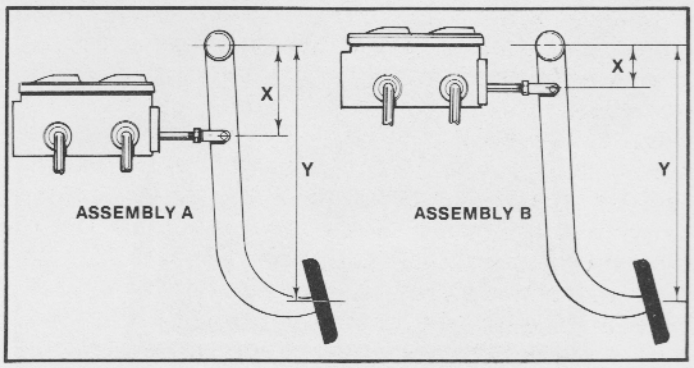
An old trick to reduce front-wheel rolling resistance is to tighten the spindle nut to a lower-than-factory torque setting. This reduces the static load on the front-wheel bearings and theoretically reduces drag caused by bearing friction. This may have some validity with drum-brake applications (particularly in drag racing), but reducing bearing preload with disc brakes will *dramatically increase overall drag, especially in road-racing situations.* Loose spindle bearings will cause the rotor (or drum) to "wobble," resulting in not only undesirable pedal feedback (exaggerated by fixed-mount calipers) but also excessive drag as the rotors contact the linings. *Always refer to the factory service manual for the correct spindle-tightening procedure for discs brakes (and*



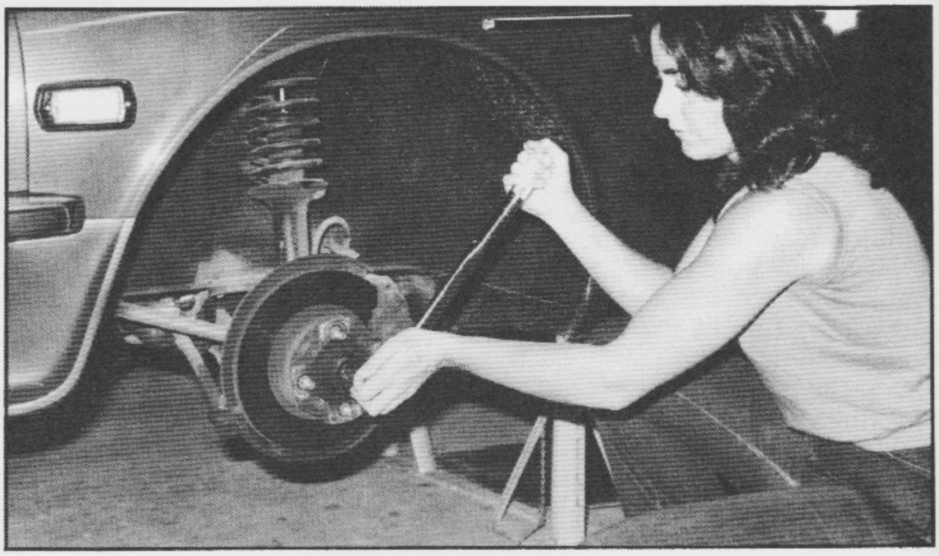
The small fitting in the brake line (foreground) is a residual-pressure value used on early disc-brake Mopars. These cars used a single-reservoir master cylinder with no built-in residual-pressure valve; when installed on front-disc, rear-drum systems, the in-line valve provided the needed pressure retention for the rear-wheel cylinders. In later models, Chrysler used a dual-reservoir master cylinder (background), that incorporated a residual-pressure valve inside the flare seat in the front reservoir (front reservoir feeds the rear-drum brakes).



Disc-brake master cylinders are available from many manufacturers. This unit was designed for late-model cars. Note the "view port" (arrow) for checking brake-fluid level without removing the cap. A master cylinder for Mopars should incorporate a 7/8- to 1-inch piston bore for optimum pedal pressure and stopping potential.



The mechanical advantage offered by the brake pedal linkage establishes the amount of pedal pressure required to operate the brakes. Pedal advantage is determined by the master-cylinder plunger-rod placement (length X) relative to the overall length of the pedal (length Y). Y divided by X will determine the ratio; the larger the number, the greater the mechanical advantage.



Disc brakes must have the spindle nut torqued relatively tight to prevent rotor wobble and unwanted pedal feedback. A "click-type" torque wrench—set to the factory torque specification—will ensure the correct pre-load and optimum brake operation.

all drum brakes used in road racing or high-speed cornering).

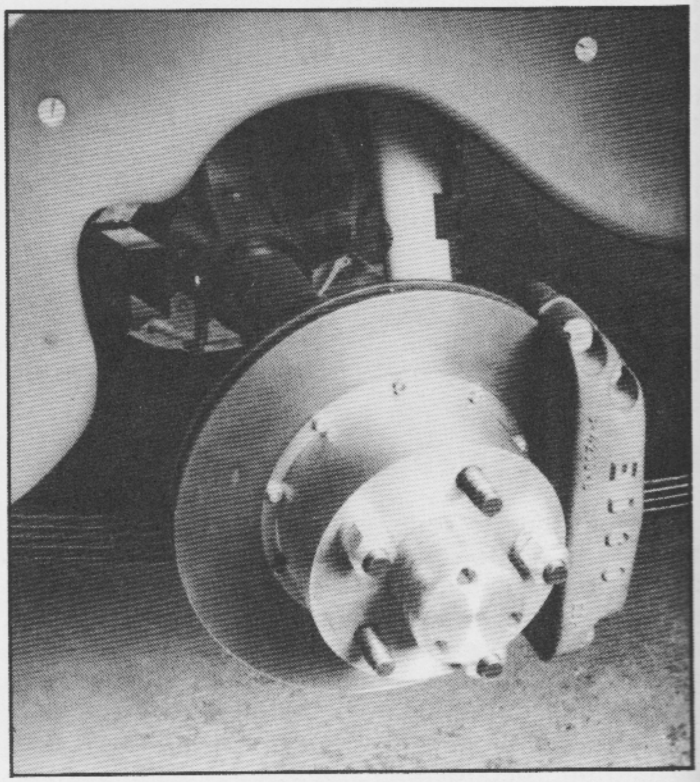
DISC-BRAKE PARKING BRAKES

All-disc systems, including the Kit Car package, do pose one common problem: what to do about a parking brake. In most installations, there is little choice. A parking brake must be fabricated from mechanical calipers, often the type used on go-karts. But some disc brakes (e.g., those used as an option on Chrysler Imperials) have a small built-in drum brake. In either case, mechanical linkage (brackets, cables, etc.) will have to be fabricated, and occasionally special adaptors may be required.

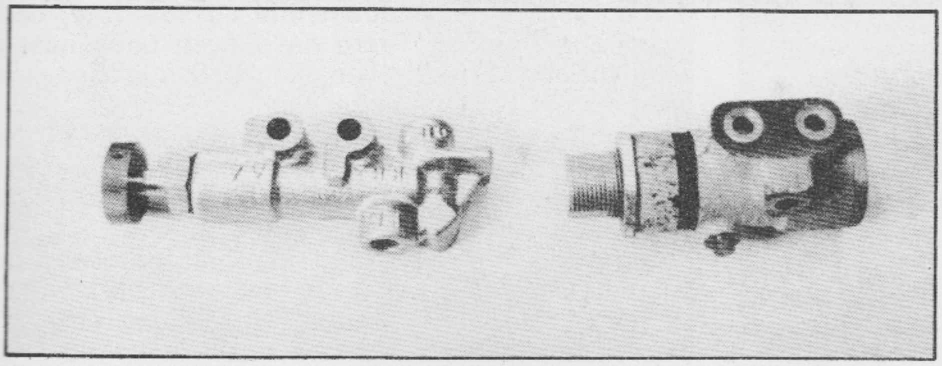
But if you're insistent on using rear-wheel disc brakes, don't *discover* that a Hurst "Line-Loc" can be used as a parking brake. Some enthusiasts have even installed Line-Locs on all four wheels. It sounds good! Just depress the brake pedal, actuate the Line-Loc, and *voila*, instant parking brake! What's wrong with that? Well, for starters, it's the electrical system that keeps the Line-Loc solenoid actuated. Therefore, a dead battery or electrical short can release the "parking" brake. But even more importantly, the Line-Loc, wheel calipers, and cylinders were not designed for continuous operation. The slightest "ooze" of brake fluid will soon release the pressure and the brakes. Simply, *do not use a Line-Loc for a cheap and easy parking brake.*

When you are considering the effort and cost

While a LineLoc is a great drag race part, it is not the parking brake "cure-all" for four-wheel disc brakes. Neither the LineLoc nor disc-brake caliper seals were designed for extended periods of pressure application.



Rear disc brakes, such as this Edco package, are often chosen for competition because of their fade resistance, rather than the slight increase in braking they offer. For most street cars a front-disc/rear-drum system is all that is required, since multiple high-speed stops are uncommon.



An adjustable disc-brake proportioning valve can easily correct brake imbalance on both street and competition cars. Many are on the market and all seem to work fine, but some are much less expensive. The Direct Connection valve (left) is available from Martin Automotive and Maier Racing. The adjustable Kelsey-Hayes unit (right) was installed on 1965 to 1970 Mustangs and Corvettes.

required for rear-disc installations and the wide variety of capable rear-drum choices, a disc-drum system is usually the most practical choice for all but the most dedicated disc-brake enthusiasts.

DISK-BRAKE PROPORTIONING

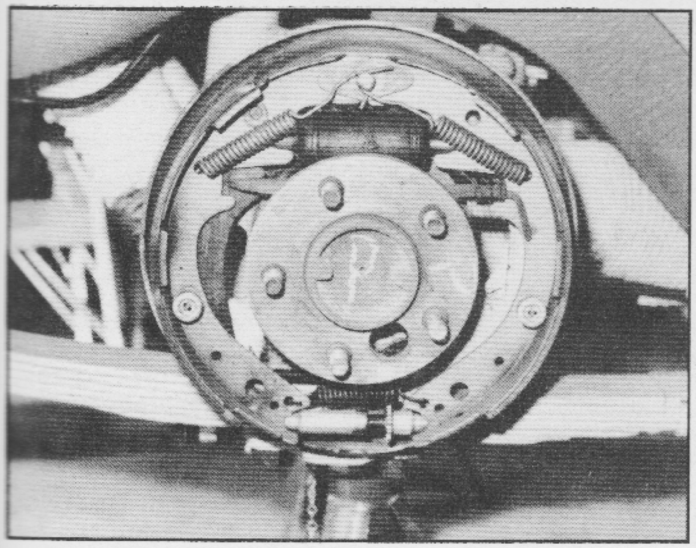
A "must-have" item for any disc-brake system, particularly a disc-drum combination, is an adjustable proportioning valve. Since disc brakes require higher line pressure to operate, drum brakes on the same hydraulic system will lock up early unless proper proportioning is provided. In fact, the disc/drum pressure requirements are so different that rear-drum lockup will occur almost immediately upon hard brake application, causing serious and dangerous handling characteristics.

All factory disc-brake-equipped Mopars incorporate a proportioning valve. On early models, it is located under the car near the rear end, and on late models it is positioned near the master cylinder. (A service manual for the particular model will aid in locating the valve.) The early-model valves (made by Kelsey-Hayes) can be adjusted by rotating the large cylindrical end clockwise (see photos) to reduce rear brake pressure and counter-clockwise to increase rear brake pressure. The late-model proportioning valves mounted under the master cylinder are "fixed" and cannot be modified.

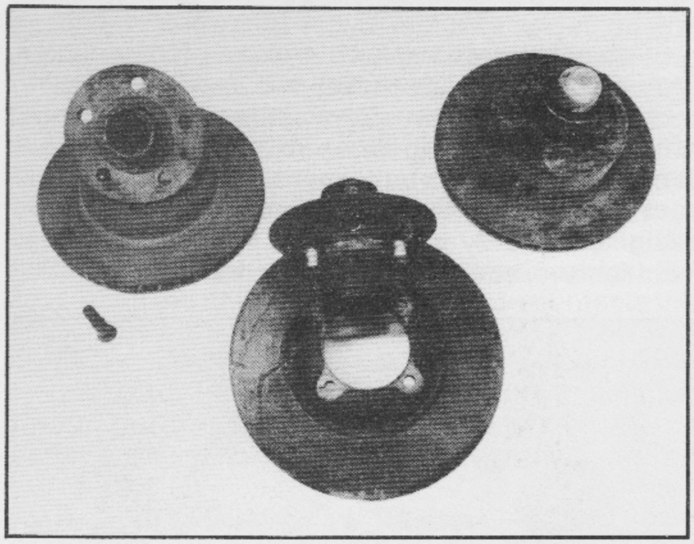
The same Kelsey-Hayes valves used by Chrysler (with a slight modification that permits additional adjustment) were supplied on early Mustang GT's and Corvettes. These valves are widely considered the best choice for performance work, and they are still available from Ford and Chevrolet Parts and from many high-performance shops. They can even be located in wrecking yards that handle Mustangs and Corvettes. Chrysler also offers an adjustable brake-proportioning valve designed for performance applications. It is available through many Direct Connection dealers and from Martin Automotive Design.

SUMMING UP THE MECHANICALS

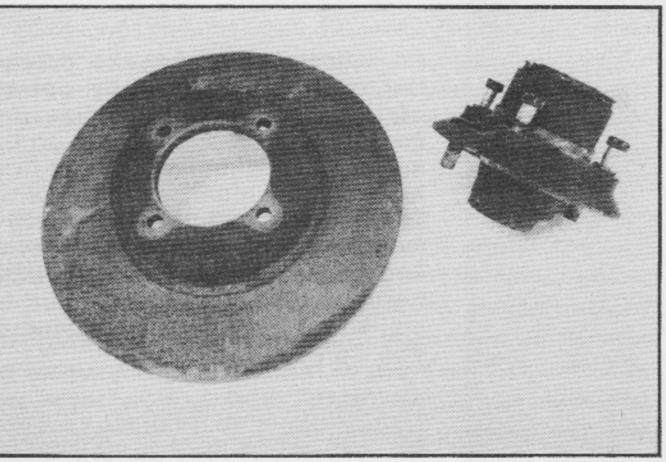
An excellent A-Body street braking system should incorporate a disc-drum combination using the 8-3/4-inch rear end with either stock 4-inch or the larger 4-1/2-inch bolt pattern. You can use 14 x 6-inch rims and DR- or ER-60 series tires, all of which will fit under the stock A-Body fenders. For all-out A-



This 11-inch drum-brake assembly—used on a competition car—has been adapted to an A-body 8-3/4-inch rear end. The installation required only one modification: conversion to larger axles with 4-1/2-inch bolt circles; all remaining parts are bolt-ons.



Detroit has installed several different disk-brake rotors on passenger cars (left to right): H-type rotor (pressed together on lugs), L-type rotor (bolt together), and J-type rotor (unicast). The J- and H-type rotors are the most commonly used on domestic automobiles.



Slightly elongating the rotor mounting holes will permit the hub and rotor to expand and contract independently, reducing rotor distortion and hub failure. However, the holes must only be enlarged *radially*; drilling out the holes will allow the rotor to "work" back and forth on the studs, causing brake noise and potential rotor or stud failure.

Body brakes, use a 4-1/2 bolt pattern and the larger B-Body disc-drum brakes. B-Body and E-Body cars must use either the disc-drum combination (very adequate for an A-engine car) or the 11-inch drum brakes on all four wheels. Because of the added weight of B- and E-Bodies, nothing less is acceptable. The ultimate *stock system* for all A-, B- and E-Body cars is the C-Body police-car disc/drum system.

CALIPERS AND ROTORS

Chrysler has produced three basic calipers and two basic rotors for disc-brake cars. Each design has its strong and weak points. However, they are all adequate for street use; and as mentioned earlier, some pieces can be successfully used in competition.

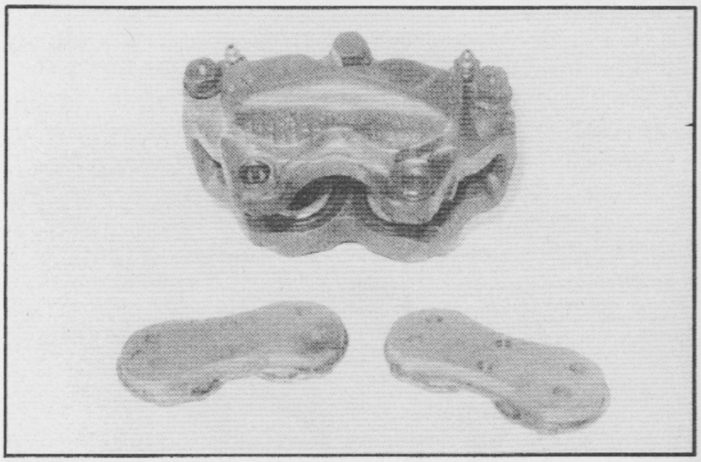
Within the three basic catagories of rotor design, Chrysler has manufactured two models. The "L" and "H" rotors are two-piece units while the model "J" is "uni-cast"; i.e., the hub and rotor are not separate assemblies but cast as one piece. The uni-cast "J" rotor is the most common of the three designs, primarily because it is less expensive to manufacture, and is quite suitable for street use. But in competition applications, the uni-cast rotor can warp and fracture. There have even been cases where the rotor has broken off the hub due to severe loading and heat warpage.

These problems are virtually non-existent—or can be easily solved—with the "L" and "H" models since the rotor and hub are separate assemblies. Bolts secure the rotor to the hub on the "L" design, while the model "H" is pressed together and held in alignment by the wheel studs. Most racers prefer the "L" or "H" rotors, but the most desirable unit is the "L" type since the rotor can be easily unbolted and replaced.

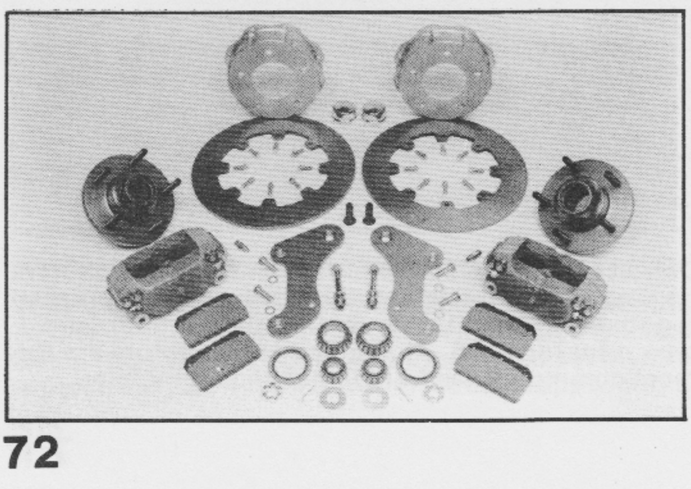
Rotor failure or damage is usually due to excessive heat. As the temperature increases, the rotor tends to expand; but when the rotor is attached to the hub (as with the uni-cast model), the expansion is restricted and the rotor tends to warp and wobble. But the "H" and "L" two-piece models allow the rotor to expand more freely, reducing rotor stresses. Heat-induced loading can be further reduced by radially elongating the wheel-stud holes in the rotor. This modification permits the rotor to expand and contract independently of the hub and can significantly reduce rotor warpage at high heat levels.

There are three common caliper designs: fixed, floating, and sliding—and Chrysler has used all three. The fixed caliper is rigidly mounted to the spindle, while floating and sliding calipers are "free" to move side-to-side as the rotor flexes and bends under load.

Fixed calipers are manufactured in 2-, 4-, and 6-piston configurations. The 2- and 4-piston models

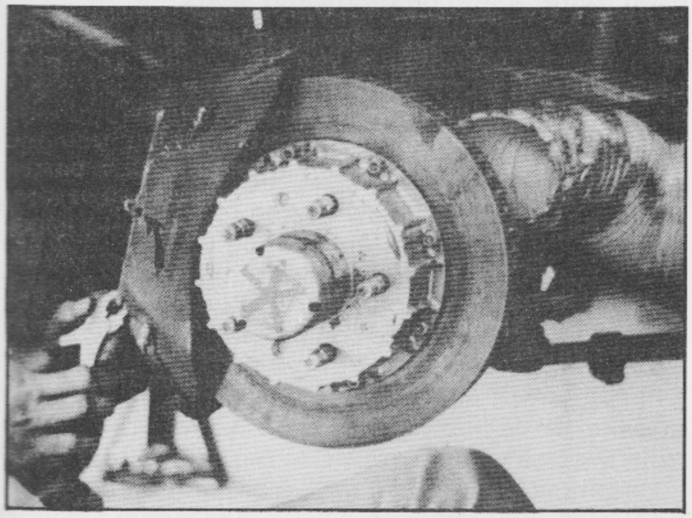


The AirHeart 4-piston caliper and rotor assemblies are used on many Trans-Am cars. While single and 2-piston calipers are fine for street and short courses, the greater stopping power, fade resistance, and swept area of a 4- or 6-piston design is required for longer tracks and endurance racing.

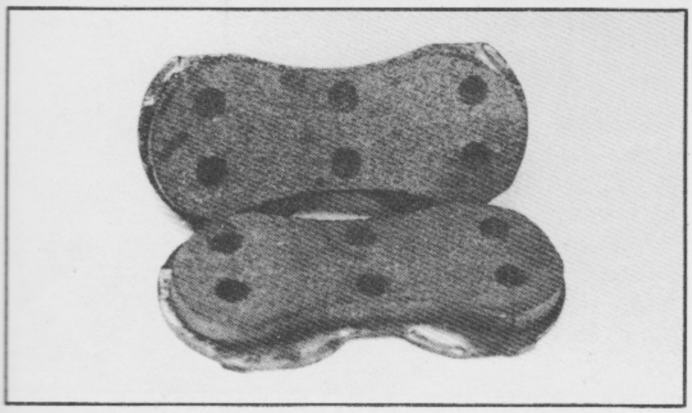


Competition brake systems can cost well over $1000. This complete front-brake package uses special rotor carriers ("tubs") that slide over the hubs and permit the rotors to expand without warping. Also included are four-piston calipers with metallic linings (although the linings are not riveted; I recommended using *bonded and riveted* linings, see text), caliper supports, wheels bearings (for the special hubs), and all attaching hardware.

Although autocross races are run on tight, relatively low-speed courses, an excellent braking system is required to be competitive. Mopars should use front discs, 11-inch rear drums, bonded and riveted pads and linings, and a 4-1/2-inch bolt circle on all axles.



Many calipers permit the pads to be changed very easily. By extracting a small retaining pin (arrow indicates location), the linings can be removed from the top of the caliper. Cool air is ducted from the front grille and can extend brake life and efficiency for "road work."



Although these pads are fully metalic—designed for competition road racing—the bonded and riveted feature should be considered a requirement in ALL high-performance applications. The combination of bonding and riveting will prevent the pads from loosening at high temperatures and speeds.

are more common, but Chrysler exclusively used the 4-piston caliper on early fixed-caliper-equipped cars. However, the fixed-caliper system is more complicated than it appears on the surface. Chrysler used three different manufacturers to obtain fixed calipers, and each manufacturer applied its own "theory" to caliper design. The smallest of the three is the Kelsey-Hayes unit used on A-Body cars (and some B-Body cars with smallblock engines). A larger caliper, supplied by Bendix, was used on most of the big-block B-Body cars from 1965 to 1969. And the largest, the 4-piston fixed caliper, was built by Budd and used on 1965 to 1972 C-Body cars, some pickup trucks, and some vans. None of the three models is interchangeable; *the entire spindle, rotor, and caliper assemblies must be changed as a unit.* Obviously, the larger the caliper, the better the braking; and "upgrading" always means replacing the smaller calipers with larger models. Never go the other way!

Four-piston fixed calipers are the best overall choice for both street and competition use. While they are heavier than their sliding or floating counterparts, they are far stronger and provide better stopping power. The wider area across the four pistons allows using a longer and larger brake pad and ensures even pressure distribution. The main drawback with fixed calipers can be their undesirable pedal feedback, but this is evident only when something is out of alignment. Warped calipers, improper rotor alignment, or loose rotors can cause brake "chatter" and uneven pad wear. These problems can be corrected. However, if left unattended, they can result in unstable and unsafe braking.

Unlike fixed calipers, floating and sliding calipers can move side-to-side on the spindle mount, greatly reducing rotor alignment problems and the accompanying pedal chatter. The *floating* caliper is held in place on the spindle by retaining clips, but the caliper assembly is free to move (actually slide) on flat portions of the mount. The *sliding* caliper, on the other hand, is positioned by two pins and two bushings; the caliper slides on the mounting pins. Rather than the opposed pistons used on fixed calipers, the floating and sliding calipers use a piston and "anvil" to squeeze the rotor. This design is similar to a cupped hand with the piston on one side. As the piston moves outward, the pad presses against the rotor, then the anvil—with its own pad—contacts the rotor; they both "squeeze" and force the rotor to a stop. The operation is very simple, efficient, and requires less manufacturing cost than multiple-piston designs. But because of reduced piston surface area and the reduced metal mass in the caliper, the braking forces under high loads are not equal to those produced with the 4-piston fixed caliper. However, for high-performance street use, the lower cost, easier maintenance, self-alignment, and very capable braking efficiency make them an excellent choice.

The impressive Kit Car disc-brake assembly uses a floating caliper on the front (to minimize drag and

Back in "the good old days" of sedan racing, competitors would "run what you brung." Just pop off the hub caps, crack open the exhaust, tape the headlights, and more often than not, completely ignore the brakes and brake linings. Even moderate driving on hills or back roads will overheat stock linings, but the rigors of road racing will completely overpower stock brakes. A few reasonably inexpensive modifications, however, can prepare any Mopar to stop effectively on the street or at the track.

Photo by Toni Cortes

cost) and a fixed caliper on the rear. The front calipers and rotors are very similar to those used on 1972 and 1975 vans and pick-up trucks. This is an excellent braking system, but beware; the Grand National Kit-Car caliper (and some C-Body calipers) use an extra-large 5-inch bolt pattern, instead of the usual 4-1/2-inch Mopar bolt circle. Check the parts carefully before you order.

FRICTION LINING MATERIAL

Whenever the braking system is required to perform more work, installing a better lining material should be considered one of the required modifications. Improved linings are made with a superior harder-than-stock lining compound. Stay away from cheap replacement linings that will easily burn and crack; they are never recommended for high-performance use. Better linings are often *semi-metallic* in construction, providing an improved "grip" on the drum surface during braking—at a slight loss in brake-drum/rotor life. *Fully-metallic* linings are also available; although they are almost fade proof, they do not stop well when cold (sometimes almost not at all). Do not use fully-metallic linings on street-driven vehicles; they can be very dangerous until warm. However, for competition use, there is no better lining available. But regardless of material, the replacement linings should be both *bonded and riveted.* Bonding and riveting will ensure that the linings *cannot* come loose under high heat and high load. Linings attached only by bonding can loosen under high stress and cause brake failure.

