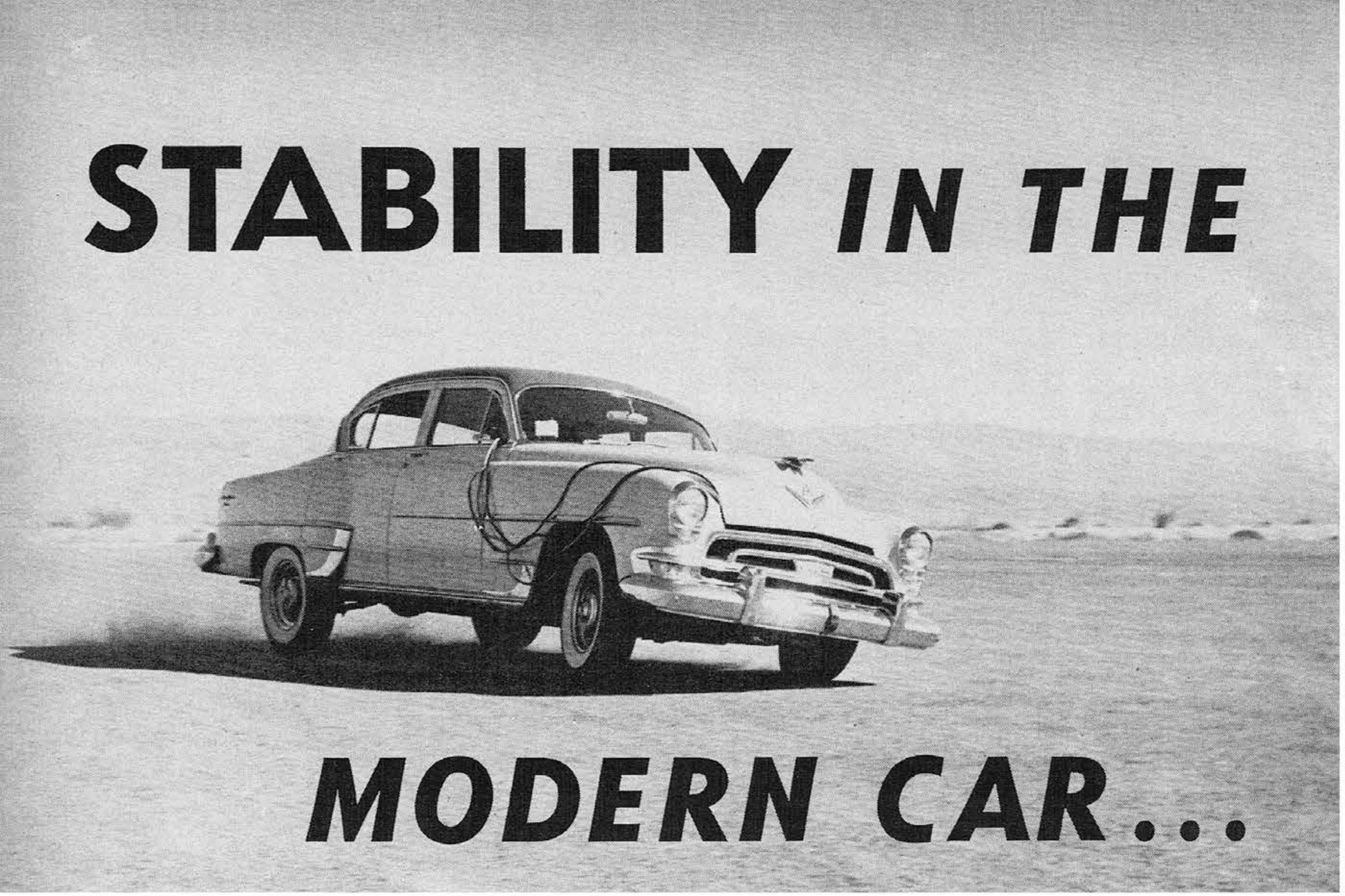
STABILITY IN THE MODERN CAR...



