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(Without Powerplant)

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EERING GEARS — BRAKES — UNIVERSAL JOINTS
DIFFERENTIAL GEARS — MISCELLANEOUS PARTS

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Formerly Engineering Editor, Automotive Industries

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PREFACE TO THE THIRD EDITION

For this edition the text has been completely revised. New material will be found particularly in the chapters on Frames and Their Brackets, Brakes, and Power Brakes and Auxiliary Energy Dissipators. The introductory chapter, now headed "Chassis Types and Layouts," has been extended to cover more types of automotive vehicle than previously, and dimensional data in it have been brought up to date. The second chapter, "Power Required for Propulsion and Acceleration," has been largely rewritten to bring it more into line with recent experimental results.

Within the field covered by this volume, developments in recent years seem to have been most intense in connection with brakes. The high speeds of modern highway travel and the high density of traffic make powerful, safe, and reliable brakes absolutely essential. Besides, it is highly desirable, especially in the case of vehicles used chiefly in city streets with their traffic lights, that only little physical effort be required to apply the brakes. Improvements in brake design therefore have aimed to provide greater braking power with less pedal pressure, greater reliability under severe operating conditions —as in descending long, steep grades—and more economical use of lining material.

Minor changes and additions have been made in other chapters. This edition contains more than sixty new illustrations and a corresponding amount of new text; and it is larger than the preceding one, even though a considerable amount of more or less obsolete material has been eliminated.

THE AUTHOR.

FROM THE PREFACE TO THE FIRST EDITION

This book completes my series on automotive engineering subjects, which includes three other volumes. Like the others, it is intended both as a handbook or reference book for engineers and as a textbook for use in engineering courses.

The book deals with the various parts of the automotive chassis, with the exception of the powerplant. It discusses the functions of these parts, their theory, the materials of which they are made, and their design. Most of the chapters also contain brief notes on the origin and the historical development of the parts with which they deal. Shortcomings of earlier designs or types are pointed out, and the reasons for their abandonment are made clear. Production processes are not covered, except in a few instances where special methods of recent origin are applied and yield superior results.

Rules for the proportioning of parts are given in almost every chapter, and these, together with the numerous illustrations of stock parts, should prove helpful and stimulating to the designer. Recommended stresses in general are on the conservative side, provided the materials used conform to standard specifications and are suitably heat-treated.

In quite a few of the chapters design formulae or equations expressing the operating characteristics of the parts dealt with are developed from basic principles. Most of these equations have been available for a good many years, and the engineer who wants to make practical use of any one of them only needs to insert his values and perform a few simple arithmetical operations. He does not need to go back to the fundamentals. The reason for including this sort of material is its educational value. Designers often are faced with problems for which no applicable formulae are to be found in either the engineering handbooks or in current technical literature, so that an original solution is called for. In that case familiarity with methods used in the past in the solution of more or less similar problems will be of help.

I wish to express my indebtedness to the publishers of *Automotive Industries* and *The Automobile Engineer* and to the Society of Automotive Engineers for permission to reproduce illustrations which first appeared in their respective publications.

THE AUTHOR.

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CHAPTER I

Chassis Types and Layouts

The conventional motor vehicle has four road wheels and is steered by its front and propelled by its rear wheels. Motor vehicles, however, have been built with from three to eight wheels. The three-wheeled construction, which is now practically obsolete, has been used only for light and cheap vehicles, while the six- and eight-wheeled constructions are used for heavy buses and trucks. The eight-wheeled type is quite rare. As compared with the three-wheeled vehicle, the four-wheeled has the advantages of greater stability and greater comfort, while over the six- and eight-wheeled it has the advantage of greater simplicity. The stability of a vehicle is measured by the angle to which it must be tilted in order to bring its center of gravity directly above a line connecting center points of tire-contact surfaces. If the centers of gravity are assumed to be in the longitudinal vertical centerplane of the vehicles and at the same distance above the road surface, and if the treads are the same, this angle evidently is smaller in a three-wheeled vehicle than in one with four or more wheels.

Steering and Driving—Four-wheeled vehicles can be steered by either the front, the rear, or all four wheels, and they also can be driven by either the front, the rear, or all four wheels. Two-wheel steering and two-wheel drive are much simpler mechanically than four-wheel steering and four-wheel drive, respectively, and the latter are used only when the advantages they offer are of sufficient practical importance. Four-wheel has the advantage over two-wheel steering that it permits of turning in a much smaller circle. This is important in the case of vehicles which must be maneuvered in narrow factory aisles or other restricted spaces, and four-wheel steering therefore is used in some industrial trucks. Where the vehicle is to be steered by two wheels only, there are two considerations which favor the choice of the front wheels for the purpose. In the first place, when a rear-steering vehicle is parked close to a curb, it can be driven away only by backing, because to get it away from the curb the steering wheels must be swung in such a direction that on forward motion they would run into the curb. With front-wheel steering under the same conditions, the steering wheels are swung in the opposite direction, and on forward motion all wheels move away from the curb together. Secondly, for reasons to be explained further on, it is generally preferred to drive the car by the rear wheels, and it is mechanically much simpler to steer by the non-driving than by the driving wheels.

Certain industrial trucks of the lift type have front-wheel drive together with rear-wheel steering. On these trucks the load is carried on a lifting carriage or fork ahead of the front axle, and a large proportion of the total weight therefore comes on the front wheels, which makes front-wheel drive preferable.

Four-Wheel Drive—Driving by all four instead of by two wheels has the advantage that it affords greater adherence between the driving wheels and the road surface. This results in greater limiting tractive force or propelling effort, which latter is directly proportional to the aggregate weight on the driving wheels. With four-wheel drive in a four-wheeled vehicle, all of the weight is available for traction purposes, but with rubber tires on improved roads so much traction is never—or at least very rarely—needed, and the greater simplicity and the slightly higher efficiency of the two-wheel drive then decide in its favor. On the other hand, where vehicles must be operated away from the "beaten paths," such as military vehicles, vehicles used in connection with mining and road-building operations in undeveloped country, etc., it is necessary to provide as much traction as possible, and four-wheel drive, or all-wheel drive, is commonly used. The four-wheel-drive vehicle naturally is considerably more expensive than an otherwise equivalent one which is driven on two wheels only, because in addition to the propelling mechanism of the latter, it requires a transfer gear, an additional propeller shaft, and a front axle which, instead of being a plain steering axle, is a combined steering and driving axle.