Brake linings are only a part of the overall system, however, and do not make up for inadequate drum size, etc. Although high-quality linings will improve braking, the brakes will still fade if the swept area is inadequate. Excessive heat with insufficient dissipation area will "toast" even the best linings. And high heat levels can cause the brake fluid to boil in the caliper or wheel cylinder. In all cases, this leads to "spongy," non-responsive brakes and often to permanent damage to the drums, linings, fluid seals, and even to the wheel bearings and seals.

In high-performance applications, the brake-lining material should be riveted to the shoe as well as bonded. Linings that are only bonded (with an adhesive) can fail when subjected to high temperatures for long periods. If your local parts source cannot supply semi-metallic, bonded and riveted linings, check the Yellow Pages for distributors of Grey Rock, Raybestos, and Girling products. These manufacturers make excellent high-performance lining materials. You can also look up a competent road-racing shop, where semi-metallic linings can be bonded and riveted to your old brake shoes.

Finally, you can extend both braking efficiency and life by adding cool-air duct work. Directing air from the front of the vehicle to the drums/rotors will substantially cool and improve braking on competition vehicles or street cars that are subjected to hard driving, particularly on tight, winding roads. An important consideration in ducting installation is to use a tube size that will clear the tires, suspension, etc. Direct the air blast at the backing plates (some

It's just amazing what lurks beneath the sleek body on this Trans-Am Porsche. The air ducting alone is a work of art; air ducts for the brakes, radiator, and intercooler are all carefully fabricated to clear suspension movement while providing efficient cooling for reliability and horsepower.

competition models plumb the ducting directly into the backing plates) and secure the duct work with aircraft-type clamps. Neatness counts. (Note: Remember to block the air ducts during wet weather. Water will significantly reduce stopping power, particularly with drum brakes.)

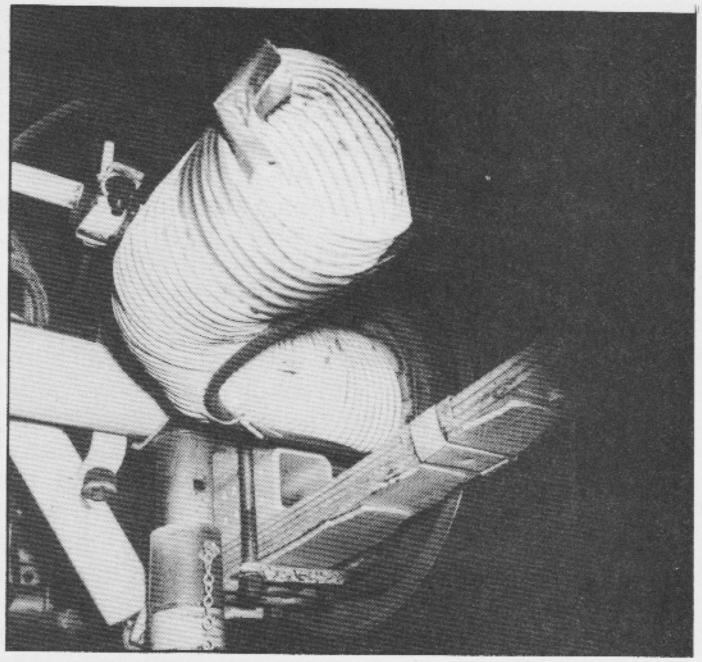
BRAKE FLUIDS

This will be short and sweet. *DO NOT USE SILICON BRAKE FLUID* for street or competition. Use only standard, D.O.T.-approved disc-brake fluid, and then select the highest temperature rating you can find. Silicon brake fluids just aren't any better than standard disc-brake fluid (except in some special competition applications where their higher boiling temperatures are an advantage). There is also evidence that silicon brake fluids absorb water too quickly for any street applications. Finally, the cost is rather prohibitive for general street use (it is far more expensive than D.O.T.-approved disc-brake fluids).

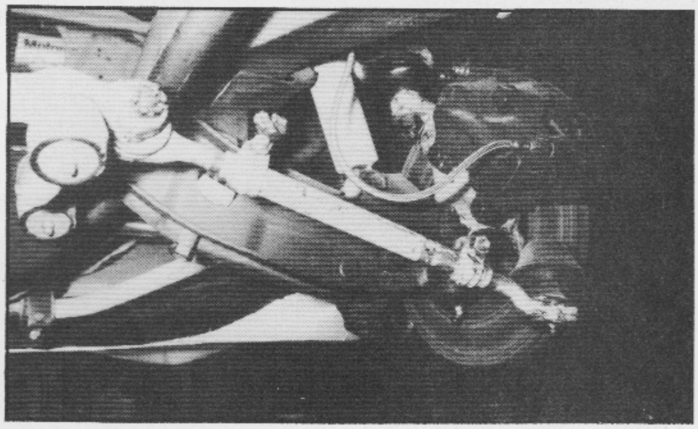
You can use disc-brake fluid on drum-brake systems. You cannot use drum-rated brake fluid on disc-brake systems because the fluid will not operate at the temperatures found in disc-brake calipers (it can boil).

BRAKE LINES

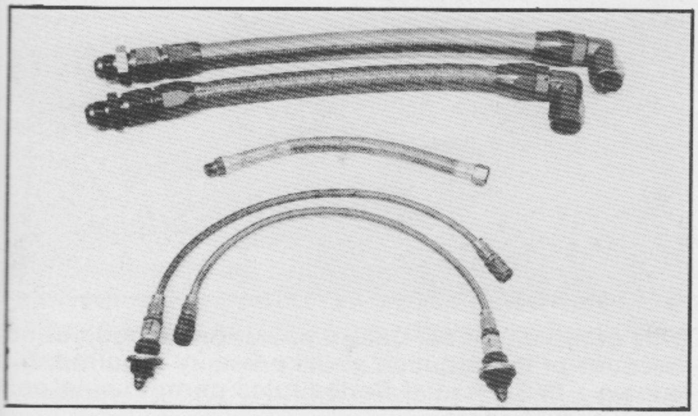
Recently, a lot of emphasis has been placed on using aircraft-type flexible lines in automotive applications. These flexible lines can be successfully used *if the proper type of line is selected.* When ordering/purchasing flex line for brakes, be sure to specify the intended application. There are several different materials used to make stainless, reinforced aircraft line, but only one type should be used in braking systems. The correct type is available from several manufacturers, like Aeroquip (Part 2807-4 or 2807-3) and Earl's Supply (teflon-lined -3 for 3/16-



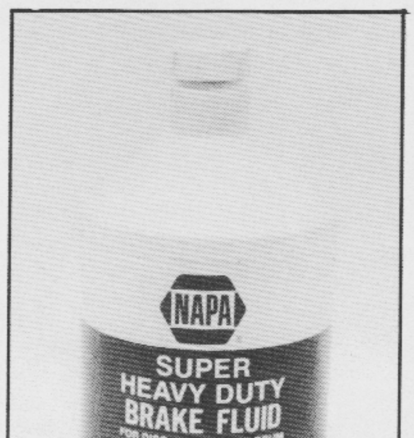
Simple but efficient ducting for this Barracuda rear-brake system draws air from under the car into the drums through the backing plates. The ducts are attached to the front portion of the backing plates, and an open area on the rear of the backing plates exhausts hot air.



Large brakes installed on this A-body racer incorporate fixed calipers and braided-stainless line.



Braided-steel reinforced Teflon brake lines are a must for competition cars. They also work well on street cars; however, even though braided line is of higher quality than factory rubber hose, it must be kept clean and periodically checked for damage or kinks. Teflon hose life can be extended by enclosing it in protective rubber tubing that reduces wear from metal contact and oil spills.

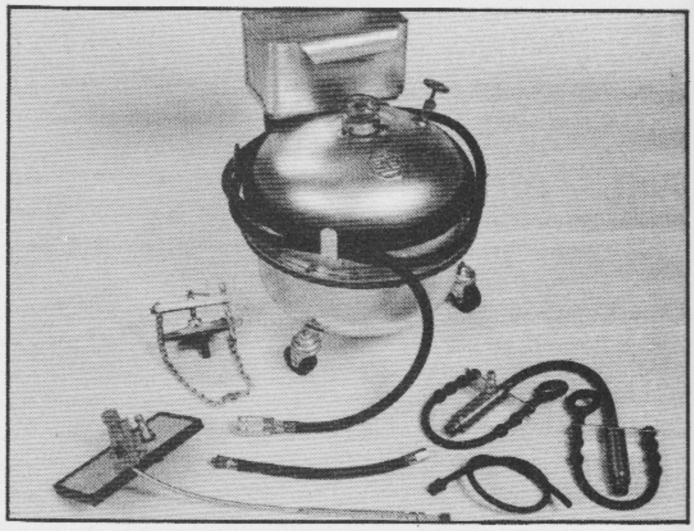


The label on this can of D.O.T.-approved brake fluid states, "For disc brakes, for drum brakes." You can use disc-brake fluid for drum brakes, but *never* use drum-brake-only fluid for disc brakes; higher disc temperatures can cause the fluid to boil.

inch and -4 for 1/4-inch diameter). The -4 will work for most applications, but under extremely high pressures (like a Bonneville or Indy car), the -3 line will expand just a bit less. Nothing else will work! You will also have to purchase adaptors to attach the lines to automotive brake fittings. All of these parts are available at better race shops. (Also contact Earl's Supply in Lawndale, California— [213] 772-3605. They are willing to discuss your requirements.)

Use *ONLY thick-wall steel tubing for brake lines*, except where flex lines are used. Do not use flex line for the entire brake-line system; it will allow far too much expansion and result in a spongy pedal and loss of braking power. The thickwall steel tubing (stainless steel is preferred) must have double-flared ends to prevent leaks and safely withstand the pressure levels. It costs more than standard steel brake line, but it has less flexibility than the standard steel tubing and is the only type of tubing to use for competition applications or all-out street applications. Follow these simple guidelines, and your brake system will be reliable, perform at top efficiency, and have a long life.

Finally, after doing any work on brake hydraulics, always bleed the system with clean fluid (preferably using a pressure bleeder.) This is even more important with disc brakes since they operate with less fluid mass and at higher pressures. Any air in the lines will result in a significantly softer pedal. Remember, your brake system may have all the right parts, but it just won't work well if there's any air in the lines.



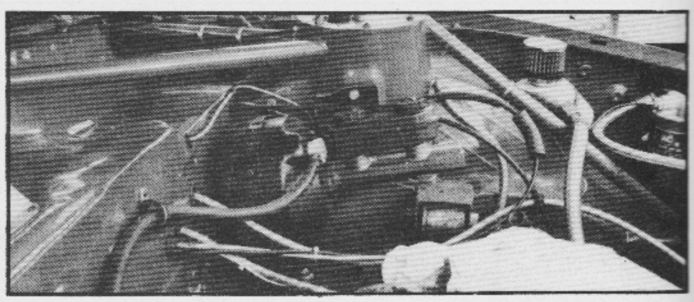
A pressure bleeder maintains a continuous pressure and supply of brake fluid during the bleeding process—eliminating the last few air bubbles in the system. This is of particular importance with disk brakes, where even the slightest amount of trapped air will significantly reduce braking.

POWER ASSISTED VERSES MANUAL BRAKES

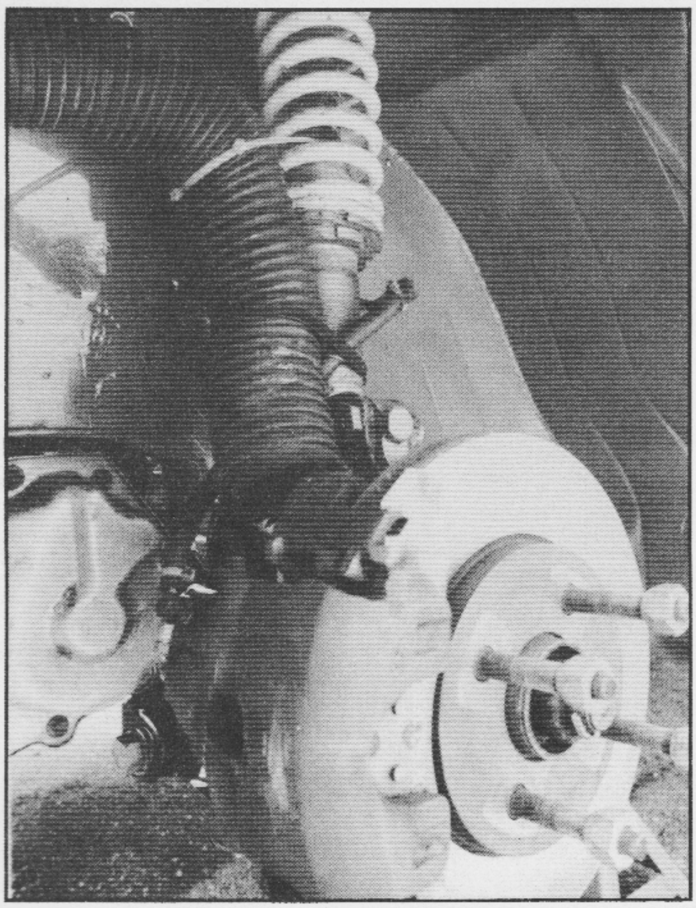
First of all, a power-assist doesn't improve braking; it merely reduces brake-pedal effort to achieve the same braking force. Power assist is most useful on disc-brake cars, where the higher line pressure required to activate discs can make the pedal feel unresponsive.

But disc brakes do not *need* power assist. In fact, very few race cars with four-wheel, disc-brake systems are power-assisted because manual brakes allow the driver to more easily control pedal pressure (therefore, line pressure) to obtain even braking.

If you are building a competition car, a power booster should be considered strictly optional (and perhaps undesirable—just another part to fail!). If the power brake booster is removed, install the largest manual-disc master cylinder available.



This competition car uses a power-brake booster not because of the reduced pedal pressure required, but because IMSA Radial Series rules permit using only stock components.



Air ducting on this professional front-wheel-drive "RS" car is carefully positioned to prevent interference with the tires and wheels. The stainless-braided line used on the front calipers is protected by rubber tubing.

MEASURING BRAKING FORCES

In a previous chapter, we discussed lateral acceleration and the term used to express this variable: "Gs" or "gravities." A similar method of measurement and the same term can be used to express braking force or *deceleration*: The higher the "G" force generated by the brakes, the better the braking system works. But braking forces are also measured in another unit—horsepower. Most brake manufacturers evaluate their systems on electric "dynos" that measure the maximum horsepower absorption developed by a specific design. Although this does not *guarantee* that the same system will perform equally well on a moving chassis, it is a good indicator of braking capability. The following formula will determine the WORK generated by the brakes to decelerate a vehicle of known weight and initial speed:

FORMULA 21

$$\text{WORK} = \frac{W \times (\text{MPH} \times 1.47)^2}{2G}$$

Where **WORK** is equal to the foot-pounds of kinetic energy produced during the speed change of the vehicle, **W** is the gross weight of the vehicle, **MPH** is the speed of the vehicle in miles per hour, 1.47 converts MPH to feet/second, and **G** is the force of gravity measured at 32.2 feet/second/second. Using this formula with a vehicle weight of 3200 pounds and a speed of 70 mph, the formula produces:

FORMULA 22

$$\text{WORK} = \frac{W \times (\text{MPH} \times 1.47)^2}{2G} = \frac{3200 \times (70 \times 1.47)^2}{2 \times (32.2)} = 526{,}132 \text{ ft-lbs}$$

Now that the work in foot pounds is known, we can determine horsepower. Horsepower is defined as the amount of work done over a known time period. The work done is 526,132 foot-pounds, and we will assume that the deceleration to a stop took 7 seconds. In addition, 550 foot-pound/seconds equal 1 horsepower, so we will include this conversion factor in the denominator of the equation. As our 3200-lb test vehicle decelerates from 70 mph to a complete stop in 7 seconds, the horsepower produced would be:

FORMULA 23

$$\text{HORSEPOWER} = \frac{\text{WORK (ft-lbs)}}{\text{TIME (sec)}} \times \frac{1}{550 \text{ (ft-lbs/sec)}} = \frac{526132}{7 \times 550} = 136.6 \text{ hp}$$

Translated, this says: A steady 137 horsepower must be absorbed by the braking system during the deceleration of a 3200-pound car from 70 mph to a stop in 7 seconds. And note that the work required (from Formula 22) will increase by the square of the speed. For example, to stop a car moving twice as fast—140 mph—Formula 22 will determine that 2,104,529 foot-pounds of work will be required. In other words, four times the work is required to decelerate the car from just twice the speed.

BACK TO G's

If the time it takes a vehicle to decelerate to a stop and the initial speed are known, a simple formula will calculate the negative acceleration (deceleration) in Gs. For our example (deceleration from 70 mph in 7 seconds):

FORMULA 24

$$-\text{Gs} = \frac{-\text{VELOCITY}}{32.2 \times \text{TIME}} = \frac{-(\text{MPH} \times 1.47)}{32.2 \times \text{seconds}} = \frac{-(70 \times 1.47)}{32.2 \times 7} = -.45\text{G}$$

However, if the time of deceleration is not known but the stopping distance has been measured, Formula 25 will calculate deceleration gravities. For example, if a vehicle slows gradually from 70 mph and reaches a stop within 365 feet, the deceleration Gs are:

FORMULA 25

$$-\text{Gs} = \frac{-(\text{MPH} \times 1.47)^2}{64.4 \times \text{distance}} = \frac{-(70 \times 1.47)^2}{64.4 \times 365\text{-feet}} = -.45\text{G}$$

Now if the same car "panic-stops" from 70 mph within only 180 feet:

FORMULA 26

$$-\text{Gs} = \frac{-(\text{MPH} \times 1.47)^2}{64.4 \times \text{distance}} = \frac{-(70 \times 1.47)^2}{64.4 \times 180\text{-feet}} = -.91\text{G}$$

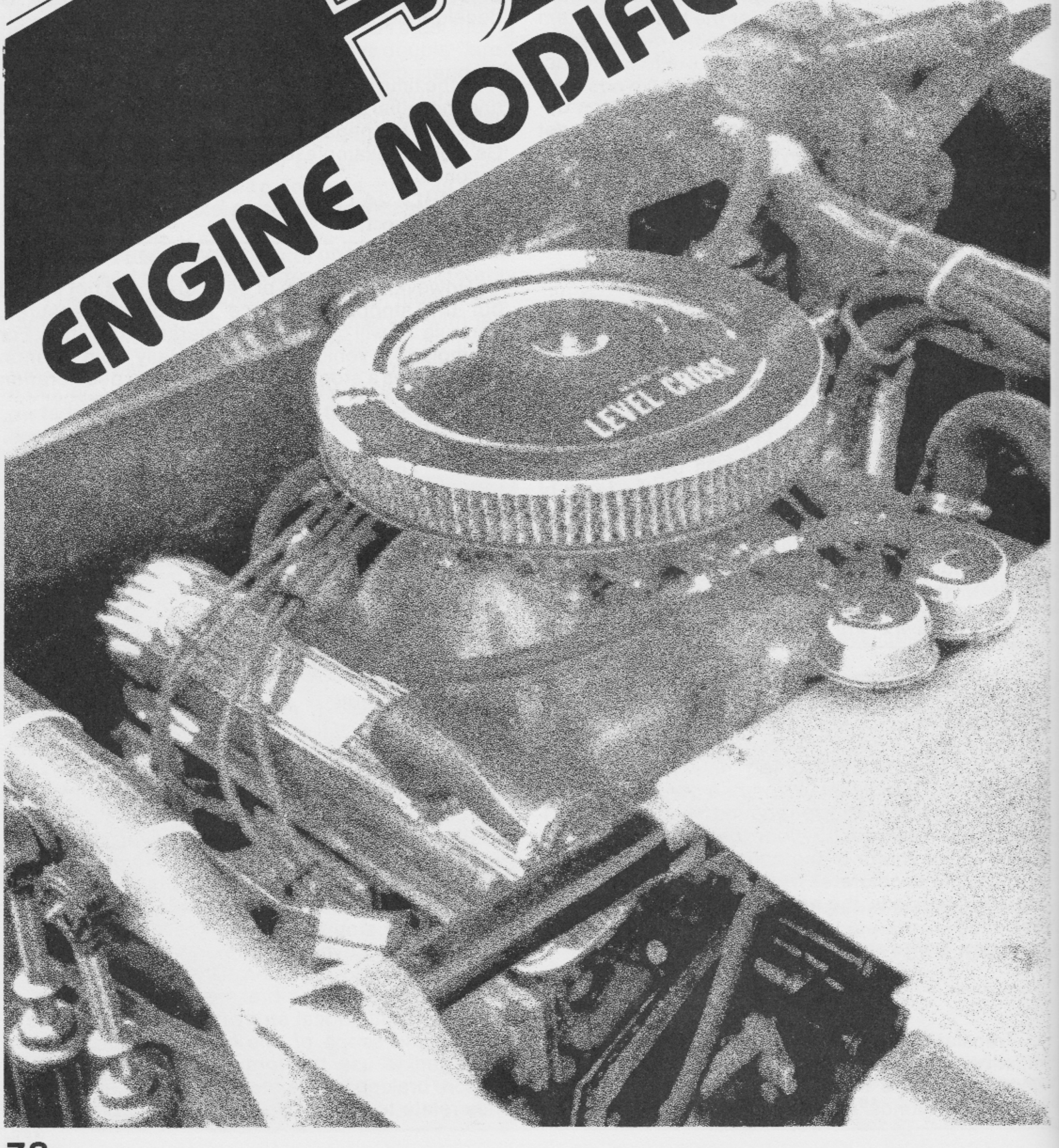
The resulting .91 G of deceleration is a very respectable figure.

Experimenting with the preceding equations will help you understand and evaluate braking loads. If you are lucky enough to find horsepower testing data provided by the brake-system manufacturer, you can perform some numerical analyses by backstepping through the equations and evaluating how a particular car might stop with specific hardware.

But braking dynamics are complex. As a vehicle decelerates, a good portion of the weight shifts to the front and complicates brake analysis. Although an exhaustive numerical study of braking dynamics is beyond the scope of this book, the reader is encouraged to use the formulas presented here to explore braking forces, stopping distances, and how they relate to vehicle weight and speed.

CHAPTER 5

ENGINE MODIFICATIONS



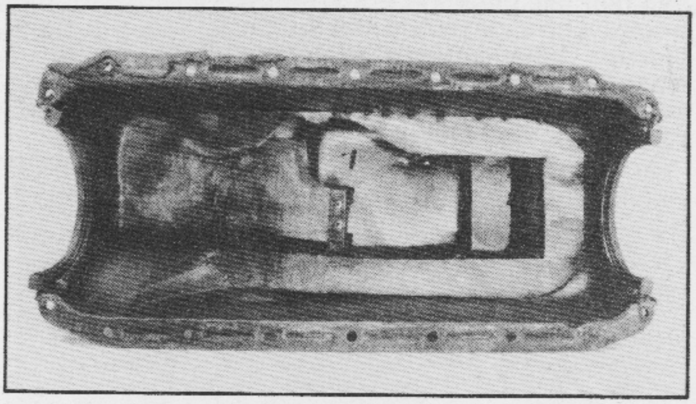
ENGINE MODIFICATIONS FOR HANDLING

Up to this point, we have discussed the ways you can make your Mopar handle better, stop better, and even get better gas mileage (by applying the principles of optimum front-end alignment and aerodynamics). But one final subject remains to be covered: engine modifications required to ensure good performance and reliability. (Note: Refer to the S-A Design book *MOPAR PERFORMANCE* for additional Mopar A- and B-engine building tips.)

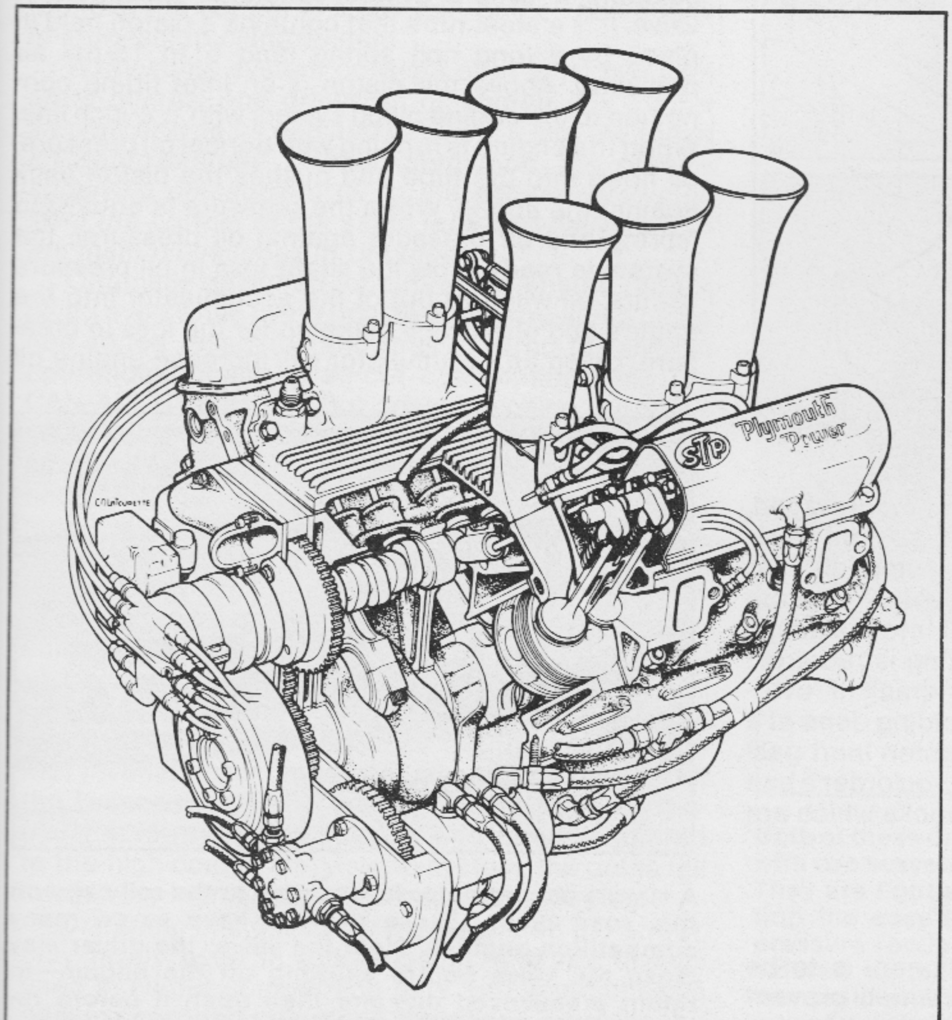
ENGINE OILING

The most critical area of modification in a road car is engine oiling. High cornering loads can force oil in the pan away from the pickup, not just to the rear—as in a drag car during acceleration—but in all directions. These unpredictable loads complicate oiling-system design, but the problem is not insurmountable. The following guidelines will help you build a reliable oiling system.