O. D. DILLMAN AND E. J. COLLIER, CHRYSLER CORP.

Excerpts from paper, "Building Stability into the Modern Automobile," presented at a meeting of the SAE Atlanta Group, Atlanta, Ga., Jan. 19, 1953.

MODERN demands for comfort and safety are largely responsible for the increased emphasis being placed on passenger-car stability. If this stability is to be attained under all driving conditions, many factors must be considered in the design of the car, such as:

Oversteering and understeering, tire slip angle, front suspension, rear suspension, steering linkage, power steering, and weight distribution.

OVERSTEERING AND UNDERSTEERING

The condition that demands the most exacting requirements of a modern suspension is simply that of negotiating a sharp curve at high speed.

One of two conditions exists as a driver endeavors to keep his car on the road and follow the road dividing strip. The car will tend either to oversteer or to understeer, depending upon the optimum design conditions that have been built into it; this, because the car in itself has the features to produce one of these two conditions, aside from the amount of turn that the driver applies to the steering wheel.

Fig. 1 illustrates how these features feel to the driver as he compensates for the behavior of his 4-wheeled vehicle. Shown in the shaded area is a deviation path that would have to be compensated for to overcome the oversteering or understeering caused by the components of the car suspension.

SLIP ANGLE

As tires have been built to operate at lower pressures, the tire area in contact with the ground has increased. This section is known as the "patch area," and here lies the secret of tire aligning torque. (See Fig. 2.)

If this low-pressure tire is applied to the car going around the curve in Fig. 1, a condition similar to that illustrated on the right exists. The suspension engineer says that the tire tends to slip. While this does not really mean that the tire scrubs across the ground, it does deflect away from the center position on the wheel rim. This deflection is produced as the result of a centrifugal force exerted on the axles by the mass of the vehicle.

If a given point is traced as it travels around the surface of the tire, a strange thing is found to happen to the line of motion, in that the car is no longer going in the direction that the front wheels are pointing. The change in direction of travel is called the "slip angle" of the tire. This slip angle is a function of the patch area and tire deflection caused by car speed and load.

The slip angle of the front wheels will tend to produce understeering in the car going around the bend. By the same token, a car heavily loaded in the rear end will create more slip angle in the rear tires than in the front ones, and the car tends to oversteer.

ROLL CENTER OF THE FRONT SUSPENSION

The three major links that produce the geometry of a front suspension are the upper and lower control arms connected at their outer ends by the steering knuckle support.

By correctly designing the placement of the three major components in this linkage, the tire and wheel can be made to travel from jounce to rebound following any desired path. There are two distinctly different wheel gyrations to which the engineer lately has been giving considerable importance.

As shown in Fig. 3, point (A) is the camber change of the centerline of the tire as the wheel travels up and down, and point (B) is called the tire scuff.

As the suspension goes from jounce to rebound, a projected point of the tire contacting the road will travel horizontally along this heavy black line. Unlike tire slip, this point does scrape across the road, or would if the tire were not rotating. The two important points about scuff are the damping action it applies to a jouncing car and its effect on determining the roll center of the suspension.

If we draw an arc through points 1, 2, and 3, the point at which the radius of this arc intersects the vertical centerline of the front crossmember, when the suspension is in the normal position, locates the height of the roll center. Since this point of rotation only indicates the behavior of the front of the car, we must look further to determine what effect this roll center has on the entire car.

ROLL AXIS

Fig. 4 locates the roll center of the rear suspension at a point below the differential carrier centerline. In this case, the method of attaching the rear axle to the frame is the governing factor of roll center. This is, of course, the springs which are attached to the axle housing spring seats.

Because the car body and frame are supported by the front and rear suspensions, the roll centers of each determine the roll axis of the entire car. That is the line connecting point A to point B in Fig. 4.