It must not be assumed that by driving on all four wheels the maximum traction can be doubled. Most heavy four-wheel trucks and buses are fitted with single tires on the front and dual tires *of the same size* on the rear wheels (so that only a single spare tire needs to be carried); hence, as far as tire loading is concerned, the weight distribution is optimum if the rear wheels normally carry twice as much as the front wheels. Maximum traction is required when stiff grades are being ascended under load, under which condition load is being transferred from the front to the rear wheels both by the inclination of the supporting surface and by the reaction to the torque on the driving wheels, which latter tends to raise the front end of the vehicle and to press the rear wheels harder against the road surface. Under extreme conditions, from 10 to 15 per cent of the total load must remain on the front wheels, because otherwise the steering is not sufficiently positive. Under such conditions, then, four-wheel drive increases the maximum available traction by only from 11 to 17.5 per cent. These figures, however, must not be misinterpreted. Whereas with a four-wheel-driven four-wheel vehicle 100 per cent of the total weight is *always* available for traction purposes, we have no assurance that with a two-wheel-driven vehicle, when poor traction conditions are encountered, from 85 to 90 per cent of the total weight will be on the driving wheels. In many cases the proportion will be less, and what might be called the normal gain in traction due to four-wheel drive is greater than the figures given would indicate.

The extensive use of trucks with more than one driving axle by the U. S. Army has led to a new set of type designations. Thus a 4 x 2 truck is a four-wheel truck with two-wheel drive, and a 4 x 4 truck a four-wheel truck with four-wheel drive. There are also 6 x 2, 6 x 4, and 6 x 6 trucks, that is, six-wheelers with drive on two, four, and six wheels, respectively.

Powerplant Location—Before discussing the relative advantages of front and rear drives it will be well to discuss the subject of powerplant location. In most of the first four-wheeled vehicles propelled by combustion engines, the powerplant was located under the seat and perhaps a rear deck. The pioneer designers naturally used horse-drawn carriages for their models, and the seat box seemed to be the most convenient place for the installation of the engine and transmission. During that early period the dos-à-dos type of body was most popular with designers, because with two seats back to back there was more room for the powerplant than under a single seat. It was soon found, however, that this location of the powerplant had serious disadvantages, for every time some slight adjustment had to be made to the mechanism, or the engine had to be serviced, the passengers had to dismount, and in the pioneer vehicles the need for "tinkering" arose only too frequently. Emile Levassor of the firm of Panhard & Levassor in France then conceived the idea of placing the engine on the front of the chassis frame, under a hinged hood or bonnet, and driving to the rear wheels through a change-speed gear located amidships. In the first rear-driven car with front-mounted engine (Fig. 1) the final drive was by side chains, but this was later replaced by shaft drive to a live axle. This general layout of the chassis, with the engine arranged longitudinally in front, with the clutch and transmission immediately in back of it, and with shaft drive to a live rear axle, soon became the standard, not only for private passenger cars, but also for buses and trucks. For passenger cars and trucks it is still used almost exclusively, while for buses it has been discarded, because a large engine mounted longitudinally on the chassis frame in front cuts down the floor space, and correspondingly reduces the passenger capacity. In modern buses, therefore, the powerplant is located

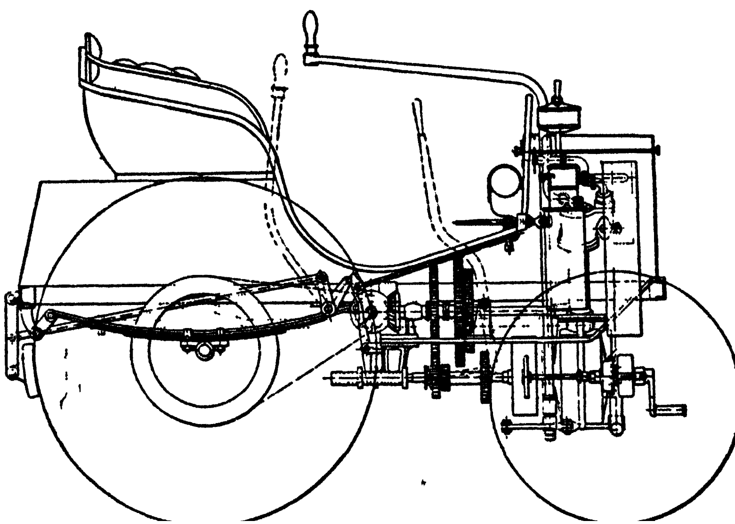


FIG 1—PANHARD-LEVASSOR 1894 CAR, PROTOTYPE OF PASSENGER CARS WITH FRONT-MOUNTED ENGINES

either transversely at the rear, under a seat extending the full width of the body, or else underneath the chassis frame amidships, so that the whole length of the frame is available for seating passengers.

Placing the powerplant at the front of the chassis under a hood not only makes it most accessible, but also offers a number of other advantages. The radiator of the cooling system preferably should be close to the engine, because long pipe connections slow down the circulation and are a potential source of trouble. With a front-mounted engine it can be located directly in front of same, where it receives the full benefit of the blast created by the motion of the car. In such a vehicle the propeller shaft connecting the transmission to the rear axle will be relatively long, which reduces the angularity of the universal joints and the loads on their bearings.

One objection that might be raised to the present conventional passenger-car chassis layout is that the powerplant occupies too much room on the frame, making the passenger compartments rather cramped, if the wheelbase is of only moderate length, or calling for a very long wheelbase if ample room is provided for the passengers. The weight of the chassis naturally increases rather rapidly with the wheelbase. That the powerplant occupies so much chassis space is due, however, to the fact that multi-cylindered, high-displacement engines are now being used, and cannot be blamed on the location of the engine. If engines of similar type and displacement were to be placed at the rear, they would have to be installed with the crankshaft in the fore-and-aft direction, and would then occupy the same proportion of the chassis length.

Rear-Engined Passenger Cars—In recent years there has been considerable propaganda in favor of rear-engined passenger cars. It seems to have been fostered particularly by body designers who were trying to improve the streamlining of their cars, although it originated with aircraft engineers, including Dr. Edmund Rumpler, creator of the "teardrop" car in Germany, and Sir Denistoun Burney, an airship designer, in England. Burney's car, which was demonstrated in this country, had a body simulating a section of a thick aircraft wing, and was fitted with a small in-line engine mounted lengthwise in back of the rear axle under a hood. Considering its low power-to-weight ratio, it had a relatively high maximum speed, and it also showed high fuel economy, as would be expected in view of its excellent streamlining and its low powering, but its low acceleration rendered it unsatisfactory from the American standpoint. It never went into production, which may have been due to the fact that its combination of such (at the time) untried features as a rear-mounted powerplant, independent suspension all around, and frameless construction would cause almost any manufacturer to hesitate.

Considerable numbers of rear-engined passenger cars have been produced by the Daimler-Benz Company of Germany (Mercedes). Aside from the fact that a rear-mounted powerplant facilitates streamlining, it offers the following mechanical and operating advantages: It permits of combining the engine, transmission and final drive gear in a single unit, ensures favorable conditions of visibility for the driver (owing to the forward location of the driver's seat), obviates annoyance of passengers by heat, noise and fumes from the powerplant, and ensures increased adhesion of the driving wheels, because the proportion of the total weight on the drivers will be greater. Disadvantages of rear engines include the need for remote control of engine, clutch and transmission; greater difficulty in effecting engine cooling, rendering it necessary to maintain air under pressure in the engine compartment, or providing an extensive piping system; and greater difficulty in keeping the interior of the car comfortably warm in cold weather.