An oil pan for road racing and high-performance street use must be thoroughly baffled in the front, rear, and sides to prevent oil starvation during cornering, acceleration, and stopping. The generous ground clearance in a 1/4-mile drag car greatly simplifies pan design; just go deep. The deeper sump allows a larger oil volume and "captures" the oil, directing it to the pickup located at the rear of the sump. But the lower front end of a road car completely rules out a deep sump. To obtain extra sump volume, the pan can be made wider where the chassis and suspension will permit; but most importantly the baffle design must retain oil around the pump pickup. Chyrsler Engineering has developed a recommended design for baffle and sump construction (see illustration). But regardless of the precise baffle design, a windage tray should be considered an essential ingredient. It speeds oil return to the sump, provides a slight horsepower increase and acts as an upper-sump baffle, helping to reduce oil

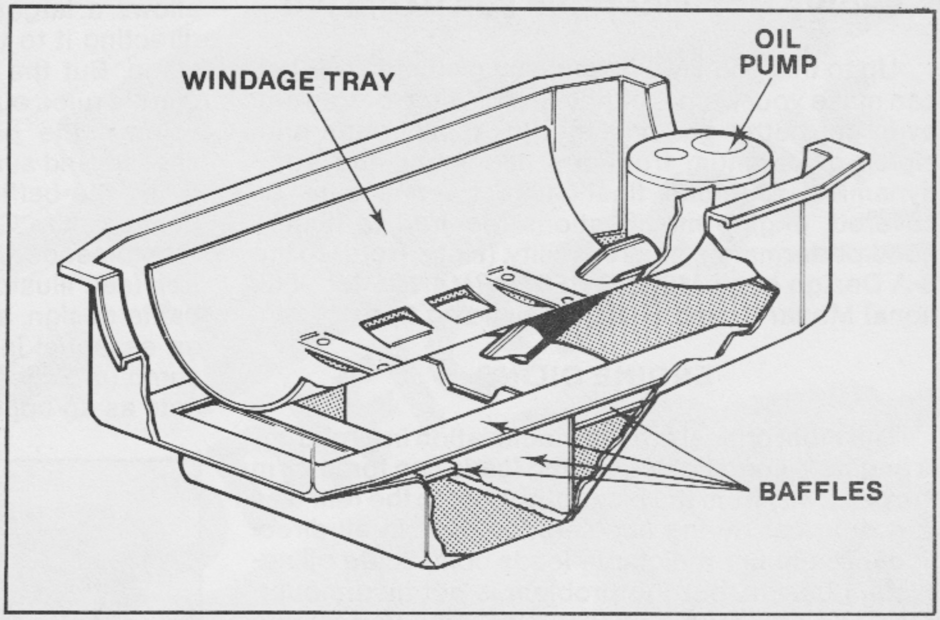


A simple but effective baffle system—which works in conjunction with the factory windage tray—has been added to this stock A-engine oil pan. While not incorporating all the lastest technological tricks, it functions well in a street-and-weekend racer.

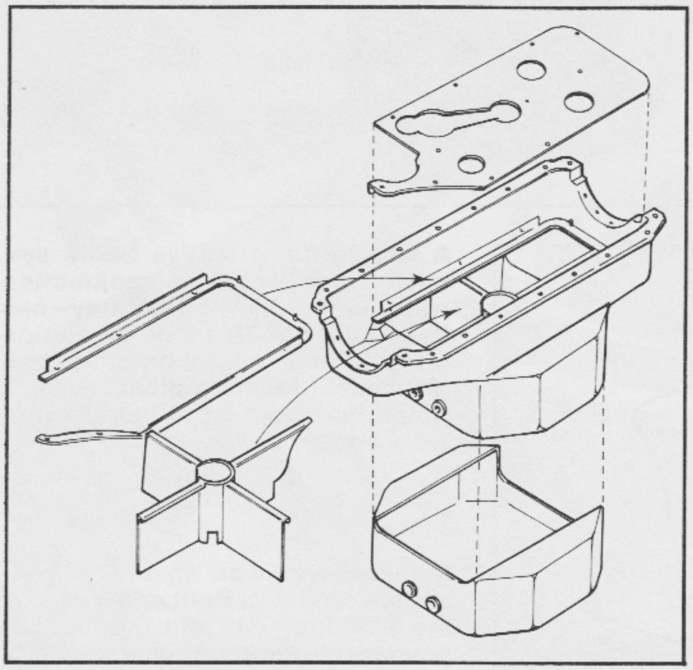


If there ever was an ultimate A-engine, this is it. Built originally for the STP Indy car and the Dodge-powered Formula 5000 cars, this all-out masterpiece uses a dry-sump oiling system with a *gear driven* scavenge and pressure pump. The distributor is directly driven off of the camshaft sprocket. The injectors may not be the most practical setup for a street car, nevertheless this would be a great crowd-pleaser in an early A-body Mopar.

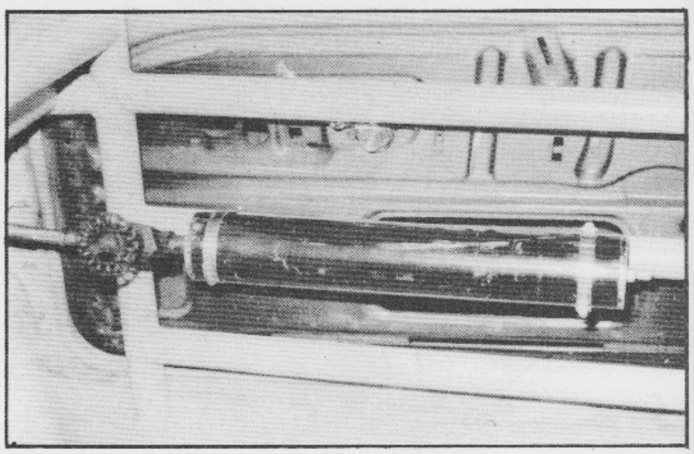
The baffling illustrated at the right is designed for a stock front-sump, A-engine oil pan. Make sure that the large baffle that runs the full length of the pan will clear the windage tray and the oil pump pickup tube. This is an ideal pan for street or amaetur road racing.



The modifications shown below are recommended by Chrysler for a competition wet-sump oil pan. The baffles and deep-sump section are added after all but 3-5/8 inches of the stock ban (measured from the pan rail) has been cut off. A baffle cover—with a clearance slot for the pickup tube—virtually seals the extended sump. This pan, like the milder version on the right, is designed to be used with a windage tray.



most of the time, but how can we eliminate the slight loss of pressure that will occur during particularly hard cornering or evasive maneuvers? Perhaps you're saying, "Who cares?" On a street machine, your comment is probably appropriate. But in road racing and autocrossing a slight loss of oil pressure at 6000 to 7000 rpm can be disastrous. There is a solution: a simple device called an accumulator valve. It is a steel tube that contains a piston held in place by a long coil spring (and 5 to 15-psi air pressure). Above the piston is an inlet fitting connected to the engine oiling system with 1/2-inch line. When the engine is running with normal oil pressure, oil flows into the tube and pushes the piston back against the spring. When the pressure is equalized (spring and air pressure against oil pressure), the system is ready. Now if a slight loss in oil pressure occurs, oil will flow out of the accumulator into the engine and effectively make up for the loss in pressure. Using an accumulator will increase engine oil aeration during hard deceleration.



A simple accumulator is mounted to the roll cage on this road racer. These devices have saved many competition engines. Note the valve; the driver may close the valve before shutting off the engine—to retain pressurized oil—and then open it before restarting to pre-oil the engine.

Because Mopars are no longer commonly raced, pan modifications will usually require custom work. However, Moroso and Milodon offer sump kits that can be added to a stock pan. Since the intricate baffles and/or doors are already built into the sump, only a minimum of cutting and welding is required. But because oil-pan material can crack if overheated, it is advisable to have any welding done at a professional shop that has TIG (tungsten inert gas) welding equipment. And be sure that you order a pan kit for *road racing*, not one for oval tracks which are designed for left turns only.

OIL ACCUMULATORS

Oil starvation, as we have mentioned, is to be avoided at all costs. A well-designed pan will prevent loss of oil pressure during cornering and stopping volume requirements, since the accumulator and the attaching lines can retain 2 to 3 quarts of oil. In other words, the entire lubrication system could require up to 12 quarts of oil (this assumes a larger sump, a 2-quart accumulator valve, and 1/2-inch lines about 3 or 4 feet in length).

The accumulator valve is most commonly marketed under the "Accusump" name in various sports- and race-car catalogs. You can also find accumulator valves in industrial supply and hydraulic-equipment parts stores. Carefully check out the prices and capacities before you spend your money. Whichever model you install can provide an extra edge of insurance that may make the difference between taking the checkered flag and being an "also ran."

DRY-SUMP OILING

A well-handling car can generate about 1G of lateral acceleration. With these loads to contend with, even the most expertly baffled oil pan will allow some air to reach the pump pickup. Any air in the lubrication system can be potentially harmful to crank and rod bearings at high rpm. And if air cannot be prevented from entering the oil-pump inlet, the best solution is to move the inlet to a position/place where it can only draw air-free oil, regardless of lateral-acceleration loads. This "sacred" position is found at the bottom of a carefully baffled sump tank in a dry-sump oiling system.

A dry-sump system consists of at least *two separate pumps: one draws an oil-and-air mix from a shallow engine oil pan and deposits it into the top of an approximately 12-inch deep tank; the other—supplying pressurized oil to the engine—draws oil from the bottom of the sump tank and delivers it to the main oil galleries in the block.*

Originally designed for aircraft, to ensure proper oiling even while flying upside down, dry-sump oiling has added reliability (and thus lowered the overall operational costs) in all-out competition cars, i.e., CAN-AM, Nascar, etc. But dry-sump oiling is not the best choice for all racing applications. It is expensive, complex (using a considerable amount of flexible hose, fittings, clamps, drive belts, brackets, and more), and when installed by less than an expert, prone to develop oil leaks. But despite all the potential drawbacks, even some production cars, such as a few models of Porsche, are equipped with dry-sump oiling systems.

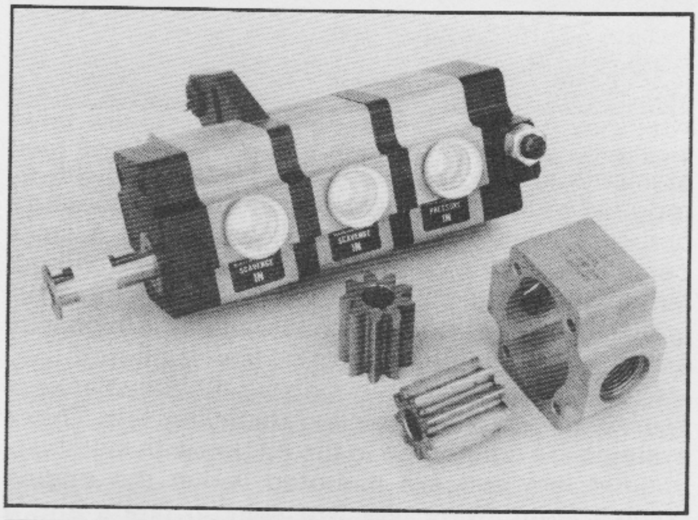
Because dry-sump systems virtually pump the oil pan dry, there is less oil to impinge on the reciprocating assembly. This reduces the frictional losses that invariably accompany a wet-sump system. But the horsepower gains are not substantial—particularly at lower engine speeds; and when compared to the high cost of a dry-sump system, the gains fall into a very high cost-per-horsepower category.

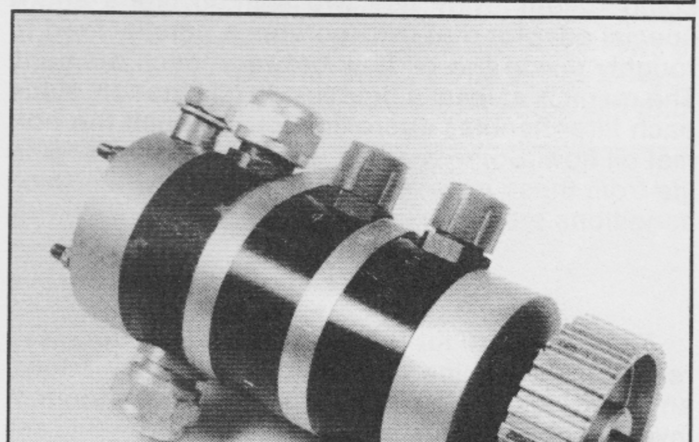
Dry-sump oiling is *never necessary* on the street, only rarely used in autocrossing, and not always used in all-out competition cars. But when chassis design or ground-clearance limitations preclude the use of an oil pan that will adequately keep air-free oil around the pump pickup, a dry-sump system is the answer, albeit an expensive answer. If you choose to install a system, carefully review your needs, shop around, and follow the manufacturer's advice. (Refer to the appendix for a list of sources.)

ADDITIONAL OILING CONSIDERATIONS

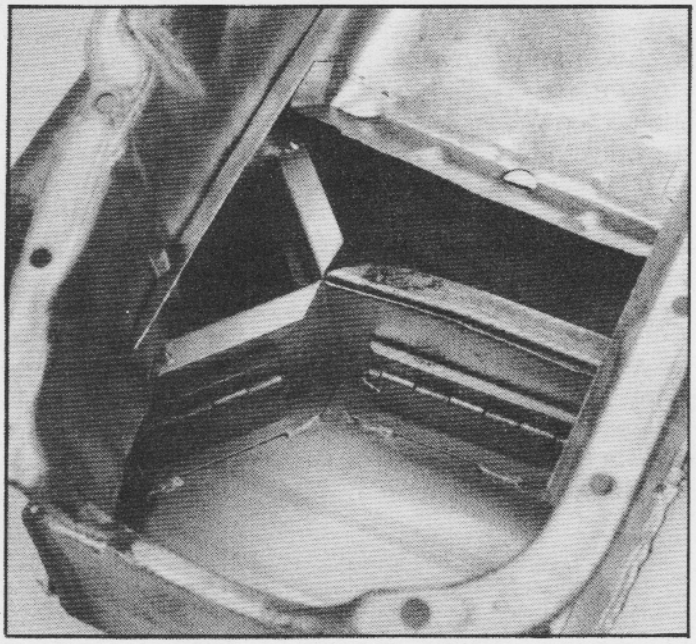
DUAL FILTERS

All modern Chrysler engines use a full-flow oil filter. This means that all of the oil from the pump is delivered to the filter before it is directed to engine internals. But *delivered* to the filter does not mean that all the oil is *cleaned* by the element. Spin-on filter

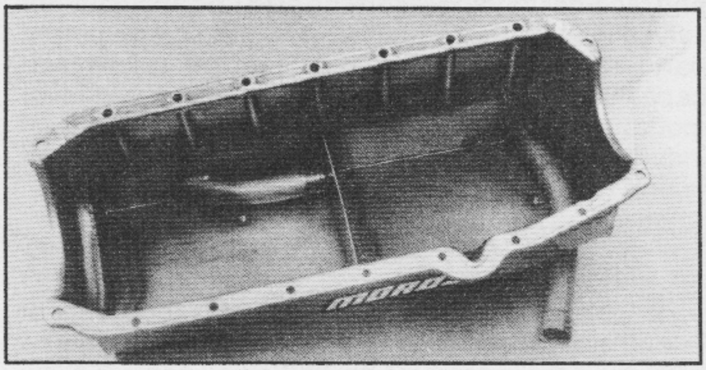




Both of these dry-sump oil pumps are three-stage units with two scavenge sections and one pressure section. They are lightweight and modular in design, permitting the easy addition/replacement of scavenge or pressure sections, if required. Dry-sump oiling is considered by many an absolute necessary for all-out competition, but thankfully, this exotica is never required on street machines.



The wet-sump oil pan (left) has "trap-door" baffles built into the sump. These doors are effective one-way passages that prevent oil from sloshing away from the pickup during hard cornering. The much simpler dry-sump pan (below) requires less baffling since the scavenge pump(s) can draw in both air and oil. The aerated oil is deposited in the top of a sump tank; the critical inlet side of the pressure pump then draws air-free oil from the bottom of the sump tank.



elements are too restrictive to clean all of the oil required by the engine. So filter cannisters contain a pressure bypass valve that allows all but a small percentage of the oil to return to the engine without filtration. As the oil volume through the filter decreases, the bypass valve closes; less oil is bypassed and a greater percentage of oil flow is filtered.

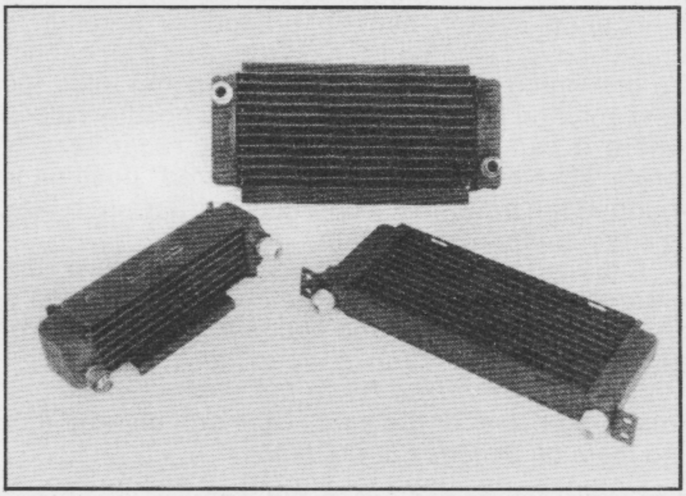
Some enthusiasts, recognizing the drawbacks of spin-on filters, have installed dual-filter kits. These systems are connected to the engine with stainless-braided line and are mounted within the engine compartment. They use two spin-on filters and a special adaptor that incorporates a parallel feed to roughly divide the oil flow between each element. The result is at least a doubling in oil filtration, since each filter handles approximately one-half the normal oil flow. Some road racers claim longer engine life from these installations, particularly when track conditions are dusty.

OIL COOLING

There is no doubt that overheated oil offers less resistance to engine wear. In road-racing situations, an oil cooler can be a worthwhile, if not required, investment. However, for street use, the benefits are not as clear cut.

An oil cooler will reduce oil temperature. But there is little evidence that this reduction in temperature offers any reduction in engine wear, or even an extension in oil life (when the temperatures under question are those normally found in street engines, i.e., below 240 degrees F). Frequent oil changes—every 2000 or 3000 miles—seem to be a more practical alternative, and they eliminate potential oil leaks than can occur from external lines and fittings.

However, an oil cooler can be an excellent addition to a supercharged—particularly turbocharged—engine. Higher combustion chamber temperatures (and turbocharger shaft-bearing lubrication and

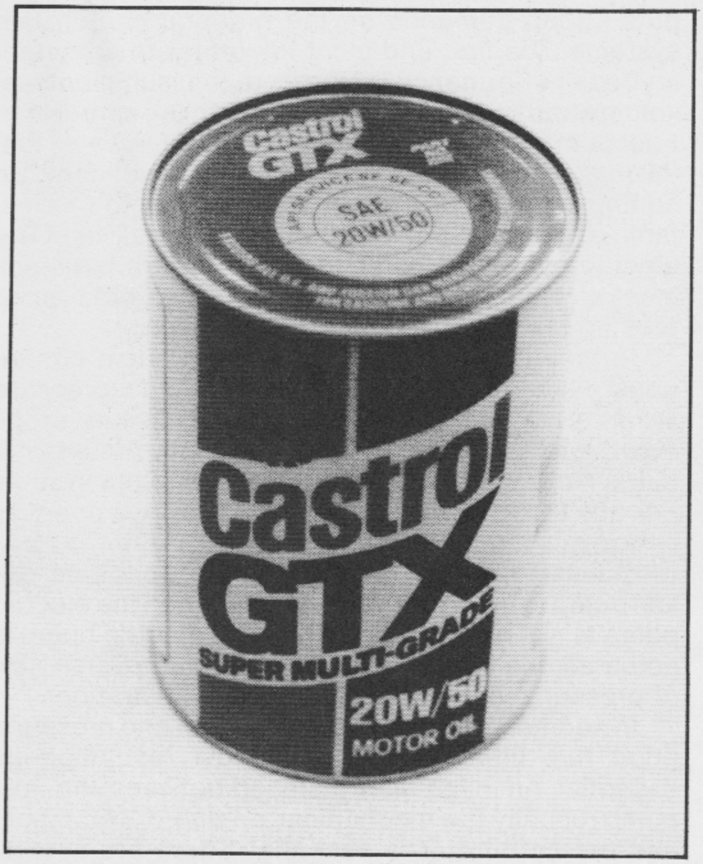


There are many sizes and styles of oil coolers. Be sure to select a oil cooler that is designed for your application. A cooler that is too small can restrict oil flow and an unit that is too big may just add unneeded weight. Make sure that any cooler you choose will withstand up to 200 psi (a high rpm blast with a cold engine can sometimes develop these pressure levels or higher).

cooling requirements) can substantially increase oil temperatures. In these cases an oil cooler can extend oil life, reduce oil deposits, and increase engine life.

OIL TYPES

Several years ago, synthetic oils were introduced to the public. Shortly after their introduction, Chrysler announced that these oils showed no benefits over standard motor oil; and considering their cost and unproven nature, they recommended that only MS-DG, organic-based engine oil be used in street and racing applications. Since then, little has changed regarding synthetic oil use. A few notable racers have found some brands of synthetic oil to



Modern multi-viscosity oil is widely used both on and off the race track. Synthetic oil has many proponents but just as many opponents; some claim that the quality of synthetic oils varies too much from lot to lot to be dependable. However, no matter which brand or type you select, make sure it is rated "API," "SF," or "SE."

have superior resistance to fatigue and outstanding resistance to engine wear. However, the formulas of these oils have changed so frequently that they no longer perform as originally tested. So the net result is mixed.

Modern multi-viscosity racing motor oil of 20W40 or 20W50 rating is a proven product. Synthetic oils are still steeped in controversy. Whichever you choose, be sure the rating is "API," "SF," or "SE"; beyond that, the choice is yours.

KEEP YOUR COOL

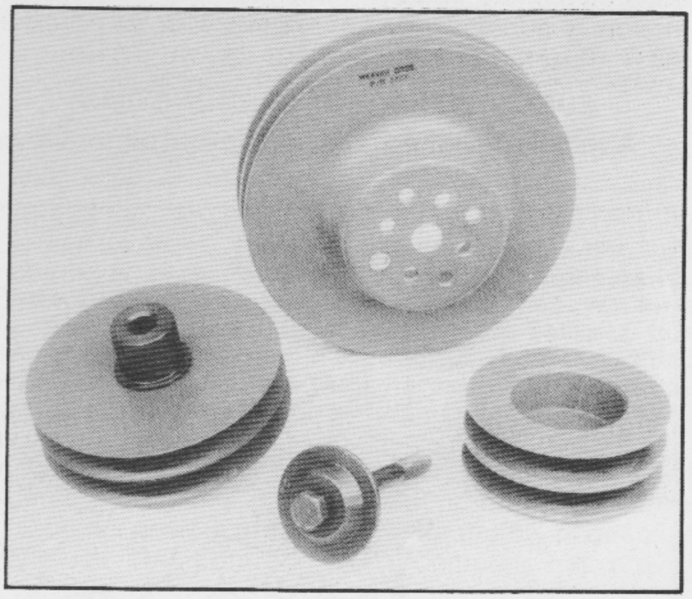
A high-performance engine produces more horsepower.

Now, that may not seem like earth-shattering information, but generating more power means generating more heat. And if the cooling system is not up to dissipating this added heat, coolant temperature can rise, causing pre-ignition, horsepower loss, and reduced reliability.

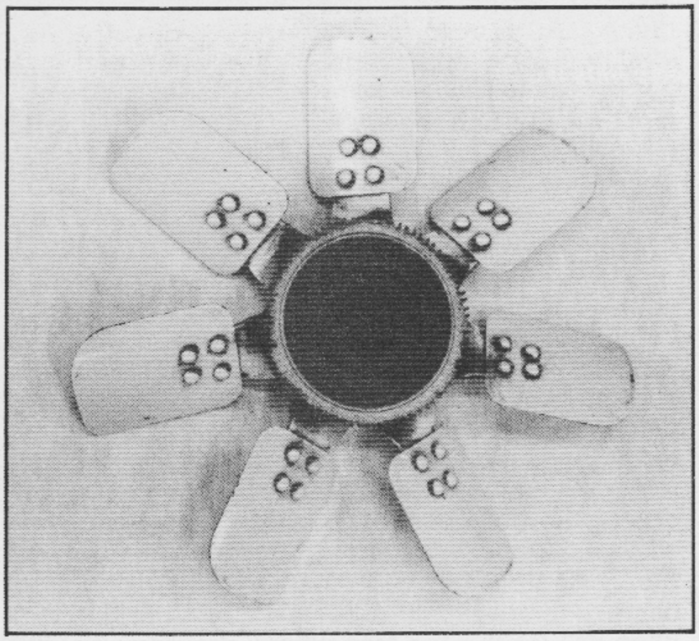
Overheating can be avoided by ensuring three things: 1) that the radiator has sufficient surface area, 2) that the fan and water pump are turning fast enough to move air and water at the proper volume, particularly at idle, and 3) that the thermostat is working properly and of the correct temperature rating (180°F or lower). If the engine overheats at idle, it is often due to insufficient water and/or air flow. A smaller water-pump pulley, a larger fan with more blades, and the addition of a fan shroud will all help. However, if the engine tends to overheat at high speed, the problem is often due to insufficient radiator surface area. Installing a larger radiator, or a larger core will usually help.

While increasing pump speed and fan size can improve idle-speed cooling, these modifications consume horsepower, especially at high rpm. This loss can be minimized by using a fluid-coupled (viscous) fan. These little miracles of technology are designed to "couple up" when the air coming through the radiator approaches an upper-temperature limit (usually 190°). When coupled, the fluid drive rotates a large fan (often seven-blade) with only slight slippage—producing lots of airflow through the radiator. However, when the incoming air is cooler than the lock-up temperature, the fluid coupling will let the fan free-wheel, drawing very little load from the engine. A fluid fan will work quite well with the small water-pump pulleys found on air-conditioned cars. These pulleys (and AC water pumps—designed to pump efficiently at higher shaft speeds) will ensure sufficient water and airflow to cool virtually any engine at idle.

But if you wish to minimize horsepower loss while still providing good cooling, a large 3- or 4-row custom radiator is the best answer. Some radiator shops can install large cores into stock upper and lower tanks. If the radiator shop also supplies the tanks, make sure that the outlet and inlet are positioned in the right place. Insist on a 3-row (preferably 4-row) core; this means that there are *3 or 4 rows of tubes* for water to flow through. The larger surface area will, to some extent, make up for slow fan-and-



Crank, water pump, and alternator pulleys come in various sizes to simplify custom tailoring the cooling system. While lowering water pump and alternator speeds will consume less horsepower, make sure that the engine does not overheat at idle or high speed.



Fluid-coupled fans are designed to "slip" and minimize horsepower loss below a certain pre-set temperature. When air moving through the radiator reaches about 190°, the fluid drive "hooks up" and produces plenty of airflow, especially if the fan is a large 7-bladed unit (like the one shown above).

water-pump speed. Although somewhat expensive—often costing up to $200—a custom radiator can be efficient insurance against overheating, when used with the smallest acceptable fan and the largest acceptable water-pump pulley.

Finally, never use plain water in the cooling system. Without anti-freeze or other anti-rust compounds, water will rapidly rust away the radiator, core plugs, and water pump. The corrosion will be amplified if there are any aluminum engine components. Always mix cooling-system water with corrosion-resistant additives, or anti-freeze with these additives, or at least soluable oil.