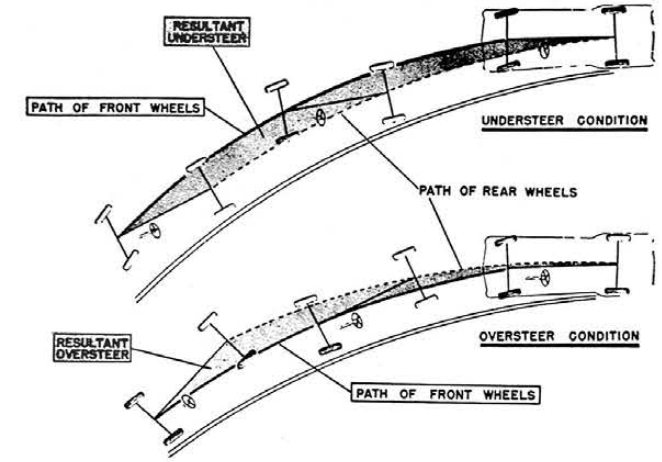
Centrifugal force applied to the car body as the car goes around a curve will react at the center of the car mass or at the center of gravity. Depending on the length of the dark line, running from the c.g. to the intersection of the roll axis, the car will have a large or a small overturning couple. Thus, by raising the position of the front roll center, the overturning couple can be decreased and the amount of car body roll reduced.

By designing a negative camber in jounce and slight positive camber in rebound, the following condition is produced when our car starts to roll into a turn. Fig. 5 illustrates the front view of two cars taking a curve at high speed. The car at the right has camber characteristics similar to those just mentioned. When the right wheel (left in picture) starts into a jounce it takes on negative camber. At the same time, the left wheel starts into a rebound and produces some positive camber. The result is a very desirable "biting" of the wheels into the curve, thus improving stability.

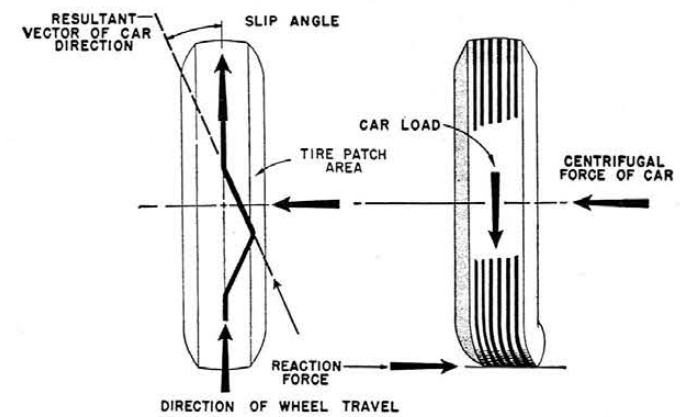
Shown in the left-hand picture is a conventional type of front suspension, which allows both wheels to take on a negative camber. Possible disadvantages of this tire heeling are excessive tire squealing, increased car roll, and again the ever-present tire slip angle contributing to understeering.

SWAY BAR

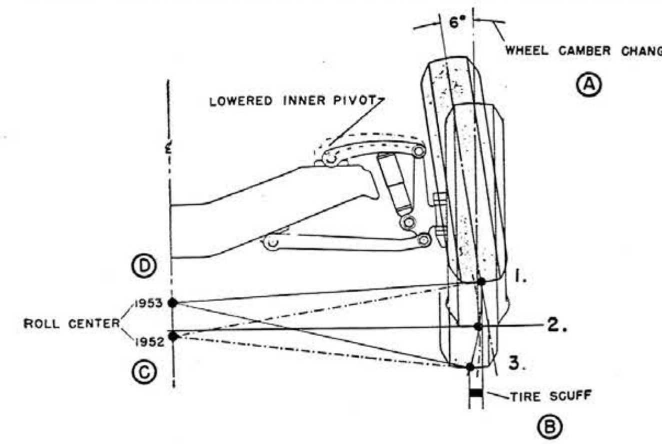
Fig. 5 also shows the sway eliminator, which connects the two front suspension lower control arms to the car frame. This anti-rolling device comes into play only when there is a differential motion between the left and right lower control arms. In substance, this sway controller is nothing more than a bar of spring steel ¾ in. in diameter. It is held at either end by a rubber bushing and supported at two points on the frame by additional rubber mountings. As each side of the suspension goes into the jounce or rebound, one at the same rate as the other, the sway bar merely goes along for the ride, with no work being done. Now, if the car starts to roll, as shown in the left-hand picture, the right wheel starts into jounce and the left wheel approaches the rebound condition. A differential action between the two lower control arms is set up and the sway bar is twisted, as shown. You almost get something for nothing in this maneuver, in that the car is literally lifting itself by its boot straps. Unfortunately, the two sides of the front suspension do not always act as a ballet team on mildly rough road travel and the result is continued differ-