A rear-engined car will, of necessity, have a large moment of inertia around transverse axes, because the weight of the powerplant at the rear must be balanced by carrying the fuel tank, spare tire or spare wheel, and luggage at the front, in order to ensure a suitable division of load between front and rear wheels. This large moment of inertia is conducive to good riding qualities, but it tends to impair the driving qualities. It slows down all motion around transverse axes, because the moment of inertia is a measure of the resistance to angular acceleration. One result is that any pitching motion of the car will cause less discomfort to passengers. On the other hand, the car will respond less readily to the steering gear, especially at high speeds, because greater resistance must be overcome in deflecting it from its course. Besides, with the heaviest part—the engine—located at the extreme rear, the car has a tendency to become unstable at high speeds. The conditions are substantially the same as if it were attempted to shoot an arrow turned end-for-end, with the weighted end at the back. In order that a road vehicle may be stable while traveling at high speeds, its center of gravity must be located forward of its center of (wind) pressure, and this is difficult to ensure if the engine is located back of the rear axle. Most of the "rear-engined" cars that have been proposed therefore have the engine mounted in front of the rear axle.

It would be practically impossible to find room for powerplants of the type and size currently used in passenger cars forward of the back axle without cramping the passengers. According to one engineer who has done considerable development work on rear-mounted powerplants, engines of about one-half the average present displacement are the practical limit.

There is a psychological objection to placing the driver's seat far forward, as it tends to induce a feeling of insecurity.

It would seem that the rear-mounted powerplant is justifiable and practical only in passenger cars designed primarily with a view to economy. By placing a horizontal opposed engine under the rear seat, a car of given interior dimensions can be built shorter and lighter than the present conventional type, and therefore should be both cheaper to produce and cheaper to operate. Limitations on space would compel the use of a smaller powerplant, another factor making for economy in production and operation. The engine probably would not be as accessible as one under a hood in front, and the space between the front wheelhouses certainly would not be as convenient for stowing luggage as that now used in the rear, but when economy is the chief consideration one must not expect the utmost in comfort and convenience.

While the rear-engined car is a possibility of the future and may become a reality because of the constant quest for novelty and sales features, only front-engined passenger cars were in quantity production in this country during the past few decades.

Front Drive—During the early thirties there was a flurry of interest in front-driven cars in the U. S. In addition to a number of racing cars, two different commercial models were developed, and one of these was in production for several years. The movement gained more of a foothold in Germany, where in 1936 23 per cent of the total production was made up of front-driven models. Such cars were produced also in France (Citroën and Rosengart), England (Alvis and B.S.A.) and Belgium (Imperia).

One advantage claimed for front-driven cars is that they hold the road better on curves, as the propelling force is always in the direction of the front wheels, instead of at an angle thereto. A positive advantage of front drive is that it eliminates the propeller shaft to the rear axle, thereby making it possible to place the floor of the body low without resorting to the expedient of providing it with a tunnel for the shaft. In a front-driven car the engine, transmission and final-drive gear can be combined in a single unit, and as the final-drive gear then is spring-suspended, the unsprung weight is reduced, which tends toward greater riding comfort and reduced tire wear.

The chief disadvantage of front-wheel drive is that under unfavorable conditions it does not give sufficient traction. Conditions are unfavorable when steep grades with slippery surface have to be ascended, and especially when a start has to be made on such a grade. Under these conditions the traction is poor, because the proportion of the load on the front wheels is then too small. In designing front-wheel-driven cars, every effort is made to place as much of the total

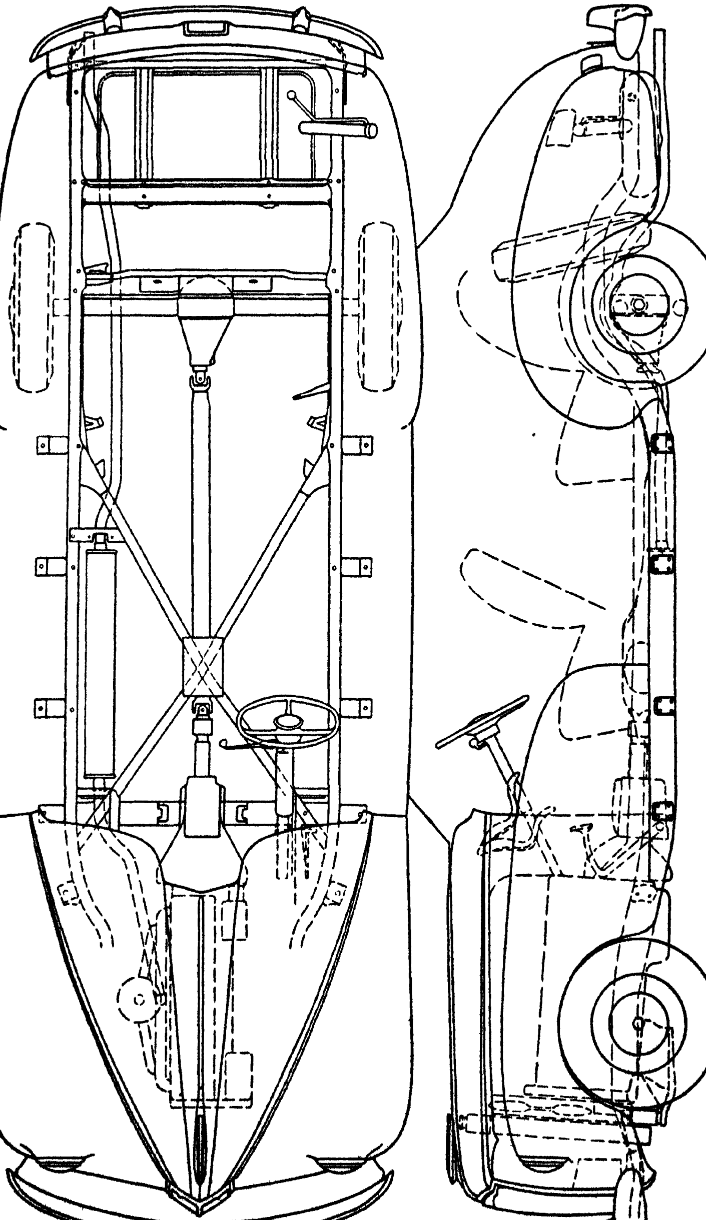


FIG. 2.—PLAN AND SIDE ELEVATION OF PACKARD ONE-TWENTY CHASSIS (120-IN. WHEELBASE).

weight as possible on the front wheels, by using comparatively short engines, such as the V-8 type, the center of gravity of which can be brought close to the front axle; and by placing the transmission ahead of the front axle, instead of between the engine and axle. But on an up-grade, where the maximum traction is needed, the load on the front wheels is reduced by both the inclination of the car and by the reaction to the axle torque. If traction is lost under these conditions, the usual remedy consists in rocking the front wheels from side to side, by means of the steering wheel, until a start is effected; or, alternately, in letting the car run back downhill some distance, to a point where the driving wheels can secure a better hold. It is claimed, however, that complete loss of traction is rather rare even in mountainous districts, and that traction always can be obtained by applying tire chains. With front drive it is necessary to use the so-called constant-velocity universal joints at the front wheels. No front-driven stock cars have been produced in the United States in many years.