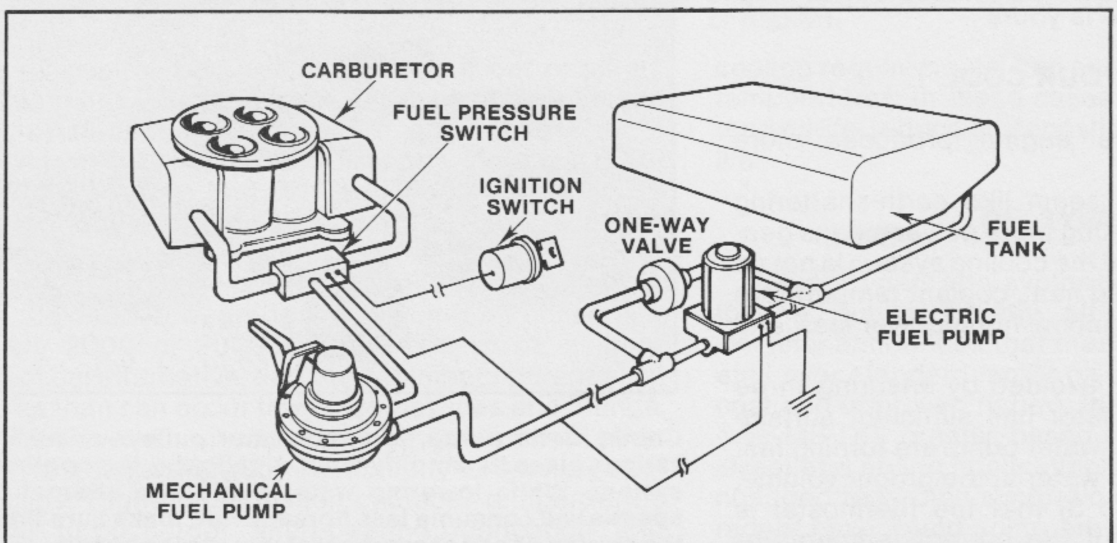
FUEL SYSTEM AND CARBURETION

A vehicle designed for handling and performance puts added demands on the fuel and carburetion systems. The first and most important requirement for peak performance is to ensure that sufficient fuel volume and pressure are delivered to the carburetor. Fuel pumps are much more effective at *pushing* fuel through a hose, rather than *drawing* it from a tank. So adding an auxilliary electric pump close to the fuel tank is an excellent way to increase delivery. The electric pump can be used in series with the stock mechanical pump—the outlet of the electric pump feeding the inlet of the mechanical pump.

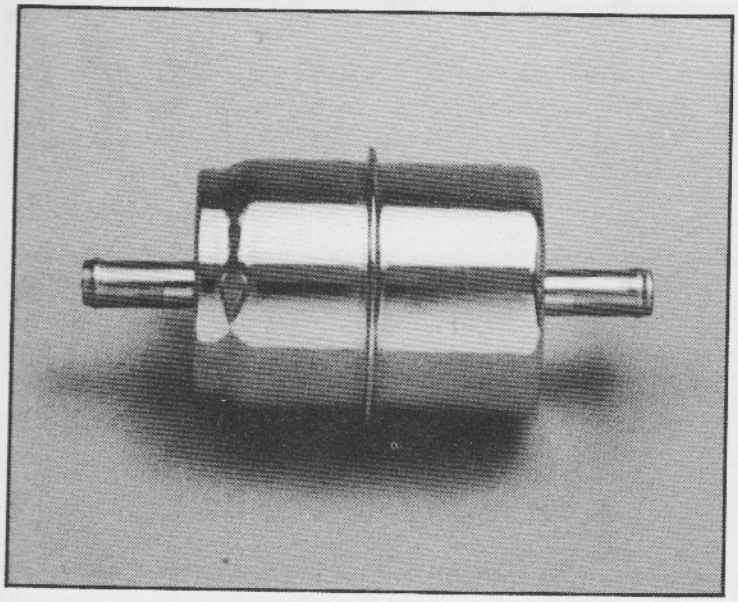
Rather than running the electric pump continuously, some enthusiasts have installed a pressure switch at the carburetor; when fuel pressure drops below a preset level (usually 3 pounds), the electric pump kicks on and adds needed flow to the system. For this setup to work properly, however, a one-way check valve must be installed parallel with the electric pump. The check valve allows the stock mechanical pump to draw fuel from the tank when the electric pump is not running; and when the electric pump is actuated, the check valve prevents the reverse flow of pressurized fuel to the tank (see illustration).

In addition to adequate fuel volume and pressure, good fuel filtration—with minimum restriction—is essential. An inline filter installed between the final pump (usually the mechanical, engine-driven pump) and the carburetor is essential. There are several excellent models available. Three popular models are the Holley, the Mr. Gasket in-line with a paper element, and the Carter in-line with porous-earthen elements.

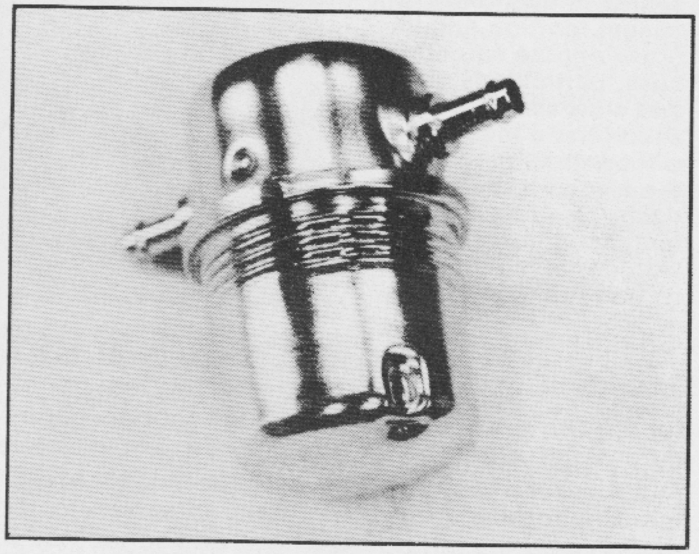
Never install a filter between the pump and the fuel tank. Remember that pumps are much less effective at drawing fuel than pushing it; so adding restrictions to the pump inlet can substantially reduce flow. The small screen filter, normally located on the end of the pickup in the fuel tank, is often adequate to keep most of the "big chunks" out of the pumps and lines. Finally, for all applications except all-out racing, the sintered-bronze inlet filters found in most Holley carburetors should be retained. If used with an auxilliary filter, they rarely become



The addition of an electric fuel pump close to the fuel tank often provides all the fuel volume required to produce 600+ horsepower. However, on street machines the pump need not run all the time. With the addition of a pressure switch at the carburetor and a one-way valve at the electric pump, this fully automatic system can be fabricated (see text).



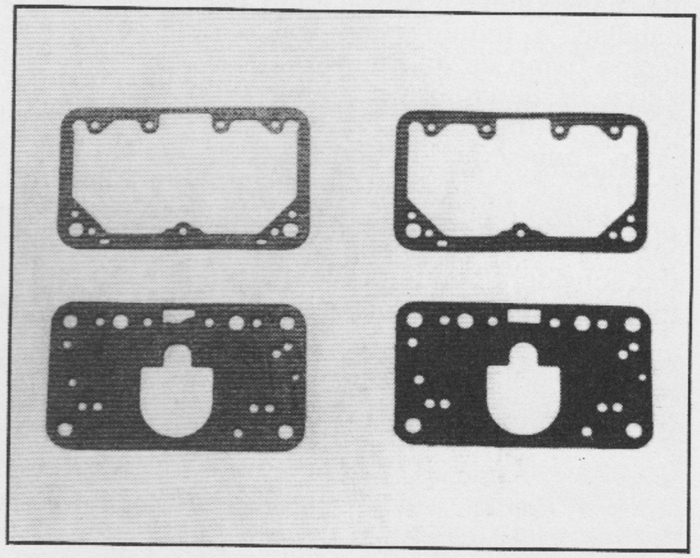
A simple fuel filter such as this Mr. Gasket (left) or Holley (below) are inexpensive insurance for your carburetor and your engine. But remember, never install a filter between the fuel pump and the fuel tank; pumps are much less effective drawing fuel than they are pushing it and a filter on the pump inlet will substantially reduce flow.



clogged and restrictive; and they add a final stage of filtration to the system.

Some carburetor models do not perform well during hard cornering. Models that use side-pivot floats or that have reduced fuel-bowl volume (with EPA regulated bowl inserts) or that lack sufficient fuel baffling can cause fuel starvation, poor performance, and stalling. Most racers agree that the best carburetor for "road work" is a Holley with center-pivot float bowls. This carburetor requires little modification and when combined with common sense tuning procedures, provides excellent, reliable performance.

Some things to remember: 1) Keep everything clean—dirt can plug air bleeds and small fuel-flow orifices. 2) The carburetor will require only minor modifications, if any. (Major jetting changes, removing power valves, or installing mechanically-operated secondaries will almost always cause more problems than they solve.) 3) Maintain proper fuel-bowl (float) level. 4) Use a good air cleaner. 5) Ensure that the throttle linkage allows all four barrels to fully open. 6) Use the latest Holley gaskets that do not shrink when dry, and/or always replace all distorted or damaged gaskets.



Holley has recently introduced gaskets that do not shrink when dry. Have you left your car standing for a couple of weeks, only to discover that it won't start or that it runs very poorly when started? If so, you have probably been victimized by shrunken gaskets. It is amazing to disassemble a Holley carburetor and find the gaskets only half their normal size! When you rebuild, insist on the new shrink-resistant gaskets.

SOME FINAL POINTERS

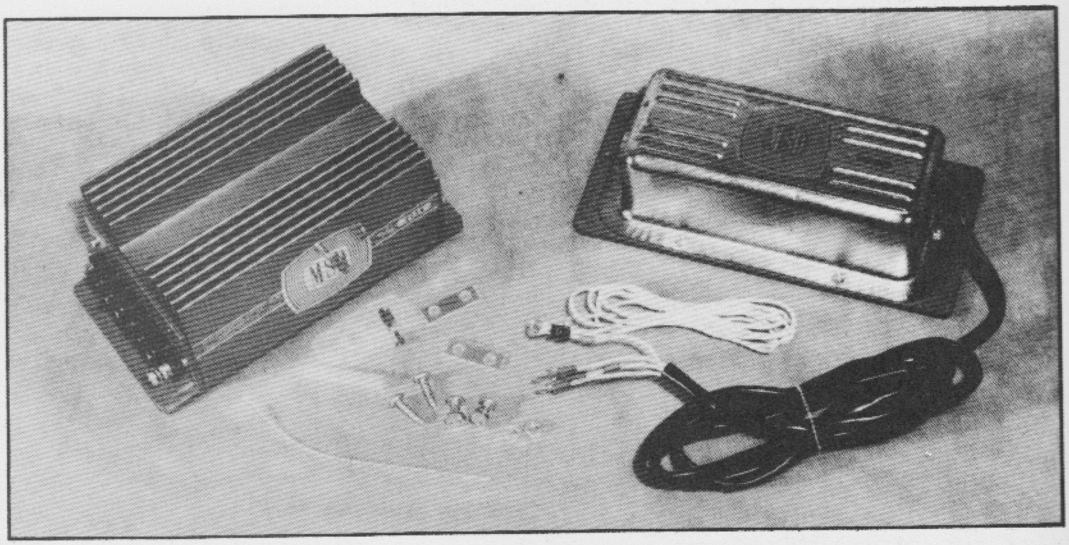
IGNITION

Choosing an ignition system for road racing and street applications is very straightforward: use a Chrysler breakerless distributor, a Chrysler or Autotronic Controls electronic box, and a good high-performance coil and ignition wire—all these goodies are available from Chrysler Performance Parts. Nothing "trick" is required. In fact, the simpler the better.

The only exception to this rule might be the Autotronics Controls multi-strike control box. This sophisticated system not only supplies a multiple spark for sure ignition, it can be purchased with a "ping control" that will retard or advance the spark timing from the dash (a neat addition for street vehicles used in areas where the quality of gasoline varies). The control box is complex but proven reliable.

ENGINE MOUNTING

Keeping the engine mounted on rubber will extend the life of the chassis and the sanity of the

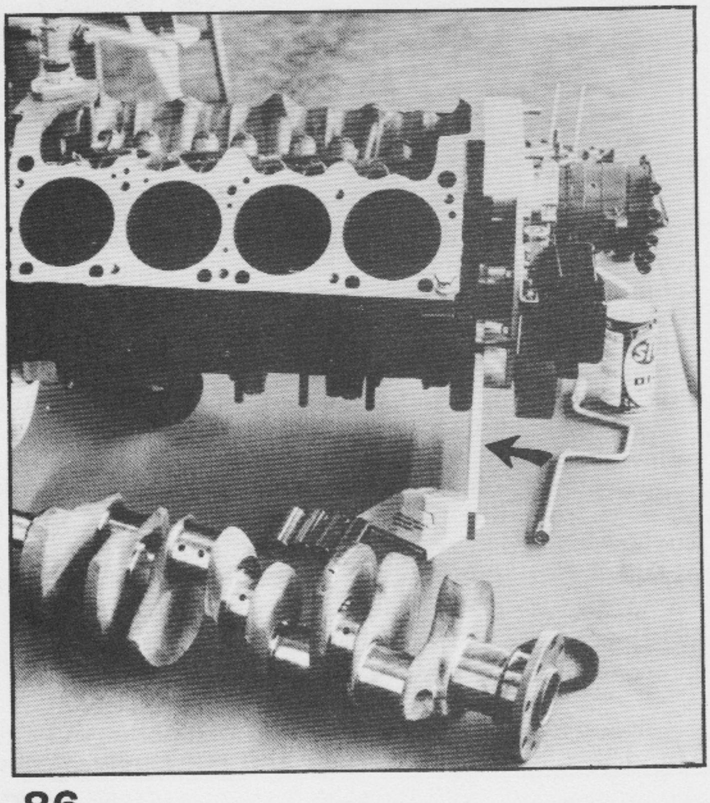


The Autotronics Controls multi-spark ignition system sends multiple sparks to each plug to ensure ignition and improve engine smoothness, particularly at idle and slow speed. They even offer a model that will control "ping" with the turn of a knob on the dash. This all adds up to increased horsepower with improved driveability and fuel economy.

driver. Solid engine mounts transfer a great deal of vibration to the chassis and provide no real benefit for street applications. However, rubber engine mounts can break; so the addition of a simple torque strap is well worth the time and the modest investment. A torque strap is normally attached to the engine near the left cylinder head and connected to the chassis along the front frame rail. When properly installed, a torque strap will restrain excessive engine "wrap up" in the event of an engine-mount failure or during wide-open-throttle acceleration. Torque restraints can be made of chain or steel cable. They should not be installed tightly but just loose enough to restrain only excessive engine twist.

For Pro-Solo I and II, road racing, or all-out competition, solid engine mounting is now widely used and considered state of the art. But this method of solid mounting does not simply involve replacing the stock mounts with steel counterparts; since steel-biscuit mounts—sometimes used in drag cars—can cause block side-wall failure when used in endurance racing. Block failure (from simple cracking to complete sidewall separation) is due to severe high-rpm vibration transmitted directly into thin-cast water jackets. This vibration must be absorbed by a "more springy" design and transmitted to stronger block sections. And this is accomplished with modern solid-mounts, commonly called "engine plates."

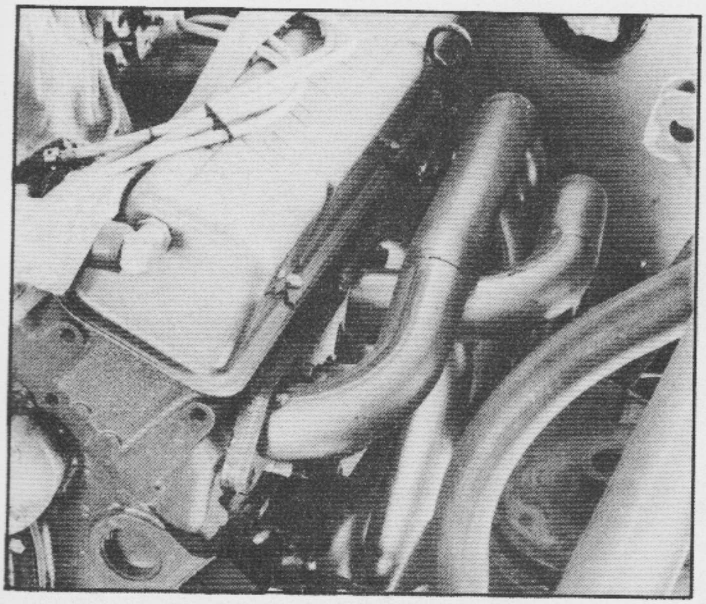
Engine plates are usually installed at both the front and rear of the engine block. At the front, the mounting plate is sandwiched between the timing-chain cover and the front surface of the block. The front cover is usually machined to allow for the plate thickness. Rear mounting is similarly accomplished between the engine block and the transmission bellhousing, with either the bellhousing or the rear-block surface machined to retain the correct clutch/input-shaft dimensions. This may sound complex and tricky, and it is; but fabricating headers, clutch linkage, etc., to "wrap around" a rear mounting plate can even be more difficult. Most of the basic parts needed for solid-mounting installations are available from the Kit-Car Direct-Connection program.



This smallblock Mopar was developed for the Indy program. Note the exotic timing cover with direct-drive distributor, Mallory coil, and front engine-mounting plate (arrow). Since engine plates absorb much more vibration than solid replacement mounts, they restrain engine movement and reduce stress on cylinder-block wall sections, preventing cracks that can occur at high rpm.

HYDRAULIC CLUTCHES

Clutch linkage is rarely easy to fabricate and install, especially in a road racer. The headers take up most of the available space; and even if there is just enough room for the torque-shaft, mounting brackets, etc., the clutch may not operate smoothly when the engine is under load. Finally, clutch linkage can be an annoying complication during routine engine service that requires torque-shaft removal



Fabricating special headers for unique applications—i.e., to clear engine-mounting plates—is a complex job not recommened for the "first-timer." Optimum header design requires tubing to clear accessories, while maintaining uniform lengths within 1/2 to 1 inch and minimizing the number of sharp-radius bends. Oh yes, one more thing: the entire assembly should be easily removable. No sweat, right?

and installation.

Instead of a mechanical linkage, hydraulic-clutch activation is an alternative that can be used on both the street and track. The hydraulics function very similarly to the braking system; a master cylinder (utilizing a single piston) is connected to the clutch pedal and a small slave cylinder is mounted onto the bellhousing; when the clutch is depressed, fluid pressure activates the slave cylinder, moving the clutch fork and releasing the clutch.

Installing a system on your Mopar will require some fabrication, since most pieces (available from Airheart, Ansen and many speed shops) are designed for Chevrolet applications. And since some fabrication will be required anyway, you may save money by modifying a hydraulic clutch from an import car, such as the Datsun "Z." But whichever system you use, a properly installed and thoroughly bled hydraulic clutch can minimize linkage/firewall flexing and simplify engine and header removal.

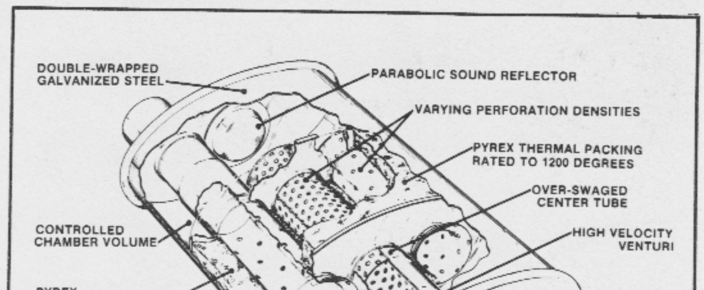
HEADERS AND EXHAUST SYSTEM

The number one road-racing header problem for Mopars is *finding any*. A-bodies can use a fenderwell header made by Cyclone; it provides adequate front tire clearance. But B-body owners and any Mopar owner desiring chassis-exit headers must look into custom designs. Some enthusiasts buy chassis headers designed for drag racing and modify them to suit their requirements. This often includes adding a flat collector, where all four tubes enter side-by-side rather than in a round pattern. Flat collectors can increase ground clearance by 1 or 2 inches, while having almost no effect on horsepower.

There are some situations, however, where modifying existing headers is more difficult than just starting from scratch. And luckily, there are many shops around the country that specialize in custom-header fabrication. In fact, some header manufacturers will even build custom systems. But expect to pay a premium price; $300 to $400 is not unusual. Custom road-racing headers should be designed to provide a wide power band and use a tube diameter of 1-5/8- to 1-7/8-inches for street and autocrossing. All-out competition systems should use 2-inch primary tubes. (You may contact Martin Automotive Design for more information and to purchase custom Mopar headers.)

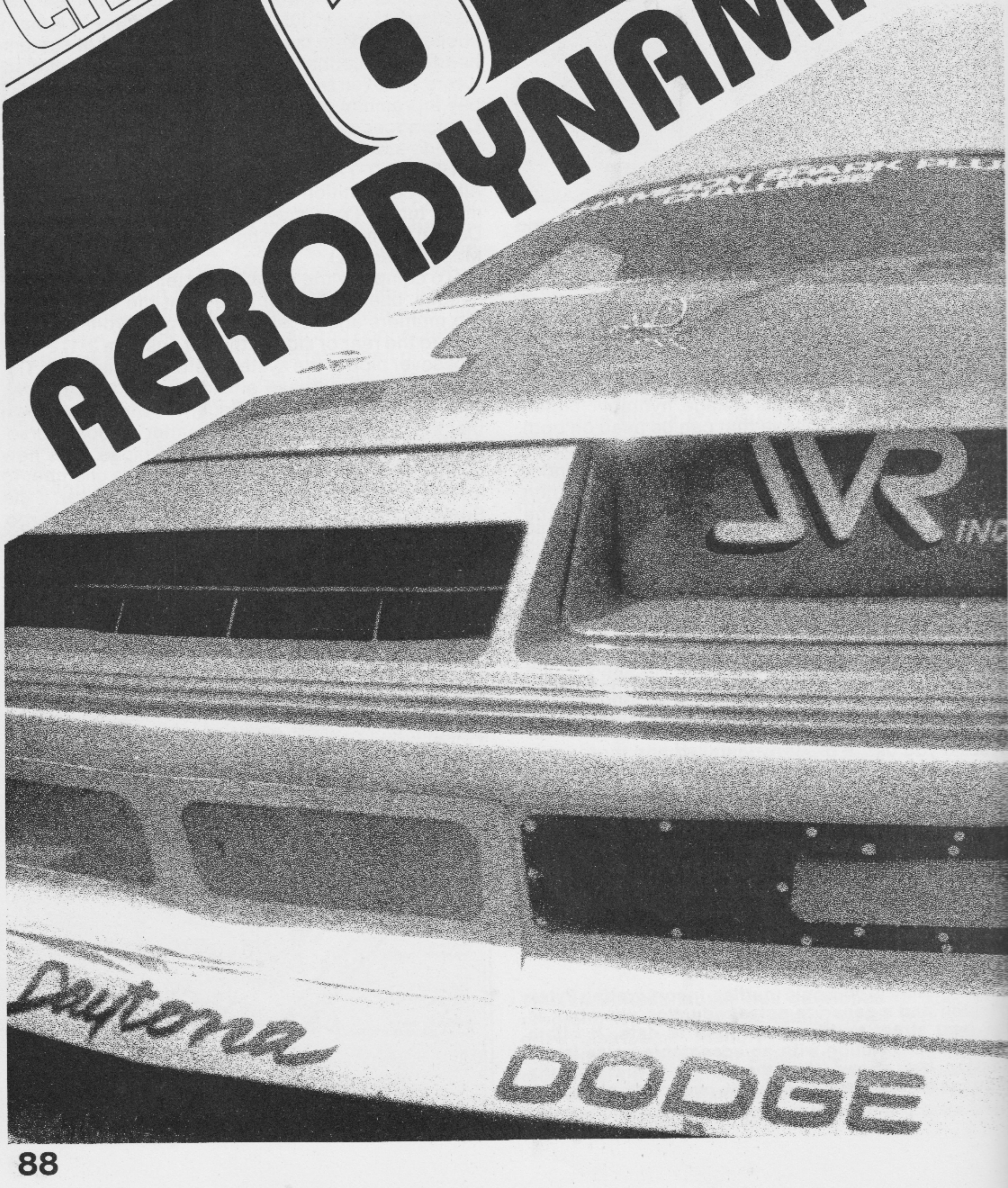
To complete the exhaust system for street applications, use 2-1/2-inch pipe from the collectors to the mufflers and at least 2-inch pipe from the mufflers to the rear or side of the car. Install a 2-1/2-inch crossover pipe in the system before the mufflers. The crossover will increase efficiency and horsepower, while lowering the noise. Recent testing has shown that the Cyclone Sonic Turbo muffler reduces horsepower by less than 5% over open pipes. This new design uses Pyrex packing in a separate chamber to lower the sound level without affecting power. If the Sonic Turbo cannot be located, the Arvin Turbo, Cyclone California Turbo, Maremont Turbo, and Supreme Super C are within about 2% of the Sonic Turbo dyno readings.

Many road course and autocross tracks are restricted by local noise ordinances and require header mufflers. The SuperTrap muffler is widely considered the best choice for these applications. SuperTrap is a state-of-the-art design that is commonly used in motorcycle racing. (In fact, some bikes produce more power with these mufflers than with open pipes.)



The Cyclone Turbo-Sonic muffler, incorporating Pyrex packing and a separate noise-reduction chamber, is the best street muffler developed to date. It offers minimum restriction with an acceptable noise level. Testing has shown that the Turbo Sonic typically reduces horsepower by only 5% over open pipes.

CHAPTER 6
AERODYNAMICS



Richard Petty's Dodge Charger was one of the more aerodynamic cars ever to road race. Despite the smooth contours, the "airfoil" shape that tends to produce undesirable lift at high speed is still very evident. Note the front and rear spoilers added to resist this upward lift.

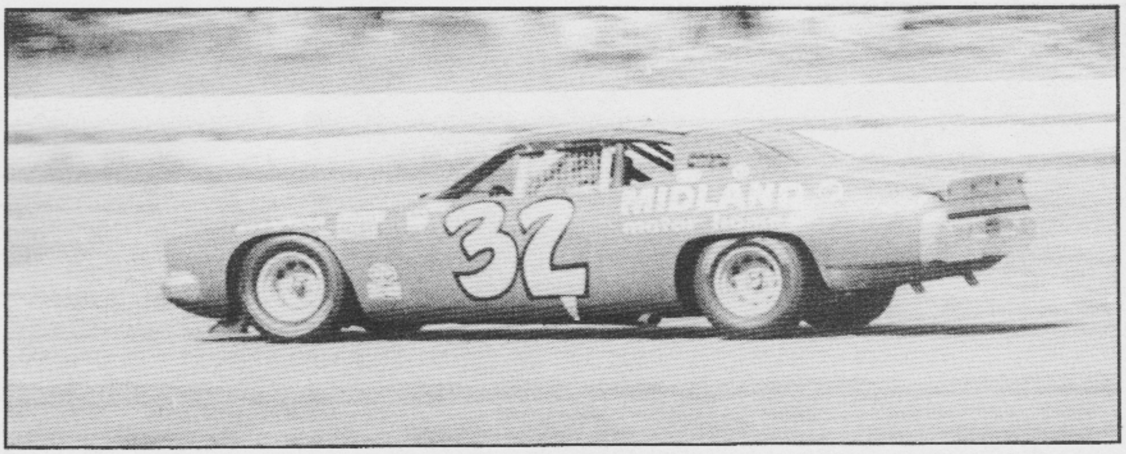
VEHICLE AERODYNAMICS & AERODYNAMICS AIDS

Since the first spoiler was bolted on a race car in the early 1960s, the field of vehicle aerodynamics has become increasingly important to not only competition vehicles but also everyday street cars. Besides improving high-speed stability, aerodynamic devices can increase fuel economy by reducing the resistance (drag) caused by moving through an "ocean" of air. This chapter provides a brief introduction to aerodynamics as well as a review of some of the devices currently used to reduce drag.