This graph shows the path of the automobile front wheels, and the resultant understeering and oversteering in turns

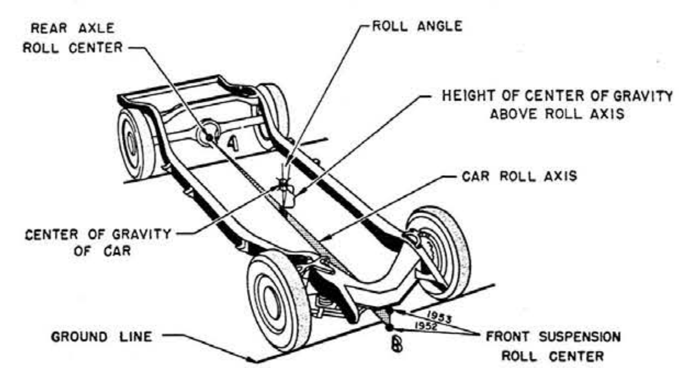


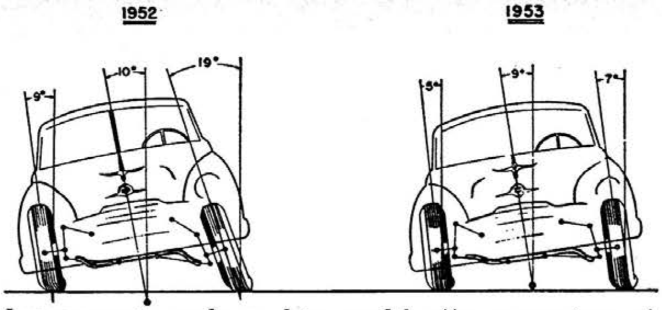
Forces that contribute to tire slip angle are illustrated



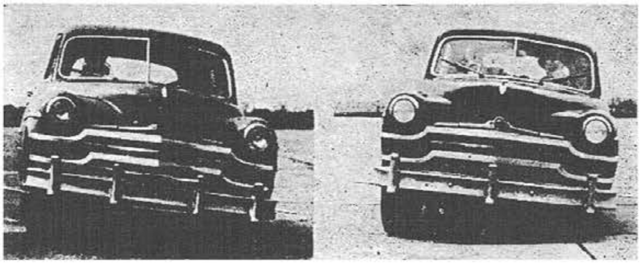
Sketch of chassis shows that the front and rear suspension roll centers determine the roll axis of entire automobile

Comparison of '52 and '53 front suspension roll centers

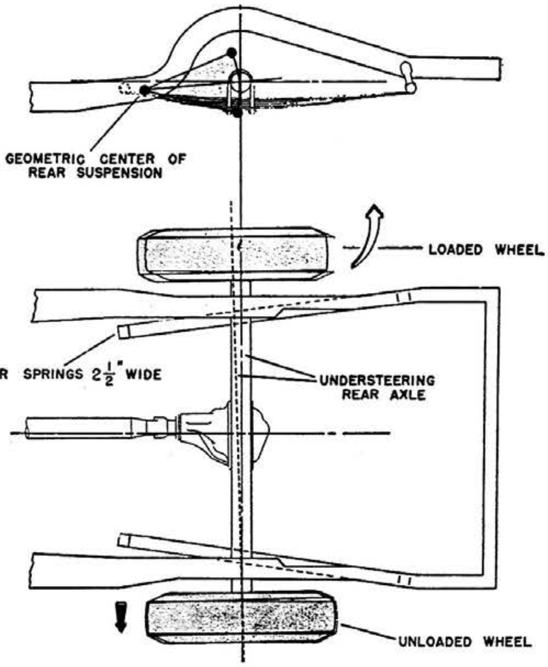




Improvements made on later model offer comparison of this car's wheel camber on curves and sway bar action