Fig. 2 is an elevation and a plan view of a modern passenger car with front-mounted engine. By comparing this illustration with Fig. 1 it will be seen that ideas regarding the proper proportioning of passenger cars have changed greatly during the half century such cars have been in production.

Frame Types—The foundation of the automotive chassis usually is a frame on which different chassis units are mounted, and to which the axles are attached by the springs, and sometimes also by other members serving to transmit or take up the axle thrust and torque reaction. The conventional frame, which is made up of pressed steel members, usually of channel section, is sometimes referred to as the panel type. It consists of longitudinal and cross members, and now usually also comprises an X member (called cruciform in England). Since the panel-type frame is relatively flat, and—except where box sections are used—is made up of members which in themselves possess little torsional stiffness, it offers little resistance to torsion. The situation was not bad as long as engines were mounted rigidly on the frame, as then the crankcase with its integral or bolted-on arms formed a substantial crossmember at the front end; but it deteriorated about 1930, when flexible engine mountings were introduced. Torsional weakness in frames is indicated by shaking or swaying of parts mounted at the forward end—such as an exposed radiator—at high speeds. To increase the rigidity of panel-type frames, they formerly often were provided with tubular cross members of large diameter. A more radical step to eliminate the evils resulting from torsional weakness consisted in the adoption of the tubular-backbone type of frame. This originated in Czechoslovakia, a rather mountainous country where the roads are said to have been in poor condition. In the earliest designs a steel tube of large diameter extended from the transmission housing to the final drive gear housing, both of which were spring-mounted. This tubular backbone was provided with cross members or outriggers on which the body was carried.

Backbone-type frames have been used rather extensively in Continental Europe. In more recent designs the central part of the backbone, a tube of either round or rectangular section, often has forked extensions at both ends, on which the powerplant and the final drive gear are mounted. An intermediate type of frame consists of two longitudinals of either tubular or channel section, rather close together at the middle but receding from each other toward the ends, so as to form forks on which the powerplant and final drive gear are carried. An obvious disadvantage of this type of frame is that it does not afford the same facilities as the panel type for the attachment of fittings.

Unitized Construction—It has long been the general practice to build motor-vehicle chassis and bodies as separate units, and then to mount the latter on the former. It originated when bodies were built in carriage shops, and it undoubtedly has certain advantages from the production standpoint. Trouble with vehicles in service resulting from inadequate frame strength suggested the idea of using the body as the main structural member, eliminating a separate chassis frame. A closed body with a solid steel roof naturally has considerable beam strength, as well as adequate torsional stiffness. The plan is not so well adapted to open or convertible cars, because the door openings in these form weak spots, but means have been found to overcome this difficulty. The method of producing automobiles without a separate chassis frame has been variously referred to as unitized, frameless, and single-unit construction. It would, of course, be desirable that for general use (as distinguished from promotional work) one of these terms should be agreed upon.

Usually the chief reason for the adoption of unitized construction is that it permits of a saving in weight, so that less material is required for production and less power for acceleration and hill climbing. The possibility of weight saving is undisputed, but there is a difference of opinion as regards the magnitude of this saving. One engineer who had studied the subject put it at about 2 per cent of the total vehicle weight, while another, with considerable experience with unitized construction, held it to be substantially equal to the weight of the frame, which in a conventional car amounts to about 7 per cent of the total vehicle weight. Of course, when the frame is eliminated, the body must be provided with extensions at the front, and reinforcements at the rear, on which to mount the suspension members and bumpers, and it must be reinforced also at other points. But a considerable saving in weight remains, and this is added to by possible weight savings in the engine, transmission, axles, springs, and wheels, all of which can be made lighter if less total weight has to be carried and propelled. When these additional savings are taken into account the higher value of the proportional weight saving quoted above is arrived at.

Another advantage of unitized construction is that, for a given road clearance, the floor can be lower. Sills or reinforcing members are required under the floor of the unitized body, but since the main dependence for structural strength is on the body panels, these reinforcements can be of less depth than the side rails of a conventional frame. This advantage is particularly pronounced when the comparison is with a car having an X-type frame, where the minimum road clearance usually is at the center of the X, about midway between front and rear wheels, where ample clearance is particularly desirable.

As the unitized car does not have a separate frame, it probably costs slightly less to produce. It is also claimed to be safer, because the forward extensions, being made of lighter-gauge material than the frame, will have less longitudinal rigidity; hence, in a head-on collision or the like, the force of impact will be smaller, and passengers will not be thrown forward so violently. But these and other advantages claimed depend to a considerable extent on the individual design and may not be really inherent in the principle of unitized construction. Some manufacturers have found it more difficult to eliminate body noises in unitized cars, but the problem can be solved. Another objection that has been raised against this construction is that steel parts on the under side of the body, being of much lighter gauge than the frame rails, are more likely to be seriously weakened by rust where strong corrosive influences prevail, as near the sea shore and in districts where calcium chloride is used to melt ice and snow on the highways. Trouble from this source can be avoided by thoroughly rustproofing exposed parts and sealing all external joints.

Unitized construction of passenger cars has been used more abroad than in this country. Several American manu-

FIG. 3.—BOTTOM VIEW OF NASH AMBASSADOR UNITIZED CAR.

facturers, Chrysler and Ford among them, have given it a trial, but abandoned it. Nash has made use of the principle for upward of a decade, and its construction will be briefly described here.