The basic shape of an automobile is similar to the wing of an airplane. This likeness is seen by examining the vehicle profile. Basically, the distance over the top of the car is greater than the distance under the car. At speeds below 45 mph, this shape has little effect on handling or aerodynamics; however, at about 50 mph things begin to happen. The air rushing over the top of the car develops a low-pressure area, while the air moving under the chassis becomes turbulent and compressed. The combined effect of these phenomena is "lift"; i.e., a force tending to raise the car off the ground. This very undesirable effect induces handling instability, usually resulting from a lack of steering response (due to front-end weight reduction). In addition, the turbulence produces drag (it takes power to make air turbulent) which consumes horsepower and fuel.

If not for the public concern over rising fuel prices, Detroit would still be manufacturing aerodynamically inefficient vehicles. However, Detroit builds what the public wants (eventually!), and these demands have forced engineers to re-examine exterior body shape and incorporate many elements developed and proven on race cars. But these developments are not just restricted to overall body design. Many of the new "innovations" can be easily incorporated on earlier cars, reducing drag and providing a complementary improvement in high-speed handling and fuel economy.

The new breed of domestic cars are designed with an obvious concern for aerodynamics and fuel economy. This 1983 Dodge Charger sports an integral front air dam and an optional rear wing. These well-engineered devices not only make new cars look sporty and aggressive, but also provide measurable benefits on both the street and track.



A B-body sedan set up for road racing—even on tight courses like Laguna Seca Raceway in Monterey, California—can run competitive times. To be competitive, however, all Mopars must employ the latest in aerodynamic devices, such as front and rear airfoils and a small forward rake angle.

BREAKING THE WIND BARRIER

One of the best ways to reduce aerodynamic drag is to "rake" the front of the vehicle lower than the rear. The optimum angle that the car presents to the wind has been found to be about 1.5 degrees. For most vehicle shapes, aerodynamic drag will increase when the angle is either increased or decreased from this optimum value. A 1.5-degree rake angle can reduce drag as much as 4% at all speeds. But, unfortunately, the wing-lift effect is still quite pronounced at nose-down rake angles; so most vehicles will experience front-end lift at high speeds, re-inducing drag and poor handling.

As mentioned earlier, front-end lift at high speed is caused by air pressure under the car exceeding air pressure over the car. Racers have developed several methods to counteract these unwanted "air forces." One of the first—and reasonably effective—countermeasures was adding a front spoiler. This small air dam mounted under the front bumper redirects some air that would normally pack itself under the chassis. The net effect is less front-end lift not only from reduced high-pressure air under the chassis, but also from a downward force directly on the spoiler. Although front spoilers induce some drag, the overall effect is usually reduced drag since the larger drag forces of the chassis have been decreased. And a very welcome side benefit is improved stability and handling, which means faster lap times (and happier—and safer—drivers).

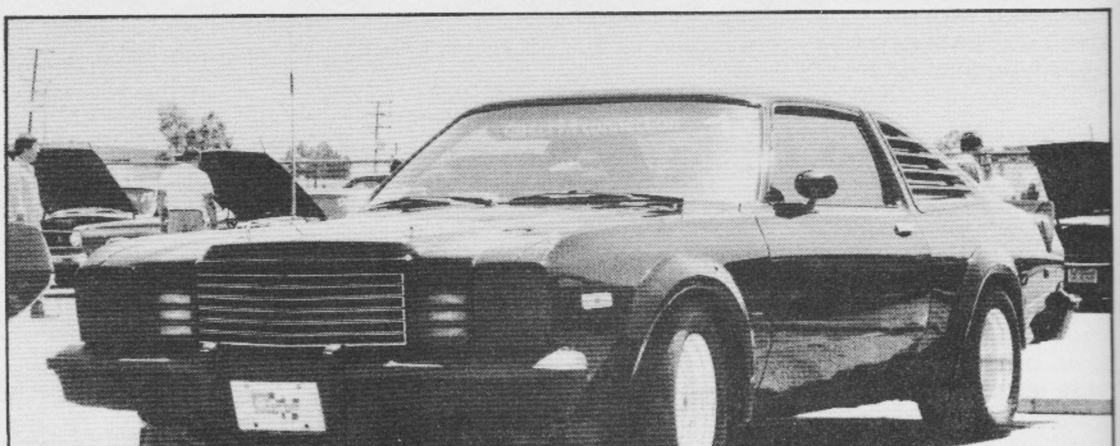
Further aerodynamic testing revealed that the air rushing over the abrupt drop at the rear end of the car (the trunk lid) created turbulence and increased drag. As racers are likely to believe, "If one works, try two," the next logical step was adding a rear spoiler. The rear spoiler substantially reduced aft turbulence and added some downforce to the rear end. As it turned out, this was just what was needed to balance the downforces generated by front spoilers and to maintain uniform loading and overall stability. Rear spoilers (and their "wing" descendants) are now considered to be essential chassis tuning tools (where permitted by the rules).

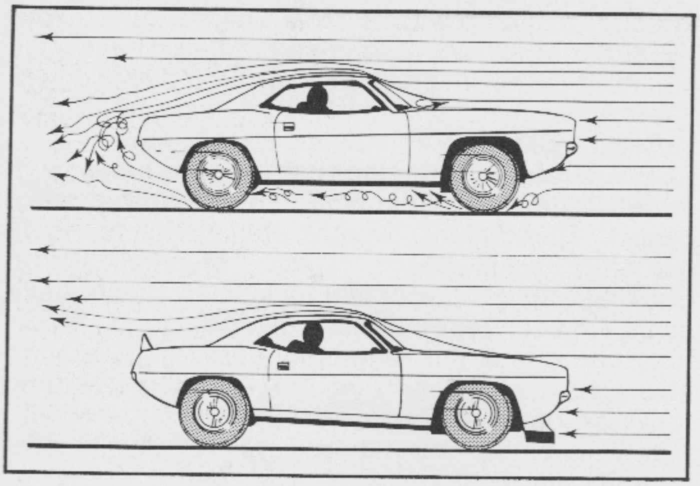
The rear spoiler reached its ultimate design with the "tail" of the Dodge Daytona and Plymouth Superbird. It gave Mopars such an advantage that these "clearly illegal designs" were banned from the raceways. Such are the ways of the sanctioning bodies; if it's really innovative and effective, ban it "for the good of the sport."

SPOILER FORCES MULTIPLY

At high speed, both front and rear spoilers add hundreds of pounds of downforce that "glue" the

At high speeds, air moving under the chassis can build up sufficient pressure to "lift" the front end. However, a small air dam mounted under the front bumper can both counteract front-end rise and improve engine cooling (lower pressure under the chassis improves engine-compartment outflow).





Without aerodyanmic aids, the car in the upper illustration is subjected to additional drag and front-end lift. Some air enters the radiator but most airflow passes both under and over the chassis. Over-the-chassis flow causes turbulence at the windshield cowl and at the rear end; while under-the-chassis flow restricts air outflow from the radiator and engine compartment. The additional pressure under the chassis also adds up-lift to the front end, reducing stability. With the addition of a front and rear spoiler and a 1.5° rake angle, most airflow is directed over the chassis and there is a substantial reduction in turbulence at the windshield cowl and rear end. The front spoiler increases front-end down force by reducing air pressure under the chassis, which also improves radiator outflow and cooling. The rear spoiler stabilizes airflow, reduces drag, and adds some rear down force to improve traction.

tires to the track. And this added traction is obtained without an increase in lateral forces that ballast would induce. But a major improvement in, what we must consider, the *use of air* was still yet to come. This advanced state of the art has produced designs that are so aerodynamically efficient that they utilize virtually no suspension travel due to the massive downforces generated just by moving through the airstream. What is this black-magic secret that uses air to such an advantage?

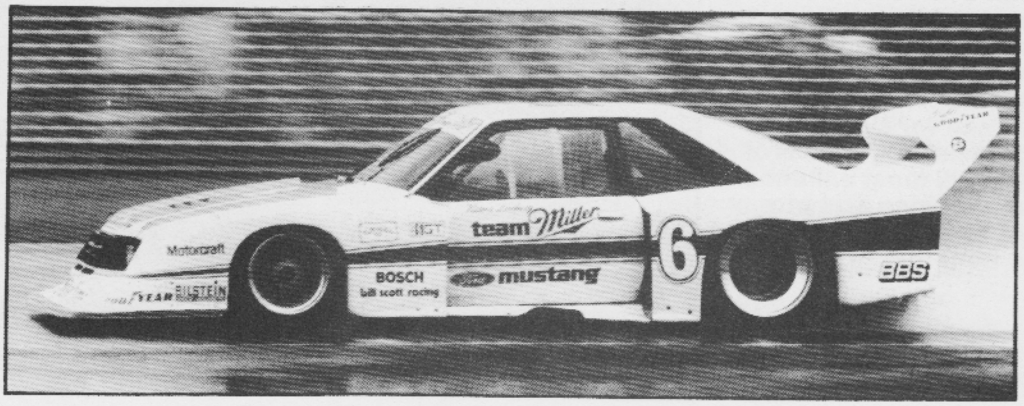
GROUND EFFECTS

Advances in front spoiler design generated larger and larger aerodynamic surfaces and eventually evolved into the *ground-effects air dam*. It looks like and acts similar to what its name implies—a flat aluminum or fiberglass panel attached to the front of the car that effectively *prevents almost all airflow under the chassis*. This seemingly inefficient "wall" creates a strong down-force pressure because it forces almost all airflow over the chassis, creating a pressure differential between the compressed *over-the-car* air and the virtually non-existent *under-the-chassis* flow. In fact, the air dam, when properly sealed to the pavement, begins to generate usable down forces at speeds as low as 35 mph.

The ground-effects design has virtually revolutionized aerodynamic applications, particularly for street cars. Since spoilers are, for the most part, ineffective below 55 mph and since optimum ground-effects designs work at street speeds, there has been a surge of interest in buying—and consequently in manufacturing—street air-dam bolt-ons. The better kits generally combine a sturdy aluminum or fiberglass panel with an abrasion-resistant rubber skirt that drops as close as 1/2-inch to the pavement. The overall design must be flexible to deflect upon impact from typical rock-sized obstacles (i.e., cans, curbs, parking blocks, etc.) while

The modified factory air dam on Joe Varde's Dodge Charger is typical of stock units modified for racing; the dam area has been increased by adding a lower panel. Blocking off (or partially restricting with fine-mesh screen) under-hood airflow is another common modification that reduces both air resistance and front-end lift. Only the air needed specifically for engine and brake cooling should enter the front-grille area.

This Miller-sponsored Mustang uses the latest in state-of-the-art aerodynamics. From the rakish front air dam to the rear wing, all surfaces fully utilize the airstream to increase stability.



incorporating a mounting sturdy enough to withstand the constant loads from a dragging rubber skirt.

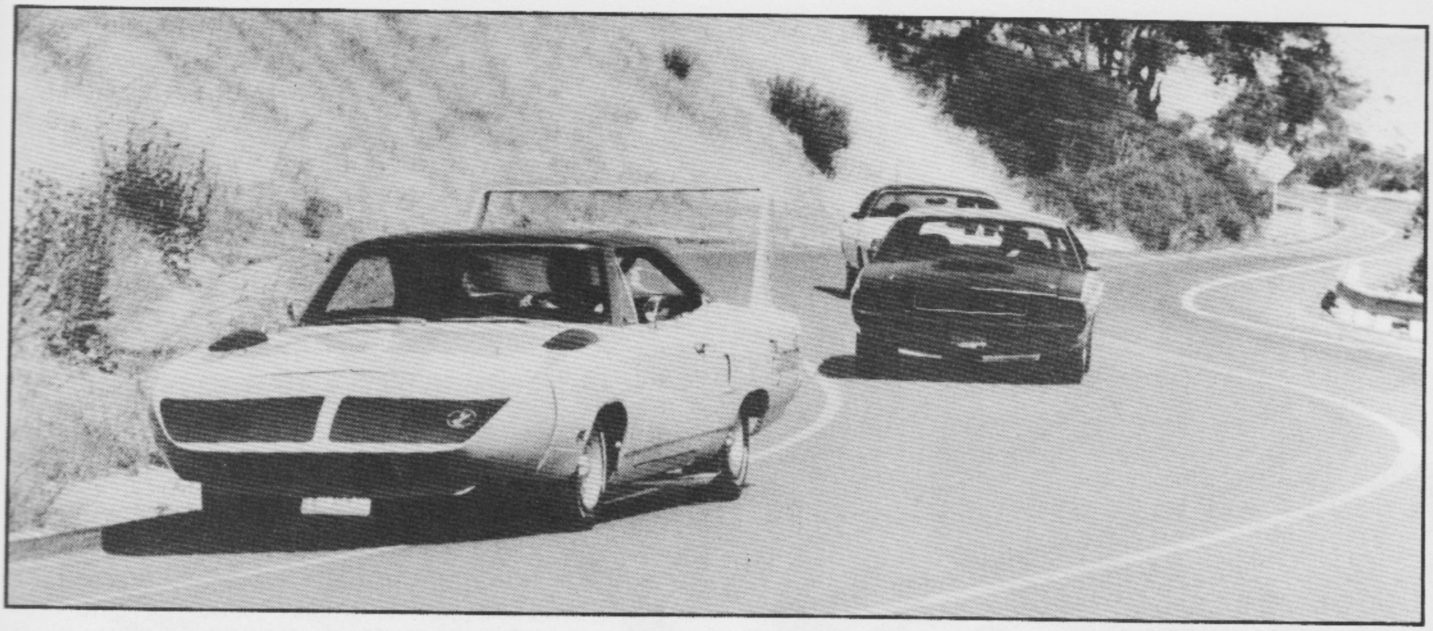
On the race track, however, the air dam initially produced some unwanted side effects. The front-end down-forces were so great that more rear down-force loading was needed to return stable handling to a now substantially unbalanced chassis. So even larger rear spoilers were developed and tested. But a point of diminishing return was found, beyond which the rear spoiler was inefficient due to severe aerodynamic drag. So a true "wing" was designed and attached to the car with pedestals high enough to put the surfaces in "clean," undisturbed air. The wing is mounted in the inverted position to induce down-force rather than lift, and the whole assembly looks very strange to the uninitiated.

But careful wing designs can produce huge down forces (hundreds of pounds) and still remain aerodynamically efficient. Porsche was the first to widely use this design in the classic "whaletail" Turbo Carreras. Now, most competitive road racers have some form of rear-mounted inverted wing. Unfortunately, unless the wing is mounted high enough above the car to be in "fresh" air (virtually precluding their use on most street cars), they don't work efficiently because aero drag negates any positive benefits.

SETTING UP A STREET CAR

For street use, where the last percent of improvement in handling is less important, aero-

Mopars have used many types of rear aerodynamic devices. The wedge-type and small wing used on E-body cars are "universal" devices that can be added to most cars, while the Superbird wing and the R/T spoiler are more specialized and designed for a particular chassis. Regardless of design, most enhance appearance and provide some aerodynamic benefits.

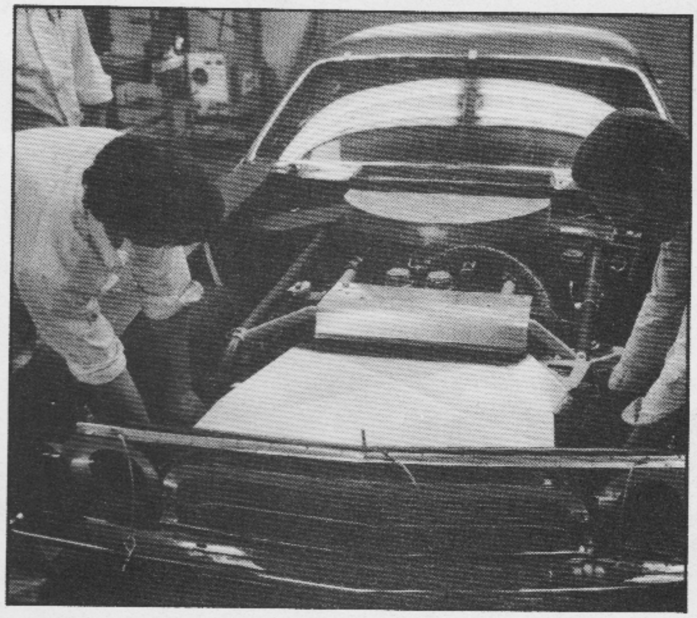


Mopars set up with good suspension, tires, brakes, and "sexy" aerodynamics are as much fun to drive on twisty back roads as they are under downtown lights.

dynamics plays a less vital role. However, choosing the right pieces can measurably improve handling and add great "show appeal" to any Mopar. A front-mounted air dam and a trunk-mounted rear spoiler are generally easy to install and not too expensive. The more ambitious may want to consider installing the high-mounted Superbird spoilers since they are still available through Chrysler parts; the visual effects will certainly be "far out." However for the more practical, the basic wedge-shaped rear spoiler is the best choice. It is an efficient design that will produce maximum downforce and minimum drag per dollar spent. The wedge should be no larger than 3-1/2-inches high, and the front surface should produce an angle of no more than 35 degrees from horizontal. There are several aftermarket manufacturers that produce rear spoilers that will attach to just about any car. Chrysler parts also offers spoilers for most late-model A-, B-, and E-body cars.

If you cannot locate a factory or aftermarket spoiler that suits your taste, you can fabricate your own from 0.032- to 0.035-inch aluminum sheet stock. This is not an easy job, and special cutting and fastening tools are required; but at least the cost is low. Make templates using heavy poster board before you begin to work with the aluminum. And if you decide to build a front spoiler or air dam, make sure that you allow for airflow to the radiator and brakes. Air dams are very effective and can cause severe brake overheating if adequate ducting is not provided.

To maintain uniform chassis loading, the front spoiler or air dam should be installed at the same time as a rear spoiler. But, if a choice has to be made, it is best to add the front device first. Most handling improvements, particularly for street cars, will be obtained from a front device, since a rear spoiler is



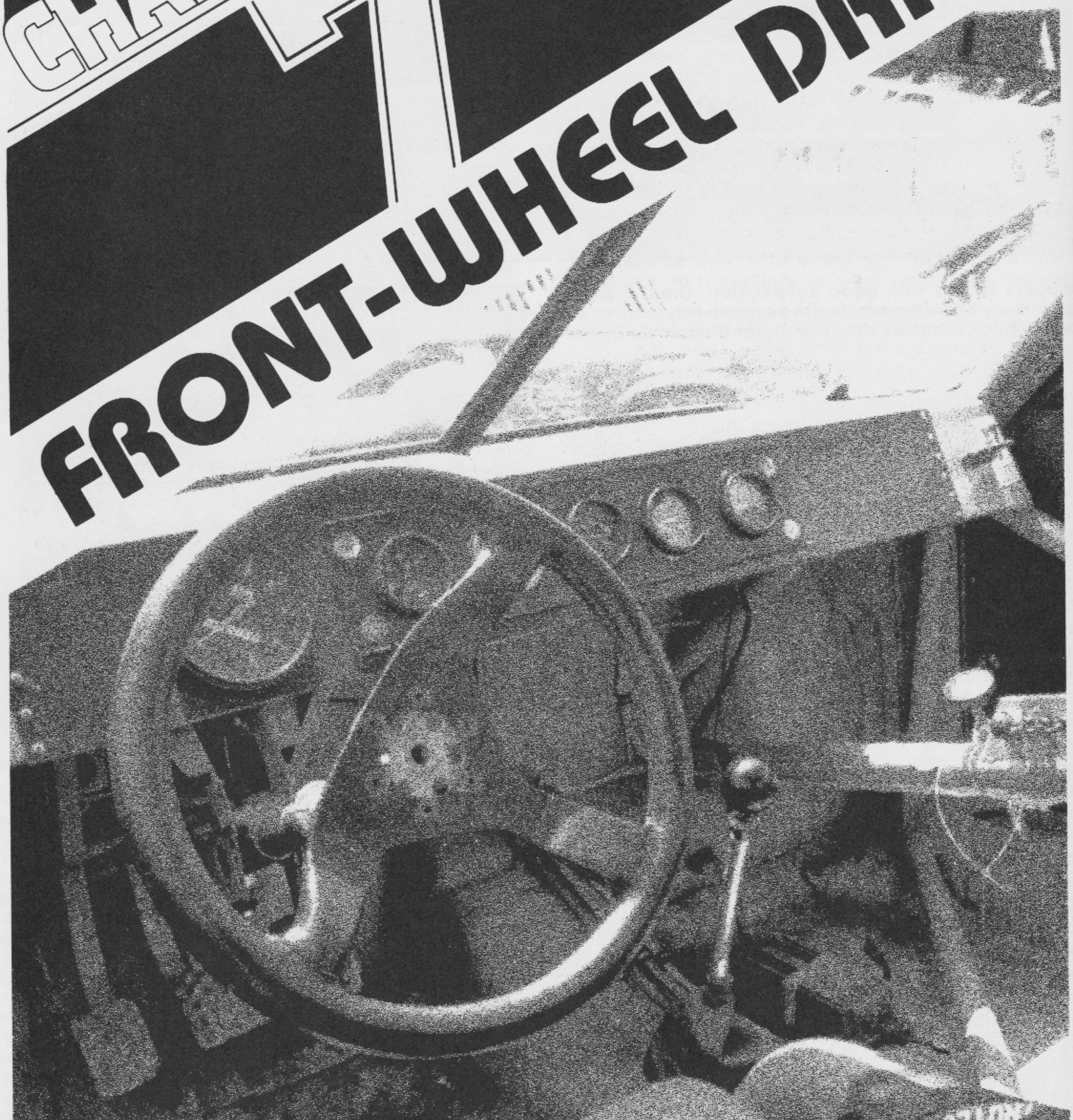
A fine-mesh wire screen over the radiator inlet will substantially reduce aerodynamic drag at high speed, since airflow will be very similar to that impinging on a smooth, solid surface . However, the screen will reduce radiator airflow by about 50%, so well-designed duct work and shrouding may be required to maintain adequate airflow and cooling.

less effective at lower speeds.

If you're a "do-it-yourselfer," you may wish to consider another aerodynamic device. Air flowing into the stock radiator grille will cause a great deal of turbulence and induce drag. But blocking off the grille area (with an aluminum sheet) will cause overheating unless special ducting is added that reroutes radiator airflow. There is a simpler solution: Installing very fine-mesh wire screen over the grille opening allows some air to enter the radiator (about 50% normal flow—usually enough to keep the engine cool) while greatly reducing turbulence. To high-speed air, the screen "looks" like a smooth, solid surface, and much less drag is induced.

CHAPTER 7

FRONT-WHEEL DRIVE



AUTHOR'S SIDENOTE

Authoritative information on front-wheel drive is not easy to come by. However, Chrysler Corporation—the leading front-drive manufacturer in the country—has not only done its homework on this new technology but freely disseminates this information, rather than keeping it "locked up" in development labs and test cells. The information in this chapter is due, in part, to this open-minded attitude and to the cooperation from the personnel at the Chrysler/Shelby center, with particular thanks to Neil Hannemann, Dennis Lopez, and Jerry Mallicoat.

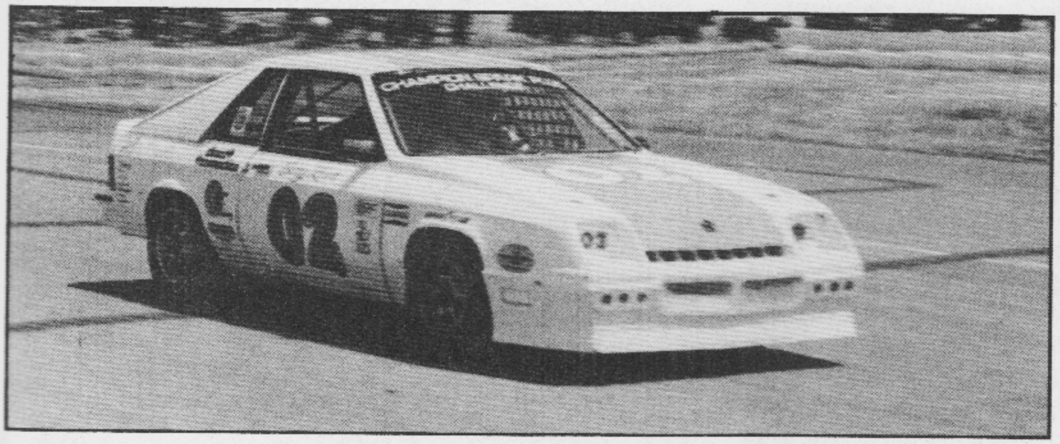
FRONT-WHEEL-DRIVE SUSPENSION

Front-wheel drive is currently the "rage" of automobile manufacturers worldwide, and more and more vehicles are being redesigned to utilize this new technology. However, many top-performance cars in the world are still front-engine/rear-drive or rear-engine/rear-drive, and all true race cars (such as Formula 1, NASCAR, Indianapolis Championship cars), and top road-racing sedans are still firmly rooted in rear-wheel-drive technology. Despite the rear-drive success record, a growing number of performance cars for street use are incorporating front-wheel drive.