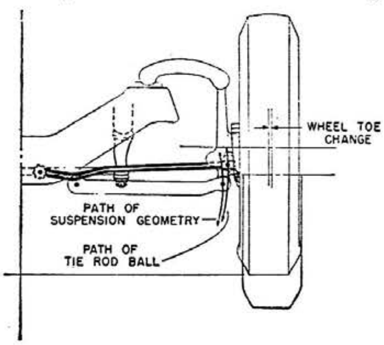


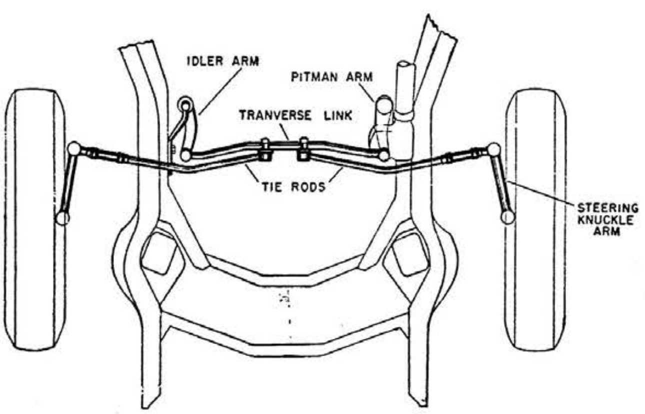
Photographs show the roll center effect on two models. Left: low roll center car; to the right: high roll center car



This is steering geometry of Hotchkiss rear suspension

Shown is '53 Dodge Red Ram symmetrical idler arm steering





ential action. This ride condition governs the diameter to which the sway bar must be held in order not to create a harshness under high-speed highway traveling.

Two test cars (Fig. 6) were used to substantiate much of the material covered. These cars were run under identical testing conditions, with only a change in suspension geometry.

The car at the left has a roll center 1.52 in. below the ground; the wheel camber change is 9½ deg on the right and 9 deg on the left wheel. The resulting car body roll angle is 7 deg. The experimental suspension on the right-hand car has a roll center 3.89 in. above the ground, camber angles of 5½ and 6¾ deg, and an all-important body roll reduction of 21%.

REAR SUSPENSION

Fig. 7 indicates the make-up of the geometry used in a Hotchkiss rear suspension. The upper view illustrates the geometric center of the entire left side of a rear suspension. This true center is not at the anchor point of the front spring eye, but slightly to the rear and up from this point. To determine what the center of the rear axle, and thus the center of the rear wheels, will do under spring action, notice the shaded area of this arc. As the wheels go from jounce to rebound, a short, quick arc is described by the center of the wheel. The point of major interest is that the wheel center is moving closer to the front of the car in both of these extreme conditions. In effect, the wheelbase of the car is reduced. In the top or plan view of the same rear suspension, we have simulated the position of the rear wheels as the car in Fig. 1 travels around the bend. The roll of the car being toward the outside, the right wheel is forced into an upward position while the left rear wheel is starting downward, away from the car frame. The result is two projected points from the top view that cause the centerline of the rear wheels to askew diagonally across the car body. In this case the car is helped to understeer.

The use of splayed rear springs in this car has allowed two important improvements in chassis design; first, the use of lower deep frame side rails, and an ideal location of the front spring eye, to compensate for incorrect rear end steering. A change in the front spring eye will either induce oversteering or understeering in the rear axle.

The extra wide 2½-in. rear springs have only five leaves as compared to the conventional type, which has nine or more. The main reason for this spring is to reduce the number of contacting surfaces and thus reduce static friction, which hinders a smooth ride. This spring has special, wax-impregnated cloth interliners, which help to reduce friction. In addition to reduced friction, the 2½-in. width allows a wider spring shackle and front eye bushing, which increases car stability. Recent testing of the 1952 1¾-in. wide spring versus the 1953 2½-in. rear spring showed a reduction in car body roll of about 10%.