Nash Single-Unit Construction—Fig. 3 is a bottom view of the Nash Ambassador model, and Fig. 4 a phantom view of the Statesman, with the framing or skeleton of the body structure emphasized. The body is built up of five main subassemblies: The underbody, which consists of the floor with its longitudinal and transverse reinforcements; the body-dash structure, which includes the front-wheel inner housing, the dash-and-instrument-panel structure, and the forward extensions; two side-unit panel assemblies, and finally, the roof, which includes the cowl and the framing for the windshield and the rear window. These units are built in fixtures on

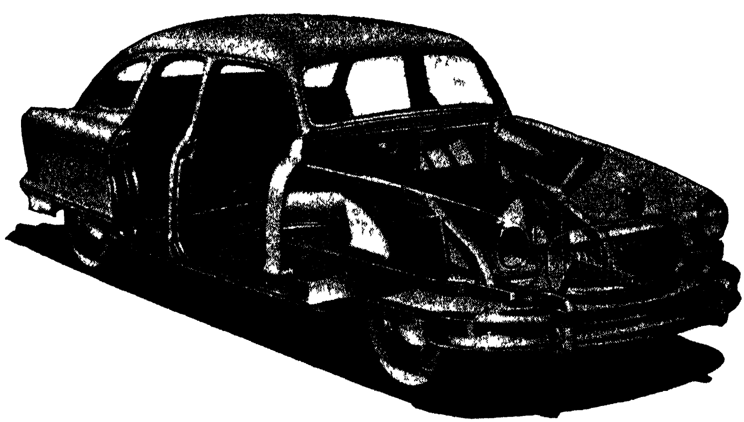


FIG. 4.—PHANTOM VIEW OF NASH STATESMAN UNITIZED CAR.

conveyor lines that terminate at definite points of the main assembly line. Final assembly starts with the underbody, to which the dash structure, the two side-unit assemblies, and the roof unit are assembled in succession. The various units are welded together while being held in fixtures, to ensure proper alignment, about 8000 spot welds being required.

Quite a number of parts that ordinarily are mounted on the chassis are here assembled on the unitized body, including the fuel tank, steering gear, clutch and brake pedals, parking-brake lever, instruments and controls, rear springs, many of the electrical units, front and rear fenders, and radiator grille. Running-gear parts, including the rear axle and torque tube, the powerplant with its rear-support cross member, radiator, front suspension assembly and exhaust system, are separately assembled in a jig or frame on the production line. Where this assembling process is completed the body is lowered onto the running gear and the necessary connections are made between the two units.

Emphasis is laid on the fact that a sort of truss connects the dash structure to the forward end of each forward extension, with the result that both the vertical and the torsional stiffness of the car as a whole decrease gradually from the center toward the front end, instead of abruptly at the forward end of the body, as in a conventional car. Sudden changes in strength or stiffness always involve stress concentration and poor utilization of material.

Nash also builds a convertible in unitized form, the necessary rigidity being provided there by a permanent overhead-rail structure.

Bus Design—Among American manufacturers of buses or coaches, frameless construction is the prevailing practice. The bus may be built in the first place with a light base or floor frame to which the front and rear axles are attached, but this later becomes an integral part of the body framework, and there is then no separate chassis. The entire frame forms a single structural unit which takes both bending and torsional stresses, and the sheet-metal covering also takes part in sustaining these stresses, thus making use of the principle of stressed-skin structures, quite familiar in the aircraft industry. Most of the frame members usually are pressed-steel channels, though square-section tubes also are being used. The channels sometimes are provided with outward flanges at their edges, to which the body sheets can be conveniently riveted. All members are joined together by welding. Door and window posts are welded to outriggers on the base, to floor cross bearers, and to the roof carlines, and are connected together by horizontal frame members at the skirt bottom, floor level, seats, belt, letter board and on the roof. In these buses the powerplant usually is located at the rear, in a compartment entirely within the body structure. The rigidity of the body is further increased by welding such parts as the wheel housings, step wells, and powerplant-compartment bulkhead to it. The chief advantage of this form of construction is that it affords a satisfactory degree of rigidity with a minimum weight of material.

Fig. 5 shows a floor plan and a side elevation of a White 40-passenger city bus, which is of what the manufacturer calls "integral construction." The body sides are the load-carrying members. Each side member is a steel-plate girder; it

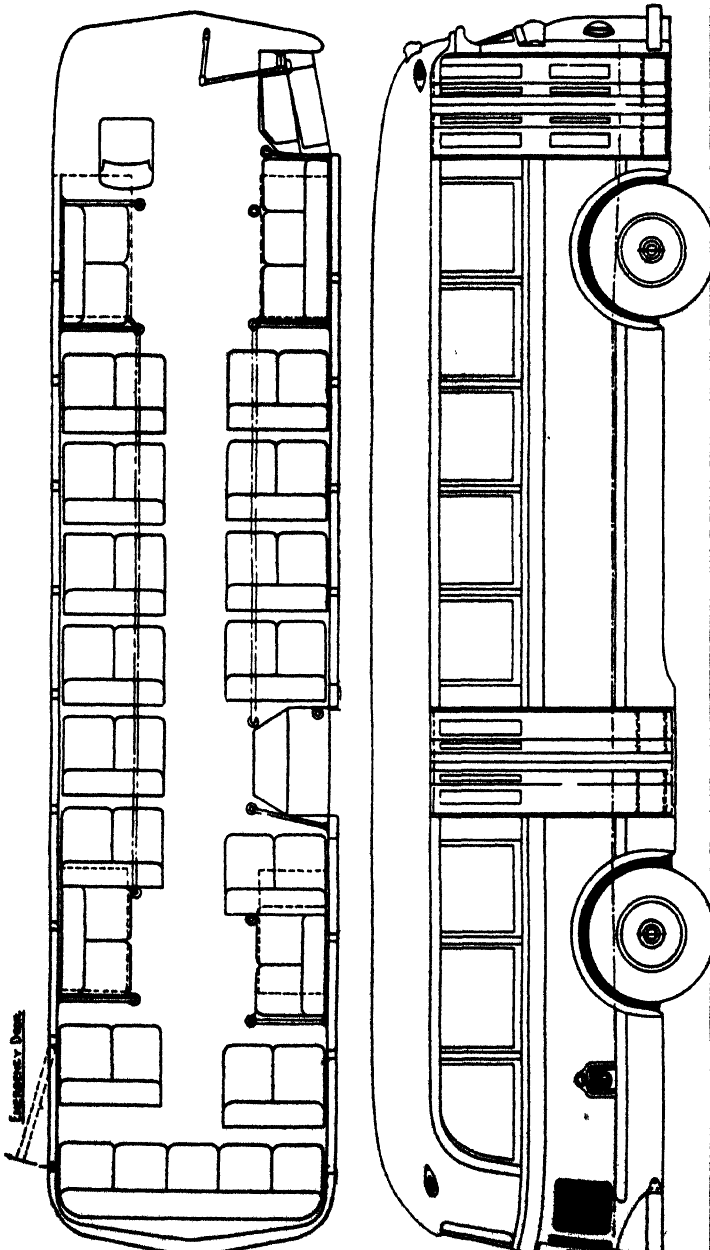


FIG. 5.—FLOOR PLAN AND SIDE ELEVATION OF WHITE 40-PASSENGER BUS, OF "INTEGRAL" CONSTRUCTION.

forms the body side panels, and it extends between the corner posts forward of the front and back of the rear wheels. Heavy transverse girders at the spring ends transmit the load through the springs to the axles. This bus has a wheelbase of 214 in. and an overall length of 396 in. It is equipped with a 12-cylinder horizontal opposed engine mounted below the floor amidships. With the fuel tank filled, the bus weighs 18,450 lb, of which 7250 lb is carried on the front and 11,200 lb on the rear wheels. The engine develops a maximum torque of 500 lb-ft at 1200 rpm, and a maximum output of 207 hp at 2600 rpm.