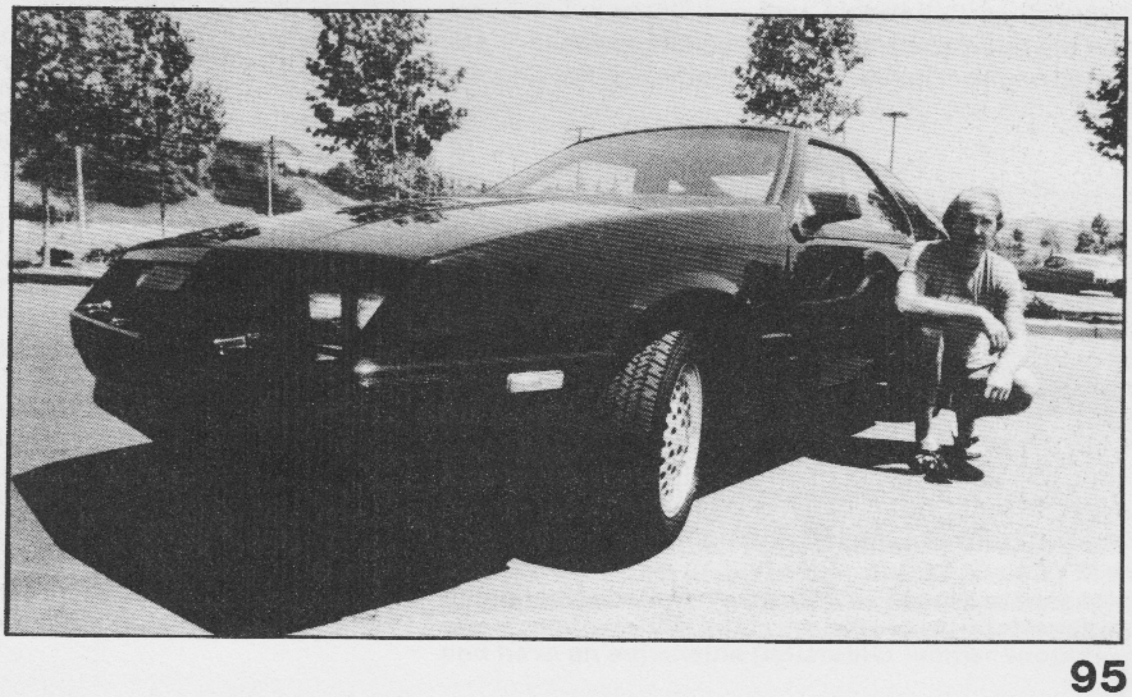
Is this shift to front-wheel drive permanent and does it mean the end of the traditional rear-drive sports car, or is the front-drive phenomenon a will-o'-the-wisp? We will attempt to clarify this issue and some of the mysteries of front-wheel drive by comparing the advantages and disadvantages of—and technical insights into—front-wheel-drive technology. But most importantly, we will reveal that Chrysler and independent racers are building better-handling front-drive vehicles.

WHY FRONT-WHEEL DRIVE?

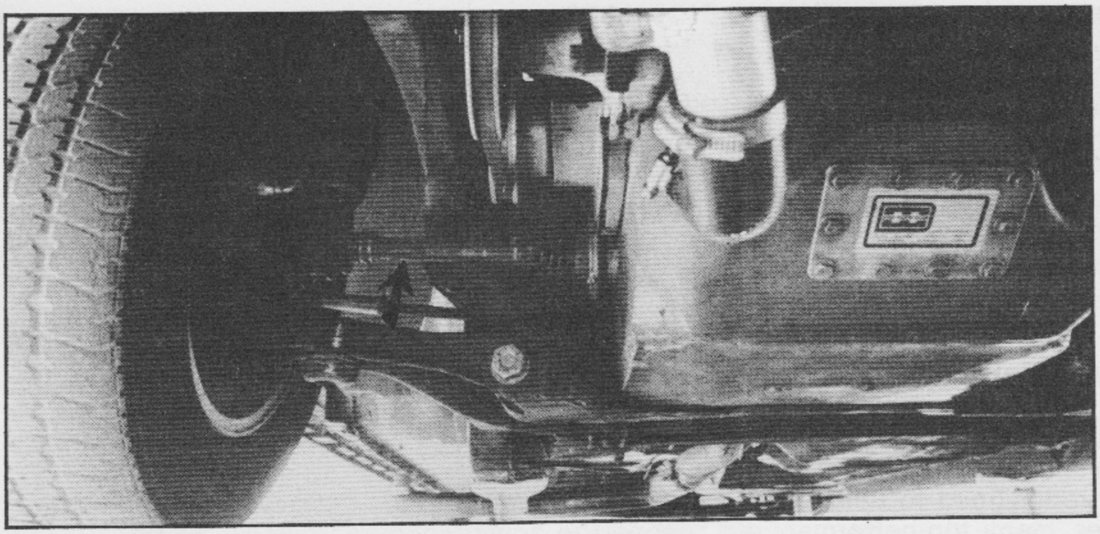
The decision to convert to front-wheel drive by



This Charger typifies the new breed of front-wheel-drive "street fighter." While not competitive with larger rear-drive V-8 cars, they hold their own even with smaller rear-drive cars. The solid, reliable engine and drivetrain in these vehicles will take a remarkable amount of punishment; all-out race preparation requires little more than changing the oil.



There is little doubt that the new front-drivers are the modern version of the old muscle cars; they offer good performance, superior handling, great looks, reliability, and excellent economy. While Mopar owners, like stalwart fan Jim Salonisen (shown here with one of his many Chyrsler vehicles), may have an old 440 GTX or Hemi Cuda parked in their garage, it's the front-driver that delivers the day-to-day transportation and fun!



Front-wheel-drive technology incorporates both the driveline and suspension in one compact package. The right "half-shaft" of Joe Varde's racer (arrow) and almost all of the remaining driveline components are factory stock; a testament to the brute strength that Chrysler builds into their front-drivers.

most manufacturers is an offshoot of the current downsizing trend that began with the first oil fiasco in the early 1970s. In a word, it's all "economics." To meet the requirements of fuel efficiency, passenger room, and comfortable ride, a new way to package the drivetrain was developed.

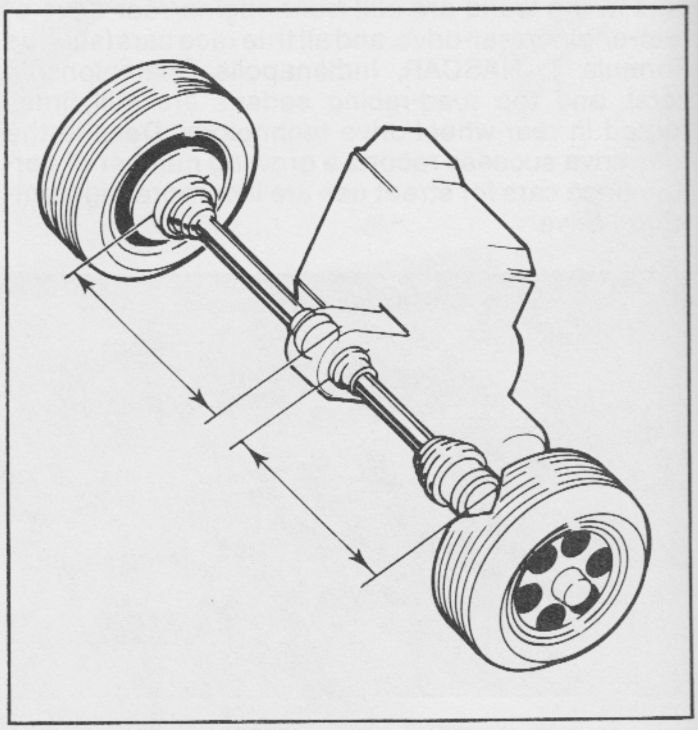
While front-wheel drive is really nothing new, it was virtually ignored until new demands from the buying public forced manufacturers to re-evaluate their basic concepts of how a car should be put together. Engineering models demonstrated that a car can be built small, compact, and inexpensively if it is a front-driver. With front drive, there is no drivetrain intrusion into the passenger compartment, no separate rear-drive axle assembly or driveshaft; and once the initial development costs are amortized, the vehicle assembly costs are lower.

Another reason front drive has become widely adopted by manufacturers is that one basic drivetrain can be installed in several different vehicles. This is quite evident in the derivatives of the original Chrysler K-car; from the humble beginnings of the Aries and Reliant, there now exist the Omni/Horizon, 600 Series, 400 Series, LeBaron, the new mini van, and the soon-to-come H car. All of these vehicles are based on the original K-car floor plan and the original front-drive powertrain.

SOME PROBLEMS

Of course, there were many engineering problems to overcome in building an acceptable front-driver that would exhibit most of the stable handling characteristics of a rear-driver.

One of these problems was *torque steer,* which can be a serious drawback in a front-driver if the basic design is inferior. Torque steer causes a vehicle to jump (steer) from side to side when accelerating through a corner or on a bumpy road. This scary event is most often caused by non-uniform power transfer between the left and right driving wheels. A mild case of torque steer makes a front-driver unpleasant to drive; a severe case can force a car out of control.



Torque steer can be a serious problem with front-drivers. The common contributing factor is the use of unequal-length half-shafts. Chrysler has done an admirable job in eliminating torque steer by designing a front-drive package that uses equal-length half-shafts with no compromise in strength and reliability.

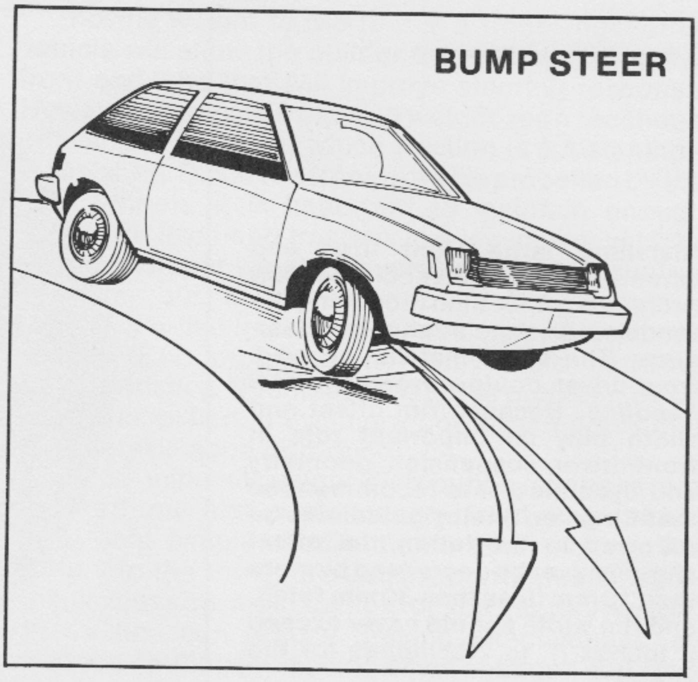
So torque steer had to be eliminated from the new generation of front-drive cars, and Chrysler's design is quite successful. Most torque-steer problems can be eliminated by using equal-length "half shafts" (a fancy term for the drive shafts on a front-driver or an independently-sprung, rear-drive system). Optimizing other design criterion—such as scrub radius, alignment, and particularly rim offset—will further reduce torque steer. In fact, less-than-optimum rim offset will directly induce torque steer, so using rims that alter this important dimension is strongly discouraged. Chrysler's current design is almost completely free of torque-steer tendencies, with the only remnant occurring in a severe jounce condition.

Another problem in front-drive cars is the unusual response to driver input, especially during oversteer and understeer (more on this later). In high-speed cornering, all front-wheel-drive cars require unique driving techniques, of which the average driver is completely unaware. And since, "Joe Average" has little idea how to handle a standard rear-drive car at high-speed, the public demand to change front-drive handling to conform to more "normal" tendencies has been almost nonexistent. However, Chrysler and several independent race teams are currently working to improve front-driver cornering characteristics.

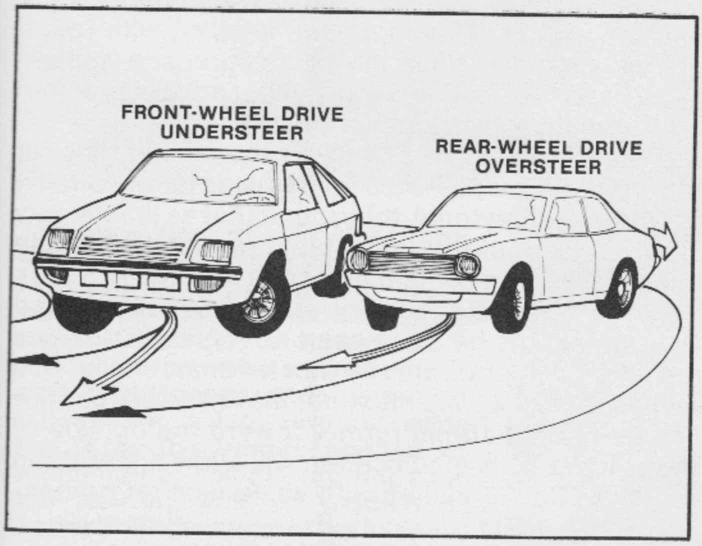
The peculiar problems of a front-driver may never be completely eliminated, but at least the future looks bright for continued improvements.

The final consideration in front-drive design was aesthetics; meaning everything from comfort and ride to beauty. Although front drive may not appeal to those with a passion for the "good old rear-wheel-drive design," Chrysler succeeded in meeting the standards of hundreds of thousands of buyers. The new Chrysler cars are built well. They feel solid and ride smoothly. The doors and hood fit. Although the lack of a trans lump in the floorboard may bother the rear-driver aficionado, it adds room and keeps the driver's compartment cooler. All things considered, the new line-up of front drivers is impressive. (If Chrysler improved the instruments, dash design, and fine detail work, it would be hard to find anything to dislike.)

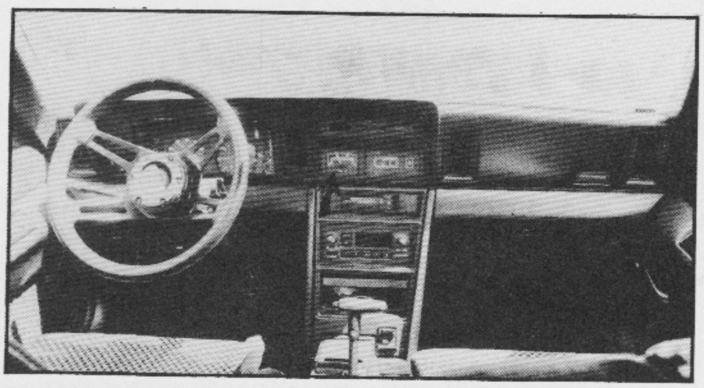
So, to re-address the question posed earlier about the permanence of front-wheel drive, the answer appears to be that while front-drivers are here to stay, the traditional rear-wheel-drive sedan or sports car is by no means going the way of the dinosaur, because rear-drive has too many advantages in handling, braking, acceleration, plus that all-important visceral "feel." But with Chrysler's commitment to front-wheel drive, there is little likelihood that the future will see any high-performance, rear-drive Mopars, like those of the 60s and 70s; and



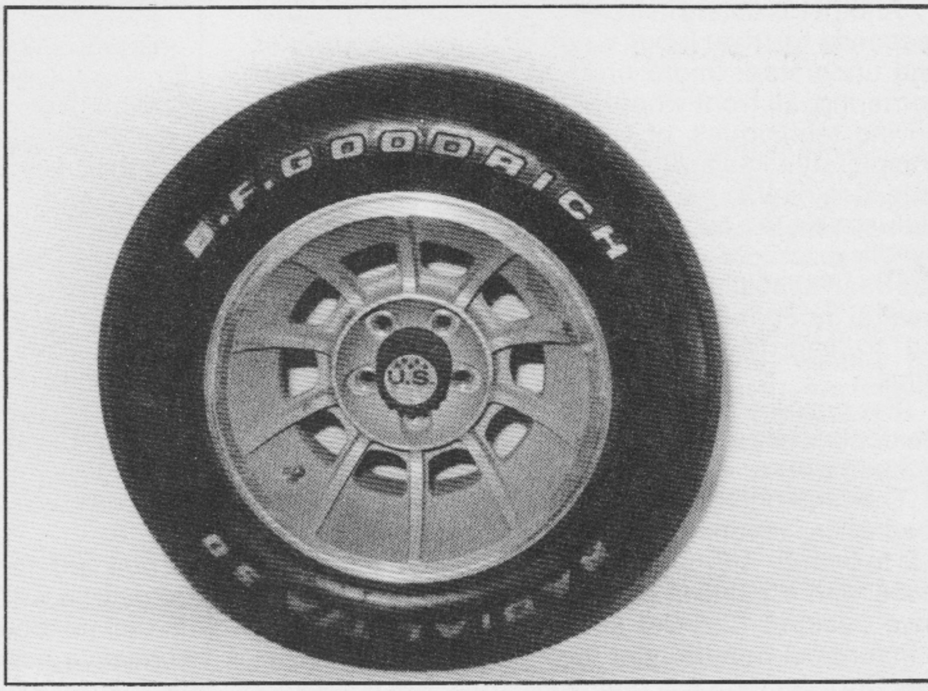
Since front-wheel-drive vehicles rely on only the front tires for both acceleration and steering, some unusual problems can develop. If the acceleration (torque) to both front tires is not uniform, the chassis can react by "steering" to the side of greater torque. For example, if uniform torque is not maintained during single-wheel jounce (as in the above illustration), the vehicle may veer off track. Chrysler has eliminated most bump-steer tendencies by incorporating equal-length half shafts and computer-aided design in all their front drivers.



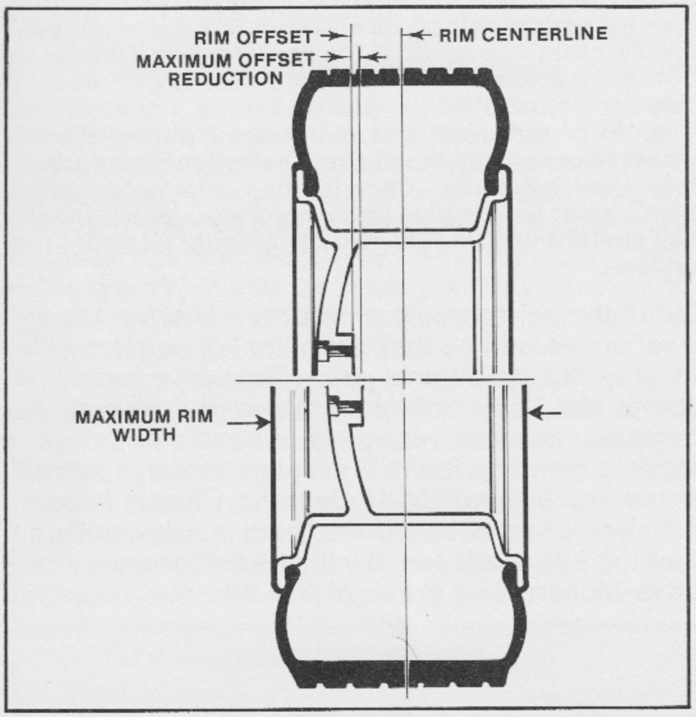
Both front-drivers and rear-drivers handle very well, but the demands on the driver are not the same. In some cases, they're completely opposite. For example, the application of heavy throttle in a high-speed turn will reduce rear-wheel traction in a typical rear-driver (vehicle shown at right), causing oversteer. Applying the throttle on a front-driver, however, will reduce front-wheel traction, causing understeer. But most drivers will never detect these differences which only become evident at road-racing speeds.



The clean and sporty interior of Chrysler's front-wheel-drive cars complement the performance of the popular turbocharged 2.2-liter four cylinder that powers many models in the lineup. Incorporating the latest technology, including the famous "talking dash," the instruments are large and easy to read (note the boost gauge—arrow). The seats are attractive, comfortable, and have an adjustable (inflatable) lumbar support.



Installing large front tires and wheels on a rear-driver creates few problems, other than modifying the fenderwells for adequate clearance. The same installation on a front-driver could *adversely affect handling*. Because rim offset and width play an important role in front-driver suspension geometry and dynamics, it is recommended that Chrysler/Shelby guidelines be followed to the letter: rim offset should never be decreased by more than 10mm (less than 30mm total), and rim width should never exceed 8 inches (7 to 7.5 inches for the street).



many stodgy rear-drive "family" sedans will eventually be replaced by more economical, space-efficient front-drive models.

IMPROVING THE FRONT DRIVERS: TIRES, TIRE SIZES, AND WHEELS

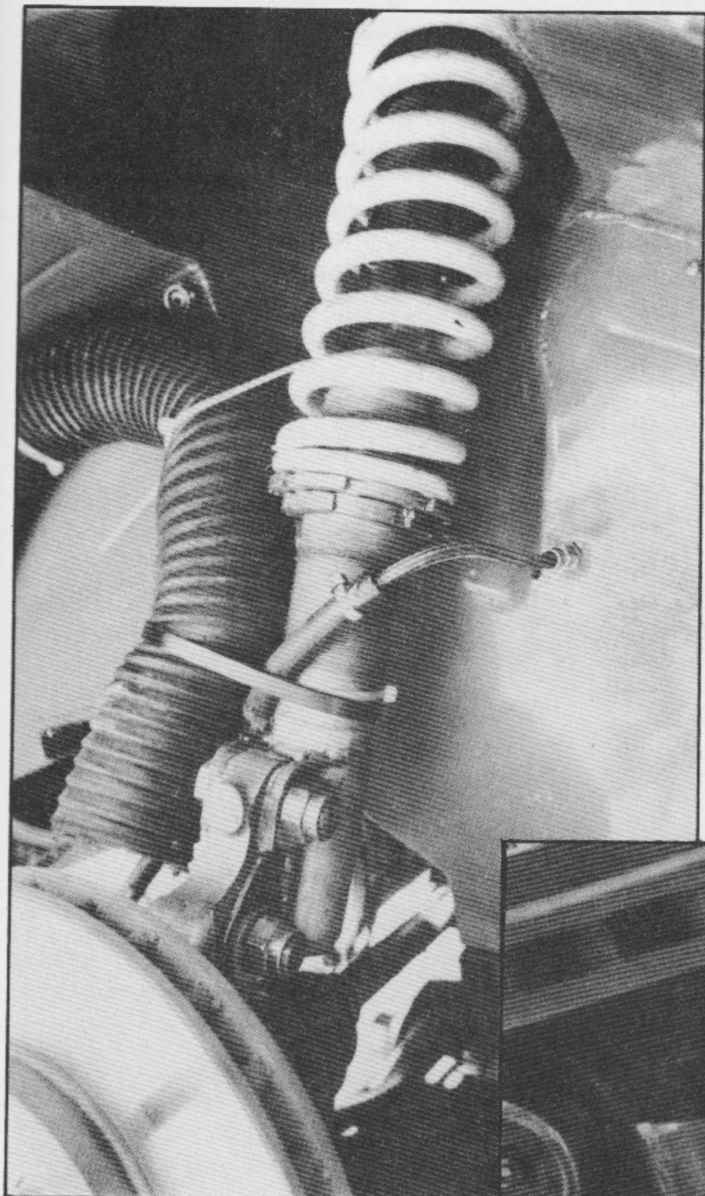
Installing wider tires and wheels is one of the best ways to improve the handling of any car. The foregoing statement is one of the most general and widely-accepted statements about high-performance suspension design. When the installation of wider tires and wheels is coupled with significant improvement in other suspension components, the level of handling can increase dramatically. This additional rule of thumb is just as applicable to front-drive cars, but there are a few caveats that must be taken into consideration.

With a rear-drive car, installing 10-inch-wide rims and fat tires on both the front and rear can be accomplished without suffering any serious problems other than finding adequate fender clearance. This is true even with rim offsets of up to 1.5 inches beyond stock. As long as the suspension has been built to augment these tire and rim-size changes, the installation will yield improved handling with few, if any, side effects. When this same approach is taken in a front-driver, the owner or builder must stay within an acceptable range of rim offsets.

When Chrysler tested wider-than-stock rims for the front-drivers (up to 8.0 inches wide for the Shelby Charger and Daytona), they found that as the rim was offset outside of the acceptable range, the vehicle suffered from a worsening case of torque steer. Therefore, Chrysler/Shelby engineers recommend that rim offsets be decreased no more than 10mm (approx. 0.5-inch) from the stock 40mm offset. *This means that new rims must not move the tire centerline more than 10mm further toward the outside of the chassis.* This modification will allow a street rim width of 7 to 7.5 inches with some nice fat rubber.

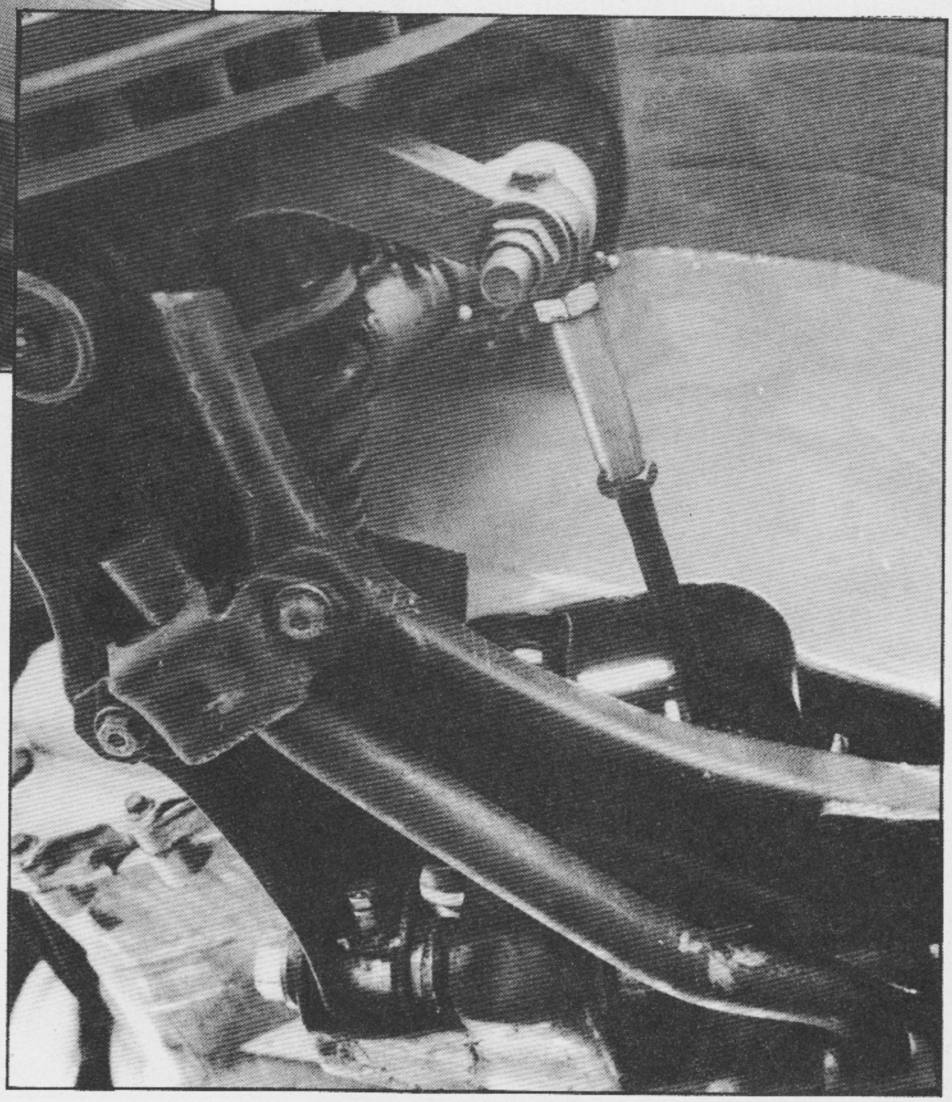
WEIGHT DISTRIBUTION

Another common method to improve rear-wheel-drive handling is to alter the weight distribution by removing weight from the front and adding it to the rear. For a rear-driver, this has several advantages: 1) it allows one to run a stiffer front end for better handling; 2) it places more weight over the rear tires for increased traction; and 3) the reduced front-end weight will substantially increase steering response. A front-driver will also benefit from weight-distribution modifications but to a lesser extent.