STEERING

The most prominent steering linkages used on American passenger cars are: the long and short tie-rod type, the center arm type, and the transverse link or symmetrical idler arm.

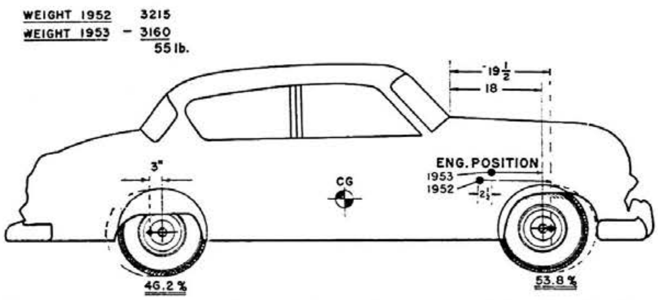
The long and short tie-rod type is the most elementary in design, consisting of one short tie-rod connecting the steering gear pitman arm to the left steering knuckle arm, and a long tie-rod joining the pitman arm to the right steering arm.

The center arm type employs a fore-and-aft drag link that actuates a center arm, usually located in the center of the front crossmember. This arm controls the tie-rods. Again two rods are used but this time they are of equal length. This linkage produces a symmetrical design of steering geometry.

The most popular type of steering linkage in use today is the transverse link or symmetrical idler arm. Fig. 8 indicates the major components of the 1953 Dodge Red Ram steering. Notice that the pitman arm and idler arm carry the transverse link in a parallelogram action. The two tie-rods are of equal length and connect the transverse link to the steering knuckle arms.

These three types of steering are only a few of the many modifications that have grown to make up the correct steering linkage to fit the design requirements of a chassis. To be sure, the economics of a design is often as important as the space requirements dictated by the chassis layout.

In the upper view of Fig. 8, you are looking into the front of a chassis equipped with idler arm steering. For the sake of clarity the arc that the suspension describes, as it goes into rebound, has been slightly displaced from the arc of the left tie-rod end. These arcs are not equal and would cross each other if they were continued far enough. But since the full wheel travel downward is only 4½ in., the remaining segment is not of interest. The mismatch in the two arcs will



Important to suspension engineer is mass weight distribution

result in the car wheel being turned by the suspension alone, as it travels. This differential motion will rotate the wheel (shown in cross-hatching) outward and cause a toe-out condition.

How then does the designer compensate for any adverse wheel toe changes? One way in which this problem can first be attacked over the drawing board is by using a slightly different arrangement of the symmetrical idler arm. First, the basic arcs of the suspension are laid out, and they result in the development of one arc. By using a considerable amount of know-how, the designer selects an inner tie-rod ball end location on the transverse link. Next, the position and length of the steering knuckle arm ball end are determined. After a wheel Ackerman check, the new tie-rod length is traced through the full travel of the suspension geometry. The results, if satisfactory, are points similar to those shown along side of the suspension arc. These points represent the tie-rod end and must be multiplied by a factor, 8 for instance, to indicate the wheel toe change. In the event that the first location is not satisfactory, the inner tie-rod end can be moved to a new location, or the arc position of the steering arm is changed by raising or lowering the location of the tie-rod ball end. This last consideration should indicate the versatility of being able to move the inner rod end to the ideal point along the transverse link.