Powerplant Location in Commercial Vehicles—The inefficiency of space utilization in a vehicle powered by a relatively long engine under a hood in front was first realized by manufacturers of motor buses. When located in this position, the engine cuts down the space on the chassis available for seats, thus reducing the earning capacity of the bus. Manufacturers of buses in the U. S. therefore have practically abandoned this type, and adopted the so-called transit type instead, in which the powerplant is located either below the chassis frame amidships, or under a rear seat extending entirely across the vehicle. With the engine in either of these positions, the entire floor area is available for passenger-carrying purposes, with the exception of that occupied by the driver. In England the practice of placing the engine at the front of the bus chassis was continued a good deal longer than in the U. S., the loss in seating capacity being minimized there by locating the engine over to one side of the chassis and seating the driver alongside, instead of behind it. More recently, however, British manufacturers of buses or coaches also have switched to the transit type of construction, and at the London commercial-vehicle show in 1950 there were eight bus chassis with under-floor powerplants and one with a powerplant mounted transversely at the rear. All of the larger buses were equipped with Diesel engines.

Attention may be drawn in this connection to the fact that a V-engine occupies less space on the chassis than an in-line engine of the same displacement, and therefore makes for better space utilization. V-engines are used to a certain extent for all types of vehicle. That they are not used more for commercial vehicles probably is due to the fact that the smallest number of cylinders in which V-engines are normally built is eight, and six is generally considered a sufficient number for comercial-vehicle engines. With six-cylinder V-engines it is impossible to get both a uniform sequence of explosions and good mechanical balance, and such engines are not being built (except on a small scale by Lancia in Italy). The radial engine, which is so largely used in aircraft, is more compact in the axial or longitudinal direction than any other type, but it does not lend itself well to installation in a road vehicle. Besides, it is rather more expensive to manufacture than other types, because its cylinders must be machined individually.

COE Trucks—In motor trucks it is impractical to locate the powerplant at the rear, because trucks are loaded from the rear, and no part of the mechanism can be allowed to project above the chassis frame there. The original arrangement, with the driver's seat or cab located behind the engine compartment, is uneconomical also in this case, and here the problem has been solved by the development of the COE (cab-over-engine) type. COE trucks and tractor trucks have come into wide use in recent years. Trucks of this type provide more loading space for a given length of wheelbase, and tractor trucks designed to haul semi-trailers can be built with shorter wheelbases. The saving in length is quite substantial, for whereas tractor trucks of conventional design come in wheelbases of 130 to 170 in., those of the COE type have wheelbases of from 90 to 105 in. One objection to the cab-over-engine construction has been that it makes parts requiring occasional servicing less accessible, and it also makes it more difficult to remove the powerplant for an overhaul. However, much effort has been directed toward eliminating (or at least reducing) these drawbacks. The gain in loading space resulting from placing the cab over the engine is illustrated in Fig. 6, which shows side elevations (and a rear view) of a COE and a conventional truck, both of the same make. It is customary to mount bodies of the same length on both a conventional chassis and a COE chassis of materially shorter wheelbase. For instance, one manufacturer mounts the same body of 106 in. inside length on a COE chassis of 101 in., and a conventional chassis of 134 in. wheelbase; and also the same stake body of 142 in. inside length on a COE chassis of 134 in., and a regular chassis of 158 in. wheelbase.

An advantage of the COE truck that is much appreciated by some operators is that, owing to its shorter wheelbase, it has a shorter turning radius, and therefore can be handled to better advantage in restricted spaces, as in garages with obstructing posts, etc.

Power-Tilted Cab—One solution of the problem of making the powerplants of COE trucks more accessible consists in mounting the cab pivotally on the frame, so that it can be tilted out of the way. Fig. 7 shows the power-operated tilting cab of a White delivery truck. A 6-volt electric motor with

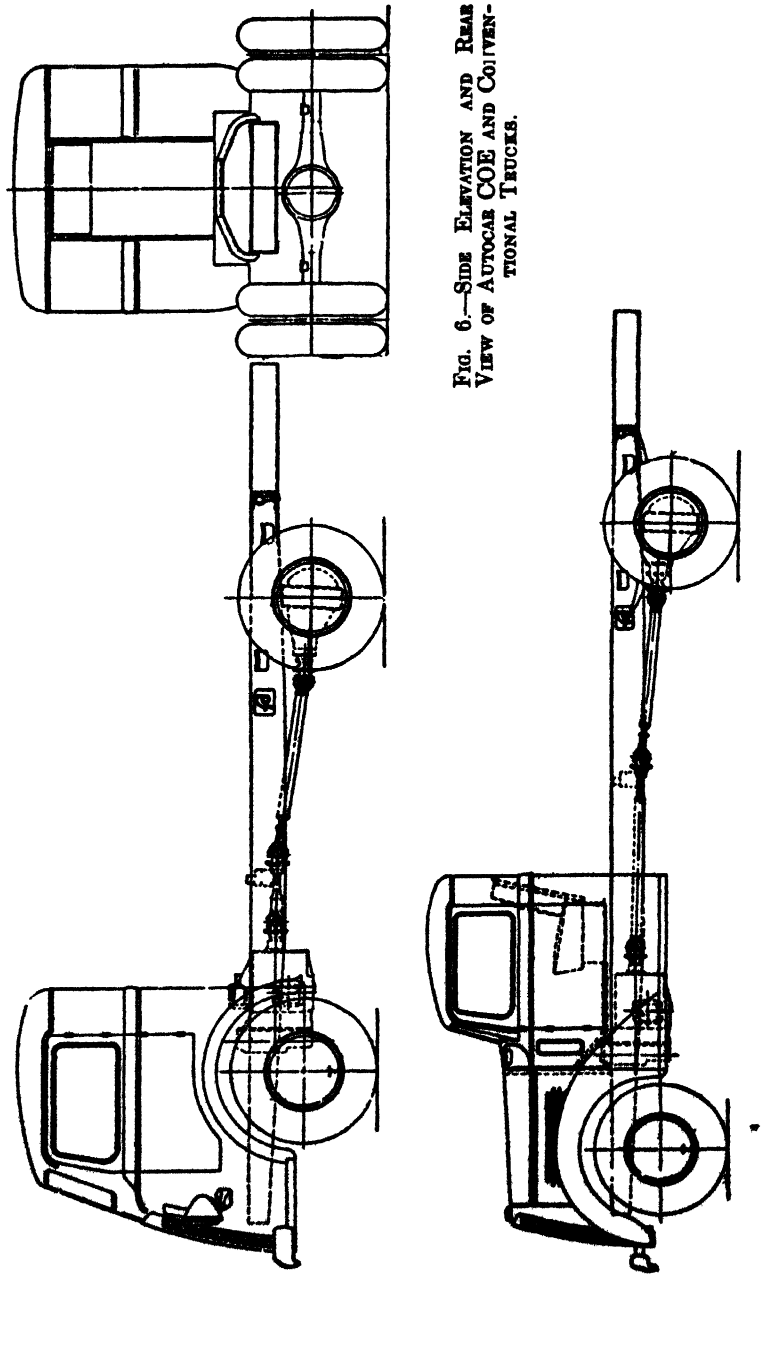


FIG. 6.—SIDE ELEVATION AND REAR VIEW OF AUTOCAR COE AND CONVENTIONAL TRUCKS.