Moving weight to the rear in a front-wheel-drive vehicle will allow the builder to use a slightly stiffer front end, and that will improve steering response. However, reducing front-end weight soon reaches a critical point beyond which traction is substantially diminished. Since the front-driver is propelled by the front wheels, it is important to maintain enough weight for front-end traction. In a low-power vehicle, substantial rear-weight movement rarely presents a problem; but in higher-performance, front-drive sports cars, like the current Chrysler vehicles, it can cause REAL problems in tractive and handling characteristics. Needless to say, this problem will be further amplified in the more powerful models that Chrysler will soon introduce.

Since minimal weight-distribution changes are required on high-performance, front-drive street cars, only simple modifications (like moving the battery to the trunk) are usually performed. In addition, excessive modification of the factory weight distribution can reduce braking effectiveness; so proceed cautiously, making small changes and testing the results. Far greater improvements in handling can be made by simply upgrading the tires and wheels and increasing the spring and anti-sway-bar rates.



The new line of Chrysler front-wheel-drive cars can be modified into surprisingly competitive race cars. The "major" modifications include replacing the stock MacPherson strut with a competition part that incorporates a spring adjustment, a high-rate anti-sway bar (note the unique method of attachment), special rod ends for the steering linkage, grade-8 or better bolts throughout the suspension, and wide tires with light-weight wheels. Despite the obvious differences in front-driver suspension, the "theoretical-handling line" is still applicable (see page 37) and can be quite useful in setting up a street or race car.

FRONT-WHEEL-DRIVE BRAKING

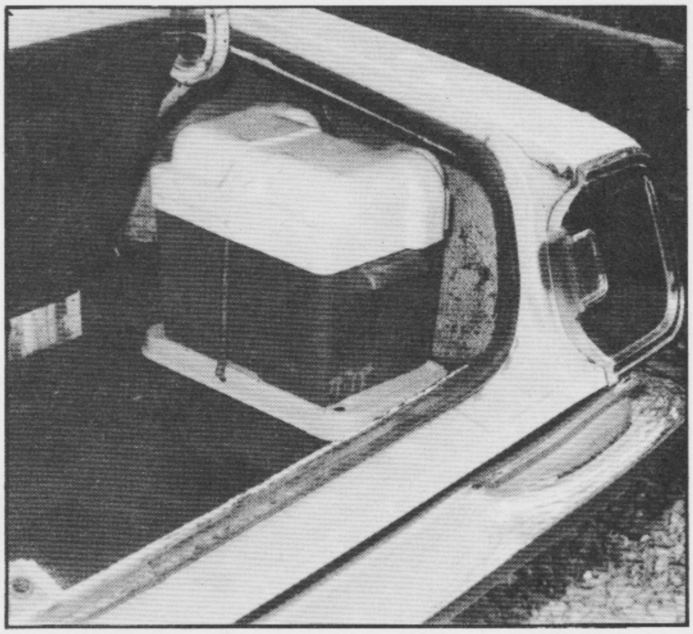
Front-wheel-drive cars carry over 60% of their static weight on the front tires. This means that brakes and brake proportioning are even more critical. In a front-driver, it is important to remember that very little additional weight will be transferred to the front (relatively speaking); so brake modifications are normally required only on the front. Increasing the size of the rear brakes can cause premature brake lock-up and oversteer. At this point in the development of front-driver technology, Chrysler recommends only front-brake improvements and all rear modifications limited to lining-compound enhancements, rather than increasing size or capacity.

Front-driver brake proportioning follows the same rules as a rear-drive car. Basically, the front brakes should lock up before the rear brakes. The installation of a manually-adjusted proportioning valve would aid in brake tuning, but it is only required if larger aftermarket tires and wheels have been installed. Very little development has been done in the area of front-driver brakes, so the enthusiast should proceed with caution and thoroughly test all modifications.

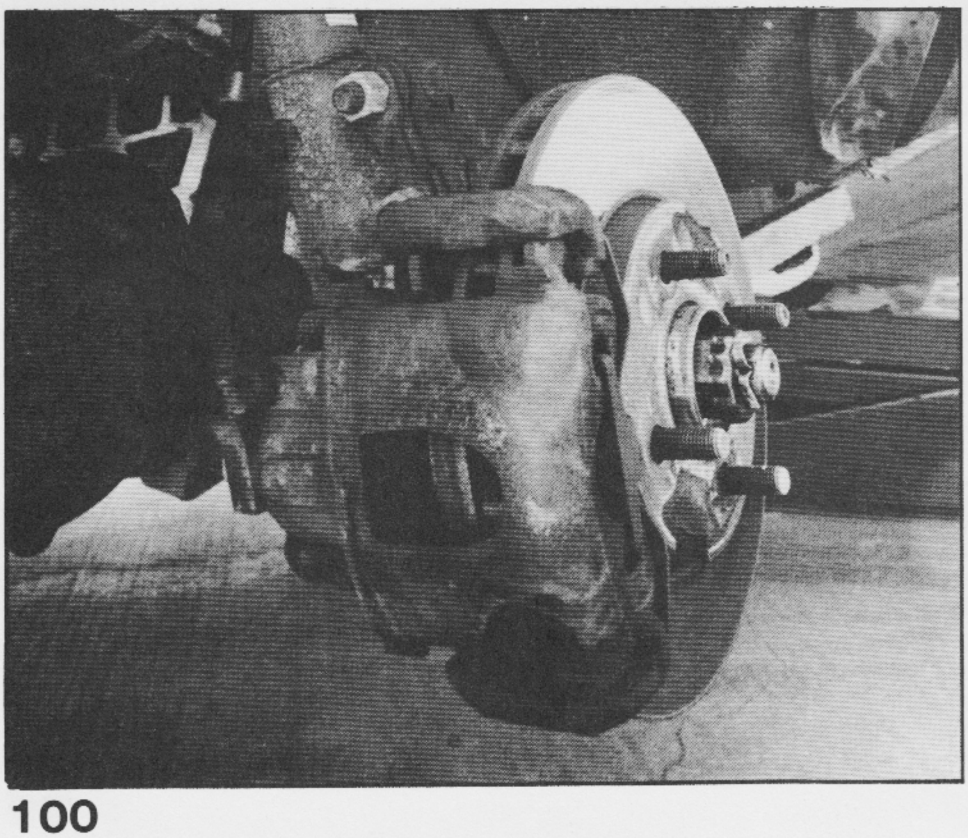
Chrysler engineers have clearly stated that production brakes on the current top-of-the-line performance cars, such as the Charger and Daytona, are "state of the art." There are no factory upgrades available, and any improvements to these brake systems would have to be done with aftermarket parts. However, if you have a low-performance model and would like to upgrade the braking system, the best improvement for the money is available in the more-capable Charger or Daytona front-brake system, sold through Chrysler parts.

FRONT-DRIVE HANDLING CHARACTERISTICS

The theories we have developed in the previous chapters for rear-drive cars apply equally well to a front-driver (e.g., using the theoretical handling line to establish optimum roll-couple distribution). However, there are important differences in front-drive handling, ranging from subtle changes in low-speed cornering and braking to substantial changes at road-racing speed, requiring radically different driver inputs. When you consider the dynamic loads that the front tires must endure, it becomes more clear why a front-driver makes such different demands on the driver.

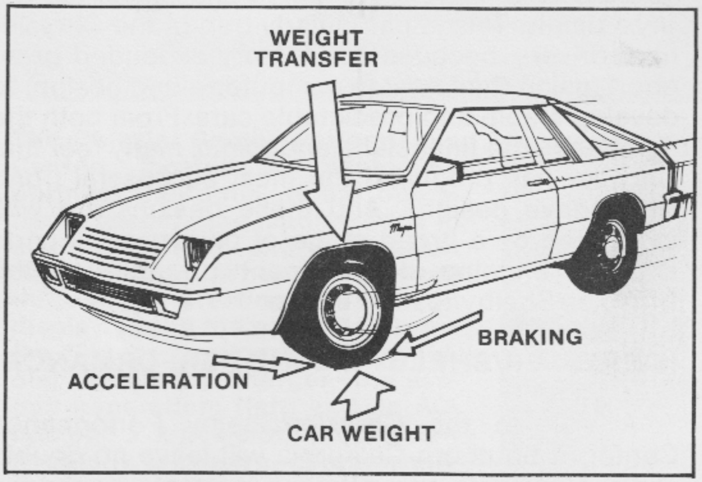


Moving the battery to the trunk will aid weight distribution on both front- and rear-drivers. And the lower ambient temperature in the trunk can extend battery life. But a solid mount must be used—like this marine battery box—to prevent battery-acid leakage; poor mounting could allow the battery to overturn, possibly causing an electrical short, a small explosion and/or fire.



While not as massive as V-8 disc brakes, these factory mini-muscle-car brakes work well on the street, and with good metallic pads they can provide acceptable braking in amateur competition events (however, it is recommended that the studs be replaced with larger 1/2-inch bolts, or the rotors be replace with 1985 models that use a 5-bolt pattern).

A rear-drive car loads the front tires only during hard braking and cornering. Under the demands of acceleration, the front end raises, transferring weight to the rear and improving rear-wheel traction. This ubiquitous design often generates controllable oversteer when the throttle is applied during hard cornering (caused by reduced rear-wheel traction from acceleration). But a front-wheel-drive car not only brakes and corners with the front tires, it also accelerates with the same rubber. Therefore, applying throttle during hard cornering will *reduce front-end traction*, causing understeer. In other words, the front-wheel racer must condition himself to *apply the throttle when the car is oversteering and reduce the throttle when the car is understeering*—exactly the opposite of the techniques used in a rear-driver.



The front wheels of a front-driver carry the loads of acceleration, deceleration, cornering, (and the added load of weight transfer in these situations), and they support their proportion of overall car weight, which is considerably greater than 50%. Chrysler front-driver suspension carries all of these loads with little adverse feedback to the driver, except for unusual throttle/handling characteristics in high-speed corners.

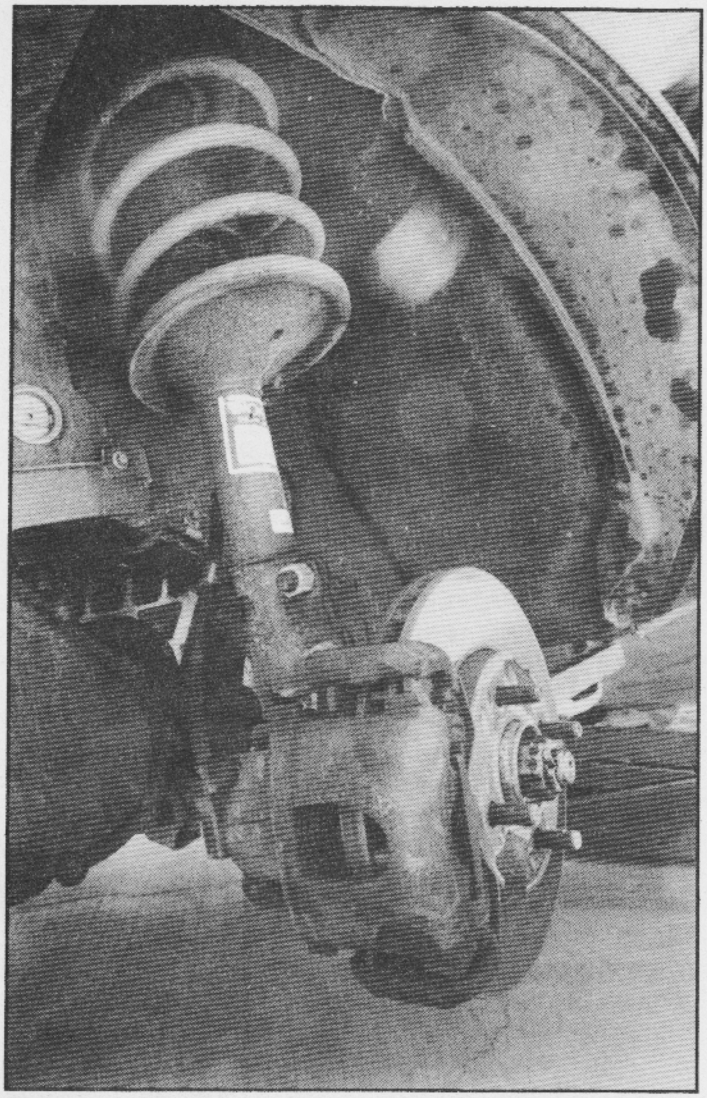
The added demands on the front rubber of a front-drive car (and the fact that the engine, transmission, and differential are located above the front wheels) require more weight on the front tires. In many designs, over 60% of the weight is over the front wheels. So this added weight, albiet necessary, adds even more load to the already overused front tires.

Obviously, the combination of weight distribution and throttle-induced understeer makes a racer's job quite unique. Getting a front-driver around a turn at high speeds is often best accomplished by entering the turn a little too fast, inducing oversteer. Then apply throttle (some say, "When it starts to slide out, just punch it and hang on!") to counter the oversteer and induce a neutral slide with the car under acceleration. Simple, heh? Yeah, and for those of us who have made rear-wheel driving techniques second nature, some serious re-learning is in order.

THE MACPHERSON STRUT

The current front-drive cars use a MacPherson-strut front suspension. This system is less expensive to manufacture and has several advantages over link suspension, not the least of which is reduced weight. However, MacPherson suspension will not generate negative camber during hard cornering, which is inherent in a standard link-type suspension. The loss of negative camber adversely affects handling by allowing the inside tire to more easily "roll over" and break traction. Chrysler is aware of this problem and is considering several possible solutions, one of

Joe Varde's Charger (see page 105) is put through its paces at Riverside Raceway in Riverside, California. The chassis is designed to exhibit only slight lean for optimum weight transfer during cornering. The low front end, air dam, and 1.5° body-rake angle help keep the front syspension "glued" to the track at high speed. Although the tires, brakes, and many other components must be stock factory parts (sanctioning regulations), the Charger is fast, reliable, and a real crowd pleaser.



While not as sophisticated as double A-arm front suspension, the MacPehrson-strut is reliable, produces suprisingly good handling, and keeps production costs to a minimum (Chrysler mini-muscle front-drivers sell for well under $10,000).

which is a torsion-bar, wishbone suspension.

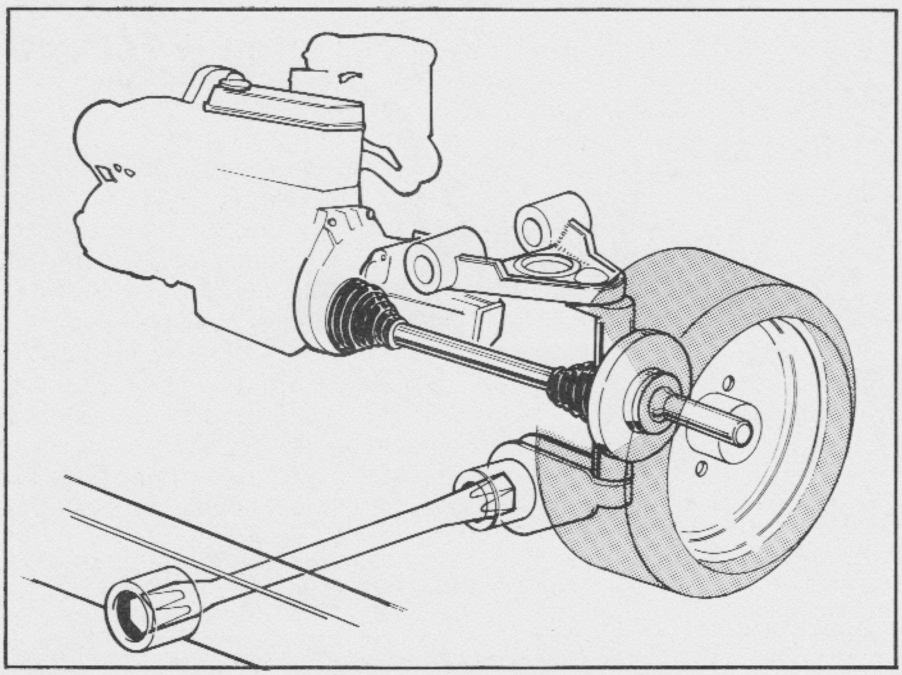
It is interesting to consider that to improve current front-driver handling, Chrysler may return to the same basic suspension system that was used in the 1960s and 1970s on cars that lead the performance-car era: the torsion-bar, wishbone design.

IN THE BALANCE

While we have mentioned several disadvantages inherent in front-drive design, they should not be considered serious defects, especially if the driver is aware of the limits of adhesion and drives the car accordingly. This is particularly true of the Chrysler front-drivers, because the factory expended great effort (using the latest in computer-aided design) to develop stable and predictable cars. From both the functional and financial standpoints, many feel that Chrysler has produced the most successful front-wheel-drive designs. And these designs are well supported by a broad range of performance parts and an ongoing development program at the Chrysler/Shelby testing center.

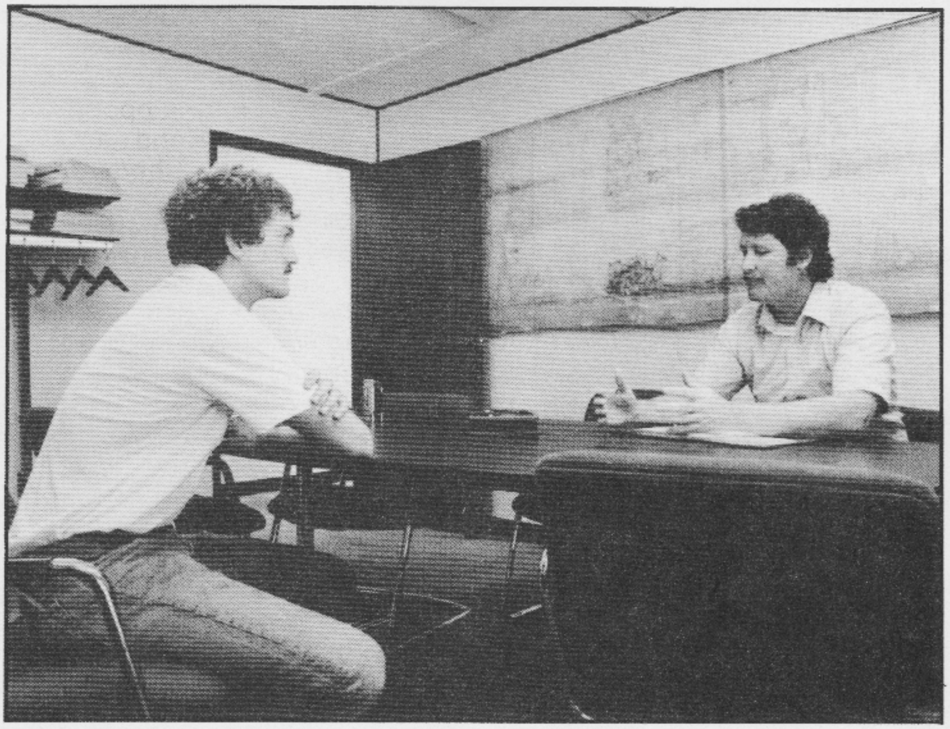
CHRYSLER/SHELBY MEANS PERFORMANCE

A visit to the Chrysler/Shelby Performance Center in Southern California will leave no doubts about Chrysler's commitment to front-wheel-drive performance and handling. The Shelby facility, located in a portion of a much larger installation used to prepare new cars for delivery to local dealers, is "home" for the leading edge in engine, chassis, and suspension development. And virtually all development revolves around front-drivers, with the exception of a rumored Dodge-powered Pantera (perhaps with a V-8); but don't hold your breath—at best it's a few years away. Chrysler/Shelby developments will lead the current lineup of Lasers, Daytona Turbo Zs, Shelby Chargers, and Omni GLHs into a future of



Handling and styling requirements may force Chrysler to return to torsion bars. While MacPherson struts perform well, they do produce less-than-optimum camber angles in hard cornering. And since hoodlines are getting longer and lower (Macpherson Struts require considerable working height), the A-arm wishbone design may once again be the answer.

The Chrysler Shelby Performance Center, located in Santa Fe Springs, California, is the operational base for the leading edge in engine, chassis, and suspension development. When we visited three very interesting cars were on display: (right) the original Cobra—the first generation Shelby; (middle) the Shelby Charger—the second generation; (left) and an AC-bodied, 2.2-powered prototype—hopefully, the next generation mid-engine hot rod.



The author Mike Martin (right) with Neil Hannemann (Chrysler/Shelby product-development engineer) at the Shelby Center discuss front-wheel-drive suspension technology. Most of the information presented in this chapter was obtained through the cooperation of Chrysler engineers and racers.

higher-performance and better handling options. In effect, new "mini-muscle" cars are in the making. In addition, both the current and future line will be augmented by Mitsubishi imports like the Conquest and Colt Turbo.

The sole purpose of this research, however, is not just to build better street cars. Chrysler has its sights set on continuing the initial success that its sponsored cars have had in IMSA road racing (see accompanying section: **Joe Varde's IMSA Charger**) and may attack a few street Grands Prix, autocrosses, and perhaps a couple of the amateur and professional road-racing categories. The Shelby Center is targeted to head development for these efforts; and in order to be successful, they must encompass all aspects of the vehicle, from engines to wheels and tires.

ENGINE DEVELOPMENT

Chrysler has several engine-development programs underway. In addition to further research into fuel injection and turbocharging, they are testing a positive displacement Camden (roots-type) supercharger along with a compound, 2-barrel carburetor (uses one barrel as a primary and the other as a secondary). And the most impressive development to date: the 16-valve (4 valves per cylinder), dual-overhead-cam engine. This engineering beauty may utilize any or all of the proposed induction systems, as well as side-draft Webers, Mikunis, or Solexes. There is little doubt that the soon-to-be-available engines will produce 160 to 190 horsepower (for the street versions), which brings up an interesting question: How does Chrysler intend to get that power to the ground and maintain stable handling on a front-driver? Chrysler/Shelby personnel feel that ongoing research will provide the answer: research from the lab, test track, and race course.

SUSPENSION

Niel Hannemann, product-development engineer

Chrysler P-parts add performance and handling to both front- and rear-drivers. The P-part program includes many outlets throughout the United States, including most Chrysler/Plymouth/Dodge dealers and many speed shops—like Vic Hubbard's Speed in Hayward, California—that stocks both P-parts and a wide range of speed equipment.

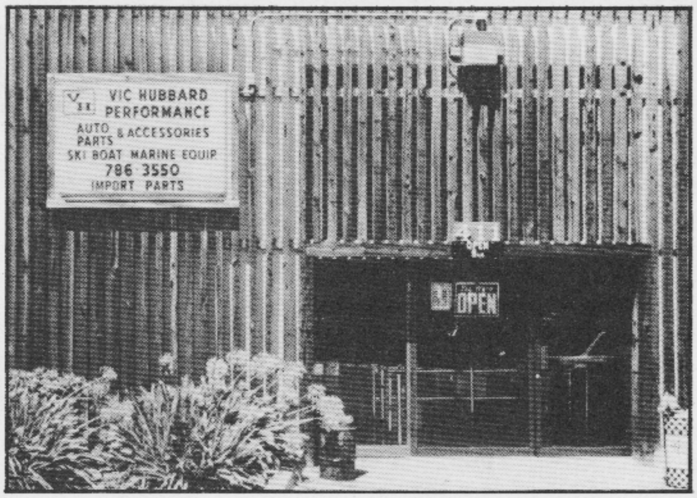
at the Shelby Center, spends his time studying front-drive suspension on the current Chrysler models. He probes into the inner workings to see where camber, wheel travel, torque steer, and the other suspension subtleties can be improved. He may use Chrysler's "number-cruncher" computer (on which the suspension was originally designed) to test his modifications. And when the numbers are right, Chrysler's skid-pad test reveals the bottom line. Neil is just one of the talented people who make front-wheel drive work better on the street and win more races at the track.

Neil indicated that installing stiffer front springs and shocks on MacPherson-strut cars will provide the best single improvement in handling; and all the required parts are available through Chrysler's Direct Connection program. But even with these modifications and better tires and wheels, Neil is the first to admit that the MacPherson-strut suspension has certain limitations that are not easily overcome. As horsepower climbs to the 250-300 level in road racing, it may become necessary to make extensive modifications to the basic design. However, Neil points out that the current configuration is no way near the end of its development curve, a remarkable fact considering Joe Varde's "easy" wins in his Dodge Charger.

A DIRECT CONNECTION

Chrysler's lead in front-drive technology is not just available to racers. The Direct Connection parts program is the factory's link to performance enthusiasts. To encourage owners of Pentastar front-drive performance cars to become interested in improving street handling and to build competitive cars for the track or autocross courses, Chrysler has instituted a performance program to supply a wide range of parts and technical help.

Many of the parts designed and developed for IMSA racing by Joe Varde Racing are available



through Direct Connection. In fact, if someone would like to race an IMSA or SCCA car, nearly all the pieces required can be purchased over a Direct Connection counter. The Direct Connection program, the Chrysler/Shelby Center, and Joe Varde's IMSA racer combine to give the Mopar owner a real advantage over the competition in both rear- and front-drivers.

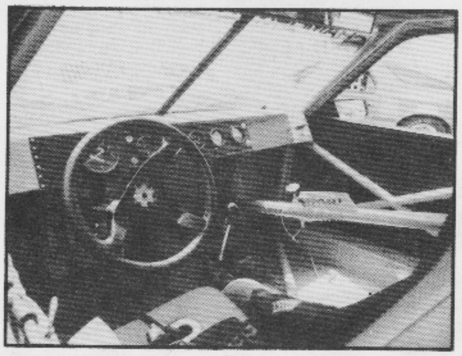
But having access to the right parts is only half the picture. Knowing what parts to select and how to install them are just as important. As soon as Chrysler institutes a "Tech Hot Line" to disseminate the latest engineering tricks, factory information will be as close as the phone. In other words, Chrysler is seriously promoting performance. Now we don't mean that you'll be seeing 40-foot trailers with the Chrysler insignia showing up at local road races, but you will be seeing more Chrysler products in the winner's circle, thanks to the open line of communication between the factory and the racers. The new Chrysler program will be quite similar to the performance project that Nissan/Datsun has called "Datsun Competition." Chrysler feels that they have the best front-drivers available, and they intend to keep it that way.

JOE VARDE'S IMSA CHARGER

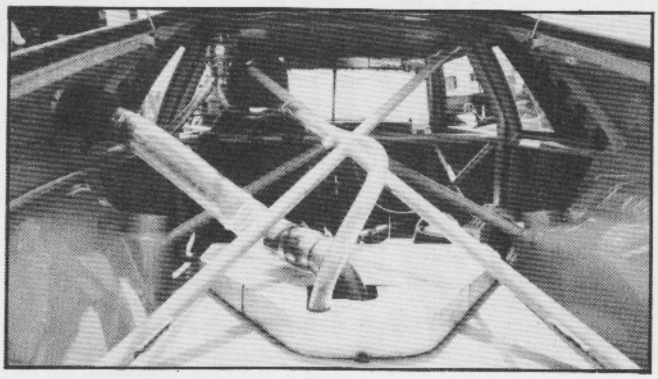
This beautiful Dodge Charger, driven by Joe Varde, is the spearhead of Chrysler's venture back into racing. And thanks to Joe's driving talents, the 2.2-powered, front-wheel-drive IMSA racer has been racking up win after win in the prestigious Champion Spark-Plug-Challenge Series races.