Now let's go from the drawing board to a car installation of this idler arm steering to see how accurate the design prediction of this layout was. Fig. 9 shows the full jounce to rebound of this linkage in a car in the straight-ahead position and for 20-deg wheel turns to the right and left. The point of interest is that, at any normal driving condition, our differen-

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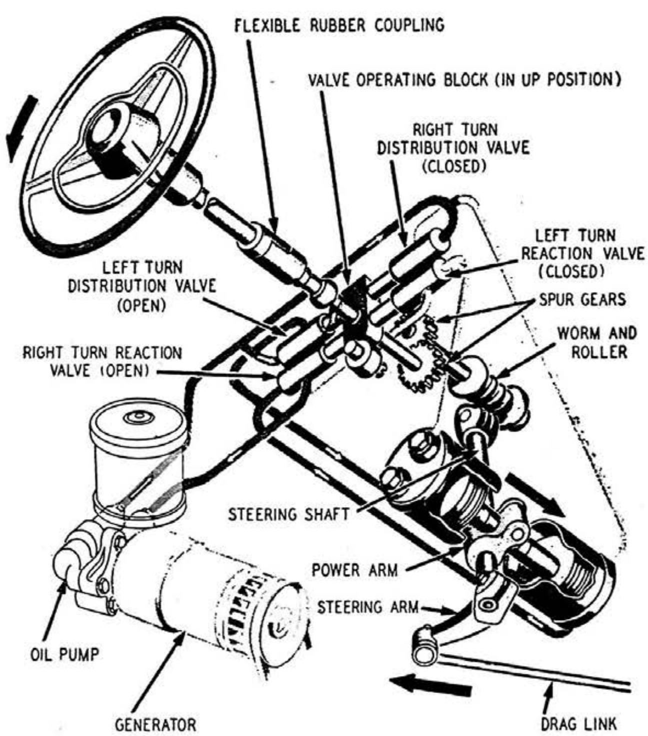
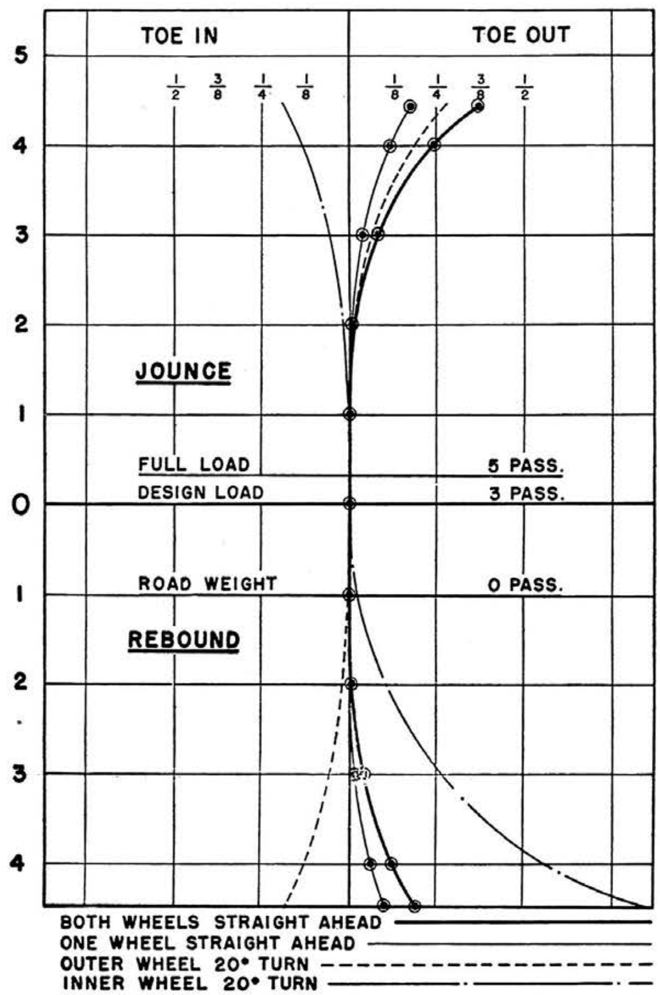


Here's how power steering operates during a left turn



Full jounce to rebound of linkage in car in the straight-ahead position and for wheel turns to the right and left

STABILITY IN THE MODERN CAR

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tial action is nil and this car should have little or no wheel fight in addition to even tire wear.

This section was included because wheel fight contributes heavily to driver reaction. Also, jounce and rebound suspension-steering influence car front end oversteering and understeering.