reducing gear operates a screw jack that is shackled to the cab. The latter is hinged at the forward end of the frame and can be tilted through an angle of 90 deg in 25 seconds. When the cab is thus tilted, all powerplant and cooling-system units are accessible for adjustment or repair. Front fenders and other sheet-metal parts at the forward end of the chassis are moved out of the way with the cab. The operating mechanism is mounted on a tubular cross member of the chassis frame. In assembling the truck, the complete powerplant, including the transmission and radiator, is lowered into position on the frame. Later on the cab is installed while sus-

Fig. 7.—Power-Tilted Cab (White).

pended in a massive cradle which permits of holding it in the tilted position while it is being pivoted at the front and while the actuating mechanism is being attached. It is claimed that the tilting-cab feature reduces servicing time by 20 per cent.

Semi-Trailers and Fifth Wheels—Much of the heavy freight now moving over the highways is carried in semi-trailers drawn by truck tractors. The semi-trailer has either one axle or a pair of tandem axles at its rear end, but no front axle, its forward end being supported on the tractor by means of a fifth wheel or coupler. To enable the combination (or train) to negotiate curves without side slip of any of its wheels, the coupling must afford freedom of movement around a vertical axis, and to prevent undue changes in load distribution between axles when encountering undulations in the road surface, there must be also freedom of movement around a transverse horizontal axis. All fifth wheels or couplers therefore have at least two pivot joints, a kingpin on one member forming a vertical pivot, and a trunnion support of the other member on its brackets, a transverse horizontal pivot. Sometimes there is an additional horizontal pivot, in the fore-and-aft direction, which gives the member so mounted a universal support, enabling it to rock virtually around any horizontal axis. These couplings with universal mountings are used chiefly for tank trailers, to protect the tanks against

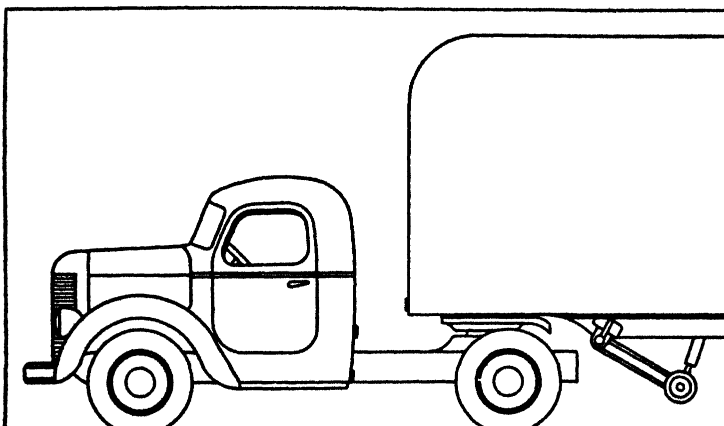


FIG. 8.—TRACTOR AND PART OF SEMI-TRAILER.

excessive torsional stresses when traveling over very uneven roads. A truck tractor and the forward part of a semi-trailer are shown in side view in Fig. 8.

Where loading and unloading take considerable time as compared with that spent on the road, what is known as shuttle operation may be resorted to. The tractor is then uncoupled from the loaded trailer upon arriving at the destination, and coupled to an empty trailer for the return trip. In that way a certain amount of haulage work can be performed with less investment in equipment.

Door-to-Door Delivery Wagons—To make it possible for motor vehicles to compete with horse vehicles in such services as milk delivery to private customers, the former must be so designed that the operator can enter and leave with the least effort, and that operation of the vehicle is simple. For this class of service a special type of vehicle, known as a door-to-door delivery, or a multiple-stop vehicle, has been developed. In such vehicles the floor of the driver's compartment is only one step above the ground, and a stool- or bucket-type seat is provided. The lowness of the floor makes it necessary that the vehicle have either front drive with a front-mounted powerplant, rear drive with a rear powerplant, or in the case of a front-mounted powerplant combined with rear drive, a deep tunnel for the propeller shaft extending through the driver's compartment. Frameless construction is well adapted to vehicles of this type, in which the powerplant is located adjacent to the driving axle. Where a deep propeller-shaft tunnel is used, this can be taken advantage of to stiffen the frame. In multi-stop delivery wagons a ledge or shelf is provided in the forward part of the body on which the driver keeps his record books and other paraphernalia. The loading space is accessible through the side doors.

A representative of this type is the Divco, the chassis of which is illustrated in Fig. 9, while a phantom view of the complete vehicle is shown in Fig. 10. The powerplant is located at the front, with drive to the rear axle, and the chief difference between this chassis and that of a conventional light truck is that the frame has a double drop, which brings the floor of the "through aisle" so low (less than 14 in. above ground with a normal load) that the driver can step in directly from the curb. The seat, which is adjustable both vertically and horizontally, is pivotally mounted, and can be swung out of the way under the steering post to facilitate loading. A special feature of this vehicle is that it can be driven with the driver either standing or seated. When standing, he can use only one foot for the control of the vehicle, and this calls for a rather unusual arrangement of the controls. There are two accelerator pedals, one on the floor for standing operation, the other in a more convenient position for sitting operation. A hand-operated so-called snubber brake acts hydraulically on the road wheels, and in addition there is a transmission brake, which is applied by means of the combined clutch-and-brake pedal. This pedal moves over a ratchet sector and the transmission brake is used as a parking brake.

The vehicle here illustrated has a wheelbase of 100¾ in., a curb weight of 4447 lb, and a gross-vehicle-weight rating of 6000 lb. Its four-cylinder engine has a displacement of 162.4 cu in. and develops 47 hp at 2800 rpm, but is governed at

Fig. 9.—Drop-Frame Chassis of Door-to-Door Delivery Wagon (Divco).