Joe Varde began his professional racing career with motorcycles and cars in 1964. In 1978, Joe completed his first IMSA road-race project (a Gremlin) and began an almost unending series of wins. In 1981, Joe's association with Chrysler's Dick Maxwell resulted in the current Dodge Charger project. Joe not only handles the driving chores but also oversees the chassis construction, engine modification, and fine tuning.

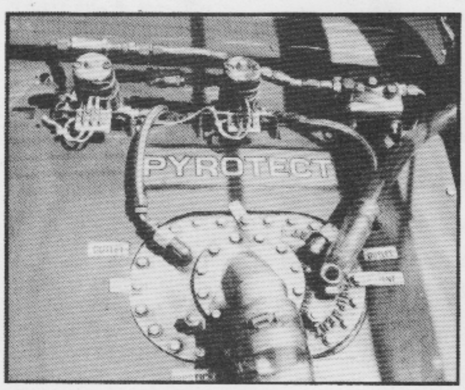
With such an impressive record of winning and reliability, the Chrysler front-drivers are certain to play an important role in future factory-backed and amateur racing. Front-drivers are reasonably inexpensive to build, require minimum modification (particularly the engine and drivetrain), and are fun to race. Parts developed by the racing efforts of Joe Varde and others have already found their way into the hands of street enthusiasts—through Chrysler's Direct-Connection program—bringing "race-track fun" to many boulevard blasters.



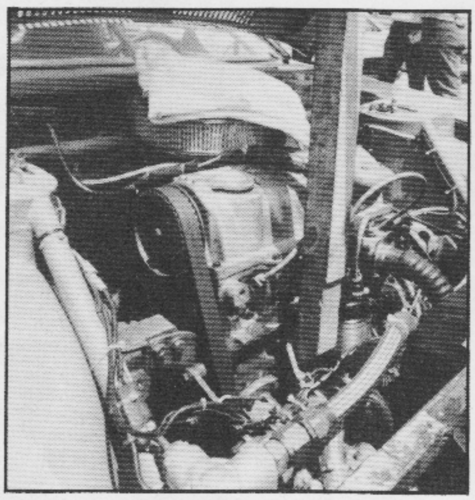
A clean and efficient drivers compartment is a necessity in any performance car, especially one designed strictly for competition. Electrical switches are mounted on the diagonal support for the roll cage, providing easy driver access. Just out of the picture (hidden by the driver's seat) is a lever (connected to an adjustable rear sway bar—see page 34) that alters the rear roll couple as the fuel tank empties.

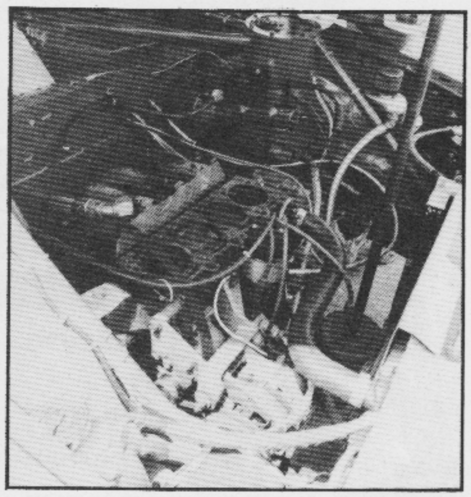


Joe Varde uses the basic Chrysler "pyramid" roll cage, incorporating cross bracing to maintain chassis rigidity. This design also uses less material and provides a substantial weight saving.



Safety ranks high on the priority list in any competition car. This fuel cell assembly not only incorporates multiple pumps and filters all connected with aircraft line, but also uses a safety cutoff for the driver's protection.





While not the height of sophistication, due to sanctioning body rules, the engines in these front-drivers still put out respectable horsepower with good reliability. Despite the rather "crude" devices used to hold the hoods open, top-quality materials and parts are used where they count.

MOPAR SUSPENSIONS APPENDICIES

LIST OF SUPPLIERS

ANTI-SWAY BARS

ADDCO Industries
26 Watertower Road
Lake Park FL 33403
(305) 844-2531

Martin Automotive Design Co.
1084 Harbor Way
Rodeo CA 94572
(415) 799-3603

AIRCRAFT HOSE AND FITTINGS

Aeroquip Corporation
300 S. East Avenue
Jackson MI 49203-1972
(517) 787-8121

Earl's Supply Co.
14611 Hawthorne Blvd.
Lawndale CA 90260
(213) 772-3605

BRAKE COMPONENTS

Hurst Performance, Inc.
(Airheart disc brakes)
50 W. Street Road
Warminster PA 18974
(215) 672-5000

Lamb Components
1259 W. 9 Street
Upland CA 91786
(714) 985-1901

Martin Automotive Design Co.
1084 Harbor Way
Rodeo CA 94572
(415) 799-3603

CAMS

Cam Dynamics
3926 Runway Road
Memphis TN 38118
(901) 794-2870

Crane Cams
100 N.W. 9 Terrace
P.O. Box 160
Hallandale FL 33009
(305) 457-8888

Crower Cams & Equipment
3333 Main Street
Chula Vista CA 92011
(619) 422-1191

Engle Racing Cams
1621 12 Street
Santa Monica CA 90404
(213) 450-0806

Iskenderian Racing Cams
16020 S. Broadway
Gardena CA 90248
(213) 770-0930

Lunati Racing Cams
3871 Watman Avenue
Memphis TN 38118
(901) 365-0950

Sig Erson Racing Cams, Inc.
550 Mallory Way
Carson City NV 89701
(702) 882-1622

CARBURETION

Braswell Carburetion
1650 E. 18 Street, Unit P
Tucson AZ 85719
(602) 884-7282

Carter Carburetor Co.
9666 Olive Street Road
St. Louis MO 63132
(314) 997-7400

Holley Replacement Parts Div.
11955 E. 9-Mile Road
Warren MI 48090
(313) 497-4245

Mikuni American Corp.
8910 Mikuni Avenue
Northridge CA 91324
(213) 873-2101

Mike Jones Carburetion
7602 Talbert Avenue
Huntington Beach CA 92647
(714) 848-5500

CHASSIS COMPONENTS AND SERVICES

A & A Fiberglass, Inc.
1534 Nabell Avenue
Atlanta GA 30344
(404) 762-9631

Advance Adapters, Inc.
P.O. Box 247
Paso Robles CA 93447
(805) 238-7000

Chassis Engineering
150 Evernia St.
Jupiter FL 33458
(305) 747-2400

Dick Guldstrand
11924 W. Jefferson Blvd.
Culver City CA 90230
(213) 391-7108

Ed Hamburger's Parts, Inc.
1590 Church Road
Toms River NJ 08753
(201) 240-3888

Genuine Suspension
2430 W. 5 Street
Santa Ana CA 92703
(714) 547-5503

Goerlich's
One John Goerlich Square
Toledo OH 43693
(419) 259-3236

Hellwig Products
16237 Avenue 296
Visalia CA 93291
(209) 734-7451

KYB Corp. of America
901 Oak Creek Drive
Lombard IL 60148
(312) 620-5555

Martin Automotive Design Co.
1084 Harbor Way
Rodeo CA 94572
(415) 799-3603

Moroso Performance Products, Inc.
80 Carter Drive
Guilford CT 06437
(203) 453-6571

Mr. Gasket Co.
4566 Spring Road
Cleveland OH 44131
(216) 398-8300

Quarter Master Industries
1350 Howard St.
Elk Grove Village IL 60007
(312) 593-8999

Quickor Engineering
6710 S.W. 111 Avenue
Beaverton OR 97005
(503) 646-9696

Rancho Suspension Products
P.O. Box 5429
Long Beach CA 90805
(213) 630-0700

VSE Products
100 Calle Del Oaks
Del Rey Oaks CA 93940
(408) 899-4359

CHRYSLER PARTS AND LITERATURE

Chrysler Direct-Connection Cat. Ctr.
20026 Progress Drive
Strongsville OH 44136
Chrysler Performance Parts
P.O. Box 1081
Warren MI 48090

Martin Automotive Design Co.
1084 Harbor Way
Rodeo CA 94572
(415) 799-3603

COOLING (FANS)

Electric Fan Engineering
P.O. Box 723
Montclair NJ 07042
(201) 783-5095

Flex-a-Lite Consolidated
4540 S. Adams
Tacoma WA 98409
(206) 475-5772

Mr. Gasket Co.
4566 Spring Road
Cleveland OH 44131
(216) 398-8300

Perma-Cool
671 E. Edna Place
Covina CA 91723
(818) 967-2777

DRY-SUMP SYSTEMS

Valley Head Service
19340 Londelius
Northridge CA 91324

Weaver Brothers, Ltd.
1980 Boeing Way
Carson City NV 89701
(702) 883-7677

FUEL CELLS

Aero Tec Labs, Inc.
Spear Road Industrial Park
Ramsey NJ 07446
(201) 825-1400

Racecraft Performance Products
313 Stokes Avenue
Trenton NJ 08638

HEADERS

Doug Thorley Headers
1561 Commerce St.
Corona CA 91720
(714) 735-7280

Eagle Headers
8341 Canoga Avenue
Canoga Park CA 91304
(818) 998-5911

Headman Headers
9599 W. Jefferson Blvd.
Culver City CA 90230
(213) 839-7581

Hooker Headers
1024 W. Brooks St.
Ontario CA 91762
(714) 983-5871

Walker Manufacturing
1201 Michigan Blvd.
Racine WI 53402
(414) 632-8871

IGNITION

Accel Ignition
175 N. Branford Road
Branford CT 06405
(203) 481-5771

Allison Automotive
1613 Flower St.
Duarte CA 91010
(213) 966-8562

Autotronic Controls Corp.
6908 Commerce Avenue
El Paso TX 79915
(915) 772-7431

Mallory Ignition
550 Mallory Way
Carson City NV 89701
(702) 882-6600

Midway Industries
(Stinger ignitions)
15116 Adams St.
Midway City CA 92655
(714) 898-4477

Ronco/Vertex Ignition
463 Northtown Road
Blue Bell PA 19422
(215) 828-2150

Roto-Faze Ignitions
23136 Mariposa
Torrance CA 90502
(213) 325-8844

MANIFOLDS

Edelbrock Equipment Co.
411 Coral Circle
El Segundo CA 90245
(213) 322-7310

Holley Replacement Parts Div.
11955 E. 9-Mile Road
Warren MI 48090
(313) 497-4245

Weiand Automotive Industries
2316 San Fernando Road
P.O. Box 65977
Los Angeles CA 90065
(213) 225-4138

PISTONS & ENGINE COMPONENTS

Arias Industries, Inc.
13420 S. Normandie Avenue
Gardena CA 90249
(213) 532-9737

B & B Performance Sales
23190 Del Lago Drive
Laguna Hills CA 92653
(714) 586-0561

Keith Black Racing Engines
11120 Scott Avenue
South Gate CA 90280
(213) 869-1518

Brooks Racing Components
7091 Belgrade Avenue
Garden Grove CA 92641
(714) 893-0595

Childs & Albert
11030 Sherman Way
Sun Valley CA 91352
(213) 765-0988

Mr. Gasket Co.
4566 Spring Road
Cleveland OH 44131
(216) 398-8300

Pete Jackson Gear Drives
1905 Victory Blvd., Unit 9
Glendale CA 91201
(213) 849-2622

JE Pistons
15681 Computer Lane
Huntington Beach CA 92649
(714) 898-9763

Manley Performance Products
13 Race Street
Bloomfield NJ 07003
(201) 743-6577

Milodon Engineering
9152 Independence Avenue
Chatsworth CA 91311
(213) 882-4727

Ross Racing Pistons
11927 South Prairie
Hawthorne CA 90250
(213) 644-9779

Speed-O-Motive, Inc.
9534 S. Atlantic Blvd.
South Gate CA 90280
(213) 564-8082

Speed Pro/Sealed Power Corp.
100 Terrace Plaza
Muskegon MI 49443
(616) 724-5011

Total Seal Piston Ring Co.
2225 W. Mountain View, No. 17
Phoenix AZ 85021
(602) 242-9421

TRW Replacement Parts
8001 E. Pleasant Valley Road
Cleveland OH 44131
(216) 447-1879

Venolia Pistons
2160 Cherry Industrial Circle
Long Beach CA 90805
(213) 636-9329

SHOCK ABSORBERS

Bilstein Corp. of America
11760 Sorrento Valley Road
San Diego CA 92121
(714) 453-7723

Carrera Shocks
5412 New Peachtree Road
Atlanta GA 30341
(404) 451-8811

Lamb Components
1259 W. 9 Street Upland CA 91786
(714) 985-1901

Koni America, Inc.
111 W. Lovers Lane
Culpeper VA 22701
(703) 825-4543

Monroe Automotive Equipment Co.
1 International Drive
Monroe MI 48161
(313) 243-8000

WHEELS

Carrol Shelby
19021 S. Figueroa Street
Gardena CA 90200
(213) 538-3402
(800) 421-1269 (fitting hotline)

Center Line Tool Corp.
13521 Freeway Drive
Santa Fe Springs CA 90670
(213) 921-9637

Cragar Weld Wheels
19007 S. Reyes Avenue
Compton CA 90221
(213) 639-6211

Gotti/Euro-Rep
2872 Walnut, Ste. C
Tustin CA 92680
(714) 838-7021

Halibrand Engineering
396 Raleigh Avenue
El Cajon CA 92020
(619) 588-4087

Jongbloed Modular Wheels
1521 E. McFadden, Ste. G
Santa Ana CA 92705
(714) 547-3073

SELECTED PART NUMBERS

QUICK-RATIO STEERING (MANUAL)

All A-body	2267-640	16:1 worm-and-ball-nut assembly (3.5 turns lock-to-lock)
B- & E-body	2537-727	16:1 worm-and-ball-nut assembly
	P4007-612	20:1 worm-and-ball-nut assembly

A-BODY REAR SPRINGS (will not increase ride height)

Part Number	Spring Rate lb/inch	Roll Rate lb/inch
2539-314	120	87
2539-324	180	130
2835-327	230	167

A-BODY TORSION BARS

Length inches	Part Number	Diameter inches	Roll Rate lb/inch
35.8	2071-628 (Rt)	0.890	108
35.8	2071-629 (Lt)	0.890	108
35.8	N.P.N.*	1.000	150
35.8	N.P.N.	1.030	200
35.8	N.P.N.	1.060	216

ALL B-BODY TORSION BARS

Length inches	Part Number	Diameter inches	Roll Rate lb/inch
37.0	N.P.N.	0.920	137
37.0	N.P.N.	0.940	150
37.0	N.P.N.	1.000	192
37.0	N.P.N.	1.030	217

ALL B & E-BODY TORSION BARS

Length inches	Part Number	Diameter inches	Roll Rate lb/inch
41.0	P2948-783	1.060	227
41.0	P2948-784	1.110	263
41.0	P2948-785	1.114	304

ANTI-SWAY BARS **

Type	Part Number	Diameter inches	Deflection lb/degree
All A-body (Front)	2071-652 (1962-66)	0.820	72
All A-body (Front)	2462-838 (1967-70)	0.880	77
All A-body (Front)	2535-533 (1967-70)	0.940	84
All A-body (Front)	N.P.N. (1967-70)	1.000	100

Type	Part Number	Application	Deflection lb/degree
All A-body (Rear)	3845-856	1967 to 1976 ***	N/A
B- & E-body (Front)	2948-723	R/T Ralley Pack ****	N/A
B- & E-body (Front)	3815-551	Police Suspension	N/A
B- & E-body (Rear)	3845-019	Police Suspension	N/A
B- & E-body (Rear)	3402-049	R/T Ralley Pack ****	N/A
All F-body (Front)	4014-033	Heavy Duty	N/A
ALL F-body (Rear)	4014-907	Heavy Duty	N/A

ADDITIONAL COMPONENTS

Part Description	Part Number	Application
Sway-Bar Mounting	2585-559	A-body, Front, 1962-1970
Sway-Bar Mounting	3845-856	A-body, Front, 1967-1976
Steering Conversion	2883-977	A-body, Straddle bracket, 1962-1967
Steering Conversion	2535-789	A-body, Idler Arm, 1962-1967

(Note: A professional installation is required to use these parts on early A-bodies to maintain correct steering geometry.)

Disc Conversion	2836-184	Disc brake rotor with 4-1/2-inch bolt circle required to convert from 4-inch bolt pattern.
Axle Conversion	2513-750-M 2513-751-M	Axles to convert 8-3/4-inch rear axle to 4-1/2-inch bolt circle.
Brake Conversion	2534-173	10-inch brake drum requried for above axle conversion.
Adjustable Proportioning Valves	P3690-675	Kit Car proportioning valve
Rear Disc Brakes	P4007-305	8-3/4 rear; bracket for disc-brake calipers.
Rear Disc Brakes	2925-220	8-3/4 rear; right caliper; 4-piston; 1969 Valiant.
Rear Disc Brakes	2925-221	8-3/4 rear; left caliper; 4-piston; 1969 Valiant.
Rear Disc Brakes	2836-184	Use only rotor from this assembly.
Removable Rear Torison-Bar Anchor	P2535-995	Anchor (2 needed)
Rear T-Bar Anchor	P4007-149	Flange (2 needed)
Rear T-Bar Anchor	1671-446	Lock ring (2 needed)

MOOG SUSPENSION BUSHINGS

Part Number	Description	Vehicle
K-408	Upper Control-Arm Bushing	A-body
K-7030	Upper Control-Arm Bushing	B- & E-body
K-791	Lower Control-Arm Bushing	A-body
K-7040	H.P. Strut-Rod Bushing	A-body
K-7026	H.P. Strut-Rod Bushing	B- & E-body
K-793	H.P. Idler-Arm Kit	A-body to 1971
K-7037	H.P. Idler-Arm Kit	A-body to 1972
K-7057	H.P. Idler-Arm Kit	A-body to 1974
K-7038	H.P. Idler-Arm Kit	All B- & E-bodies
K-373	Torsion-Bar Seal Kit	All models

MONROE SHOCK ABSORBERS

Application	Part Number (Front)	% (Ft/Rr)	Part Number (Rear)	% (Ft/Rr)
All A-bodies to 1975	HP-4855	45/55	HP-4961	50/50
All E-bodies	HP-4855	45/55	HP-4960	50/50
All B-bodies to 1972	HP-4855	45/55	HP-4960	50/50
B-bodies '73 to '75	HP-4867	45/55	HP-4960	50/50

KONI SHOCK ABSORBERS

All A-bodies to 1969	80-1423SP3	80-1539SP3

FACTORY SPOILERS

Application	Part Number (Front)	Part Number (Rear)	Additional Info
Airfoil Type; All Bodies		3570-208	3579056—Dstr./Dem. Pkg.
		3570-208	3579057—Dart/Val. Pkg.
		3570-208	3579058—'70 Barr. Pkg.
		3570-208	3579059—Chall. Pkg.
		3570-208	3579668—'71 Barr. Pkg.
		3570-208	3579060—'70 R. Run. Pkg.
		3570-208	3579061—'70 Corr. Pkg.
		3570-208	3579062—'70 Charg. Pkg.
		3570-208	3579669—'71 R. Run. Pkg.
Daytona/Superbird		3571-137	Horizontal Stablizer (1)
		3471-138/9	Vertical Stablizer (2)
		3412-687	Cap Screws (2)
		3571-134/5	Reinforcing Plate (2)
		3571-136	Vertical Stablizer Insulator
		3412-784/5	Front Support
		3412-790/1	Rear Support
A-Body	P3690878		1971-1976
	P4120359		Universal
	P3690879		1973 to 1976 Duster
E-body		3770369	1970 Barracuda
		37570371	1970 Challanger

TRUNK-MOUNTED BATTERY KIT

Description	Part Number	Application
Trunk-Mounted Battery Kit	P3690934	All Cars

*"N.P.N." means no part number has been assigned to this part; N.P.N. torsion bars are custom made by Martin Automotive Design.

**Heavier bars are available from ADDCO and Martin Automotive Design.

***Special suspension component—can be adapted to earlier A-bodies.

****Anti-sway bar used on AAR Barracuda and Challenger R/T.

NOMENCLATURE AND BIBLIOGRAPHY

NOMENCLATURE

A-Body: Chassis used on 1964 to 1976 Valiants and Darts; 1964 to 1969 Barracudas.

B-Body: Chassis used on all Coronets, Satellites, Road Runners, Super Bees, Chargers, GTXs, R/Ts, Belvederes, and Cordobas to 1979; 1976 to 1978 Furys; 1977 to 1978 Monacos; 1978 to 1979 Dodge Magnums.

E-Body: Chassis used on 1970 to 1974 Barracudas; all North-American Challengers.

F-Body: Chassis used on 1976 to 1979 Volares, Aspens, Road Runners, and R/Ts.

L-Body: Chassis used on all Onmis, Horizons, 024s, and TC3s.

M-Body: Chassis used on 1977 to 1979 Diplomats and LeBarons.

R-Body: Chassis used on 1979 New Yorkers and St. Regis.

A-Engine: 273cid, 318cid (1967 & later only), 340cid, 360cid engines.

B-Engine (or LB-Engine): 361cid, 383cid, 400cid engines.

RB-Engine: 413cid, 436cid wedge, 440cid engines.

Hemi: 426cid hemi-head engine.

I-Engine: 170cid, 198cid, and 225cid 6-cylinder engines.

Banjo Rear End: 8-3/4-inch rear-end assembly.

Dana Rear End: 9-3/4-inch rear-end assembly.

Spicer Rear End: 7-1/4-inch, 8-1/4-inch, 9-1/4-inch rear-end assemblies.

A-833: Chrysler New-Process 4-speed manual transmission—1964 to 1976—with either cast-iron or aluminum housings.

A-727: Chrysler Torqueflite automatic transmission used with the 340cid, 360cid, 383cid, 400cid, 440cid engines, and the 1962-76 426cid hemi-head engine.

A-904, A-998, and A-999: Chrysler Torqueflite automatic transmission used in all 6-cylinder cars, and on the 273cid, 318cid, and 360cid engines.

A-745, A-250, A-230, and A-903: Chrysler 3-speed manual transmissions.

BIBLIOGRAPHY

Baumeister, Avallone, *Mark's Standard Handbook for Mechanical Engineers*, 8th ed., McGraw-Hill, San Francisco, 1979.

Campbell, Colin, *The Sports Car*, Robert Bentley, Inc., Cambridge, 1962.

Costin and Phipps, *Racing and Sports Car Chassis Design*, Robert Bentley, Inc., Cambridge, 1963.

Halliday, David and Robert Resnick, *Physics I & II*, John Wiley & Sons, New York, 1966.

Harvey, Warren L. and D. B. Ressler, Jr., *Competition Set-Up, A Chassis Adjustment Analysis*, P.M.E. Publications, Morristown, 1973.

Lent, Deane, *Analysis and Design of Mechanisms*, Prentice-Hall, Englewood Cliffs, 1961.

Rathgeb, Larry, selected technical papers, The Chrysler Corp., Detroit, 1974 to 1977.

Smith, Steve, *Advanced Race-Car Suspension Development*, Steve Smith Autosports, Santa Ana, 1974.

Spotts, M. F., *Design of Machine Elements*, Prentice-Hall, Englewood Cliffs, 1971.

Titus, Jerry, "The Dynamics of Motorcar Handling," *Sports Car Graphic*, 1969.

Watkins, Martyn B., *Suspension and Brakes*, Speed and Sports Publications, Acton, London, 1970.

"What Makes Cars Handle," *Selected Technical Papers*, edited by Jim Hall with David E. Davis, Jr., Ziff-Davis Publishing, New York, 1975.

Wheeler, Gary, "Ponycars: How Trans-Am Teams Make Them Handle," Car Life Magazine, November 1968.

CHASSIS SETUP SUGGESTIONS

This information is provided to help the reader establish the basic requirements for building a well-handling car for street, autocrossing, or competition use. The data provided may require adjustment for optimum performance, and the reader is encouraged to carefully review the basic and theoretical material presented elsewhere in this book before attempting any modifications.

VARIABLE	STREET/GT	AUTOCROSSING	COMPETITION
Front Sprint Rate *	150-lb/in	200- to 250-lb/in (min)	304-lb/in
Rear Spring Rate *	110-lb/in	130-lb/in	130-lb/in
Front Anti-sway Bar	150-lb/in	250- to 290-lb/in	450-lb/in
Rear Anti-sway Bar	Optional	90-lb/in	Optional
Front Shocks	55/45%	54/45%	55/45%
Rear Shocks	50/50%	50/50%	50/50%
Front Rims	14 x 6-inch	14/15 x 8-inch (min)	15 x 8-inch (min)
Rear Rims	14 x 6-inch	14/15 x 8-inch (min)	15 x 10-inch
Wheel Bolt Pattern	4-inch (4-1/2-inch opt.)	4-1/2-inch	4-1/2-inch (5-inch opt.)
Front Tires	ER70HR14	HR60 or HR50	Slicks
Rear Tires	ER70HR14	HR60 or HR50	Slicks
Front Brakes	Large Drum or Disc	Disc	Large Disk
Rear Brakes	10-inch Drum	11-inch Drum	11-inch Drum or Disc
Steering Ratio	20:1	16:1	16:1
Caster	3-degrees Positive	4- to 5-degrees Positive	5-degrees Positive
Camber	1-degree Negative	1-degree Negative	1-degree Negative
Toe Setting	3/32-inch In	3/32-inch Out	3/32-inch Out
Differential Ratio	3.23- to 3.55:1	3.55:1	3.90- to 4.88:1
Center of Gravity Height	16-inches	12- to 15-inches	11- to 14-inches
Front Roll Center	7-inches	5-inches	3- to 4-inches
Rear Roll Center	11-inches	8-inches	7- to 8-inches
Oil Pan	Stock With Baffles	Stock With Baffles	Modified With Baffles

* Spring Roll Rate

ABOUT THE AUTHOR

It's a fact: Suspension systems don't have the impressive appeal of dual quads, cross-ram manifolds, superchargers, and the like. So the average enthusiast spends most of his time polishing the "showy" goodies and devoting much less effort—perhaps none—to the dusty undercarriage. But some individuals are driven to fine-tune, modify, and perfect all aspects of performance equipment, including the often misunderstood suspension system. Mike Martin is one of those individuals.

A California native, Mike Martin has been a hardcore Mopar fan since his early high-school days in 1964. In the late '60s, Mike began his racing career by drag racing. He eventually moved on to SCCA Solo-I and then to SCCA amateur road racing and autocrossing. But Mike was never satisfied with just racing. He had a burning desire to build faster and better-handling cars not just by *buying* what was required, but by developing the parts that would allow the average racer to build "a well-handling car at a reasonable price."

After completing his education at Cal Poly University in business administration and mechanical engineering, Mike started his own business to perform further research into the science of suspension development. Today, in Rodeo, California, Martin Automotive Design continues to grow and produce innovative street and racing components for Mopars.

So on any given day, you can find Mike either at his drawing table, steeped in thought—or at his work bench, assembling his latest creation; but occasionally, you can still find Mike at the race track behind the wheel of his meticulously detailed Barracuda, "putting his ideas on the pavement."

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