POWER STEERING

Power steering virtually eliminates all road shock from reaching the driver and stabilizes car steering during all operating conditions. Shown in Fig. 11 are the functioning parts of Chrysler power steering. As the driver turns the steering wheel, the steering column spur gear climbs up or down with respect to the worm shaft axis. This causes an up or down motion of the valve operating block. While the movement of the block is of very small magnitude, it is sufficient to open and close the distribution and reaction valves in the desired combination. This produces a difference in pressure between the power cylinders so that the pistons move the power arm in the same direction as the worm-and-roller is driving the steering gear shaft.

In regard to the problem of car stability, how does the power steering unit work as the front wheels of a car drop into a chuck hole or encounter other road obstacles? In a reverse action, the steering linkage is started in motion by a turning of the car wheel. As the driver is holding the steering wheel, the spur gear mounted at the end of the steering gear shaft tries to climb upon the gear at the end of the steering column. As a result, the valve operating block moves up or down and the valves are set in operation, causing the oil pressure in the power cylinders to oppose the movement of the car wheels. Thus, hydraulic pressure is applied as a counteracting force to dampen the shock that was previously acting on the steering linkage and greater steering and car stability is achieved.

BALANCED WEIGHT DISTRIBUTION

The final phase of designed stability in the modern passenger car is balanced weight distribution. The evaluation of mass distribution has gone from horseless carriage to the science of balance now employed in our present-day cars. Automobiles of the early 1900's may have had as much as 70% of the weight on the rear wheels. This 30% front wheel loading would make our modern car oversteer dangerously when going around curves. This is one fundamental objection on the part of some automotive engineers regarding rear-engine cars, as this type of car is usually loaded more at the rear than at the front.

By careful placement of the modern passenger-car chassis and body components, a 50-50 weight distribution is usually obtained. However, whenever possible, the balance of loading is generally shifted to 55% on the front wheels, since an understeering car is much easier to manage. The passenger load is applied primarily to the rear wheels. With a curb weight distribution of 45-55%, the 50-50 loading is more nearly approached when the car has a full passenger load.

Of equal importance to the suspension engineer is the mass distribution in the vehicle. The four most important suspension frequencies of a passenger car are: jounce, roll, pitch, and yaw. The latter two—pitch and yaw—are dependent on the moment of inertia of the vehicle at an axis perpendicular to the wheelbase passing through the center of the car, as shown in Fig. 11.

The mass of the car could be split in two sections and placed several feet off the front and rear bumpers and, if properly split, the basic 47-53% wheel weight distribution would remain the same. But the moment of inertia would be changed considerably over this car shown. The car weight has only been decreased 55 lbs. from 1952 to 1953 in the Plymouth. However, the wheelbase has been shortened 4½ in. and the distance between the centerline of the front wheels and the engine—a big factor in mass distribution—has been decreased 2½ in.

Briefly, balanced weight distribution means this. The jounce rate of the front and rear suspension has remained unchanged. This tends to retain car spring stability. The pitch and yaw car frequencies depend on the moment of inertia, as well as being a function of the square of the distance from the center of gravity to the centers of the front and rear springs. While the moment of inertia of some of our 1953 cars has been relatively unchanged, the wheelbase has been shortened, thus reducing the distance from the c.g. to the centers of the front and rear springs. As a result, the pitch and yaw frequencies have been reduced in these cars. To the engineer this means a better "isolation ratio"; to the driver it means a wider range of driving conditions with less road disturbance.

WHAT IS IT?
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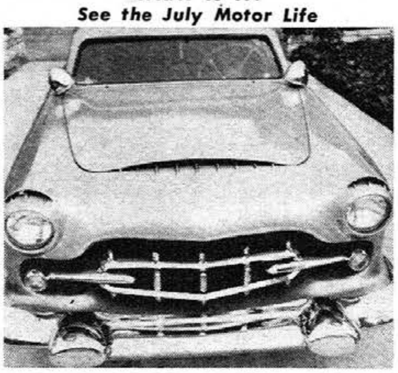

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