2640 rpm. Four speeds are provided by the transmission, whose shift lever is mounted on the steering post. Tires are 7.00/16 in. all around, and the rear-axle reduction is such as to give a road speed of 32.6 mph in direct drive at governed engine speed. There is 122 cu ft of loading space back of the through aisle, and 12 cu ft in front of it.

Legal Limits on Overall Dimensions and Weights—The overall dimensions of motor vehicles for use on public roads, as well as the gross vehicle weights and the weights per axle or per inch of tire width, are limited by law in the different states. The limits, of course, vary somewhat, but as trucks

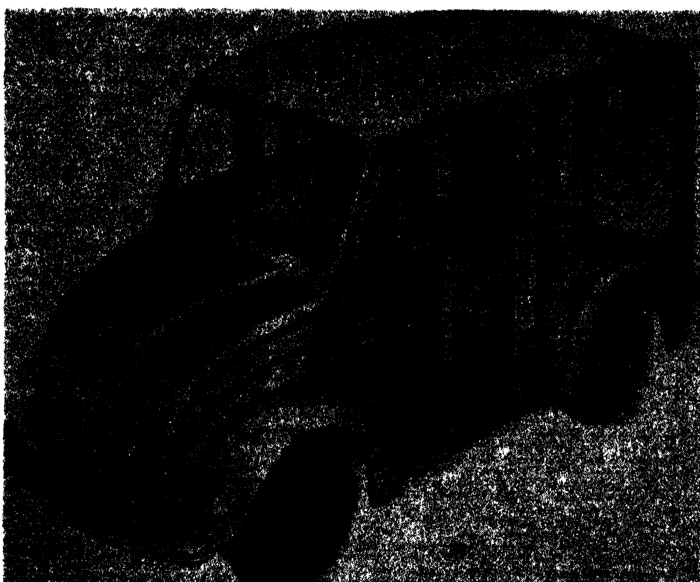


FIG. 10.—PHANTOM VIEW OF DIVCO DOOR-TO-DOOR DELIVERY WAGON.

usually are intended to be saleable and legally operable throughout the country, the minimum limits are of particular interest to the designer, especially if the minima are the same in quite a number of states. The limit on the overall width in most states is 96 in., and the limit on overall height, 12½ ft. In 1950 no states had any lower limits on these two dimensions, while a few states had higher limits. The lowest limit on the overall length of single units is 35 ft, except in Virginia, where it is only 33 ft for trucks, though buses may be 35 ft long. Tractor and semi-trailer combinations are limited to 45 ft in a good many states, though some states allow lengths of 60 and even 65 ft. There are legal limits also on the number of semi-trailers or full trailers which may be used in a train, and on the minimum spacing of tandem axles.

Limits on the gross vehicle weight of four-wheel single units range all the way from 26,000 to 44,000 lb; limits on weight per inch of tire width, from 500 to 800 lb, and limits on weight per axle generally from 18,000 to 22,000 lb, though there are limits both lower and higher in a few states. These weight limits apply to vehicles with pneumatic tires. Some states distinguish between balloon and other pneumatic tires, allowing a 2000 lb higher axle load with balloon tires. Summaries of legal restrictions on size and weight are published periodically in the motor-truck journals.

Off-the-Road Trucks—Trucks and tractor trucks of special design are being built for use in quarries and in other operations where they do not have to travel on public highways, and therefore are not subject to legal restrictions on their overall dimensions and weight. In truck operation the driver's wage always is an important cost item, but its relative importance decreases with an increase in the load carried. Therefore, for economical operation under conditions where full loads are always assured, the truck should be as large as one man can handle without excessive effort.

Owing to the rough terrain over which these trucks must operate, their speeds must be kept low. They usually are provided with dump bodies, either side-dumping or rear-dumping, while tractor trucks sometimes are used together with trailers that dump through their bottom. Frames and bodies for off-the-road trucks must be of very rigid construction, to be able to withstand the shocks produced when power shovels dump their contents of rock or excavation onto them. The frame for a 15-ton truck, for instance, may have side members of 12-in. rolled steel channels. Multiple-speed transmissions are used, sometimes together with auxiliary transmissions, and truck speeds at governed engine speed range between about 2 and about 20 mph. The low vehicle speed in top gear calls for a reduction ratio of between 15:1 and 20:1 between engine and drive wheels, and an extra gear reduction may be provided at the wheels, in addition to the reduction at the differential.

Off-the-road trucks of 15 tons pay-load capacity are powered with engines of about 150 hp. Chassis and body together also weigh about 15 tons, making the gross vehicle weight 60,000 lb. Assuming that 125 hp is available at the wheel rims, the maximum tractive force in low gear (2 mph) is about 23,500 lb, which is sufficient to take the loaded truck over any terrain where its driving wheels can secure sufficient traction. A truck of this type would have a track of perhaps 90 in. and a wheelbase of 160 in. Special balloon tires are built for off-the-road trucks, and a truck of 60,000 lb g.v.w. would be equipped with dual rear 14.000/24 in. tires and somewhat smaller single front tires, the division of load between front and rear wheels usually being in the proportion of about 1:3.

What ordinarily limits the size of vehicle that can be operated safely by one man is the effort required to steer it. A front-axle load of 8000 lb is considered the practical limit with hand steering. During World War II considerable development work was done on power steering gears, for vehicles used for retrieving disabled "tanks," etc., and it is quite likely that power steering will become a common feature of large "off-the-road" commercial vehicles hereafter.

Industrial Trucks and Tractors—Previous to World War I practically all transportation of materials and parts in factories, and of baggage, express goods, and freight on railway platforms was effected by means of hand trucks. Small electric trucks first made their appearance at the Broad Street Station of the Pennsylvania Railroad in Philadelphia and in Altoona, in 1906. Their use in industry expanded rapidly during World War I, when industrial production was keyed to a high pitch. Electric propulsion probably is at its best in that field, because—except where steep ramps must be ascended—power requirements are more nearly uniform than in road transportation, and the vehicles always remain close to a source of charging current. Besides, the electric truck has the advantages of comparatively silent operation and freedom from exhaust smoke and odor, which may explain why it retained a virtual monopoly in the industrial field until the early thirties. However, with gradual improvement of combustion engines and increase in the size of transport units, so-called gas-powered trucks were introduced and gained wide acceptance. They are being manufactured in a great range of sizes, some having single-cylinder engines of the type used on lawn mowers, while others are equipped with four- and six-cylinder engines of outputs that in some cases exceed 50 hp.

There are essentially four types of industrial, wheeled transport units, viz., tractors, ordinary industrial trucks, lift trucks, and fork trucks. The lift truck has a low platform adapted to pass under a pallet or skid on which the materials or parts to be transported have been loaded. The platform is then raised and the pallet lifted off the floor, hence the truck is virtually "self-loading." At its destination the pallet is deposited by merely lowering the platform. In a fork truck the pallet with its load is picked up by forks which can be tilted and also raised and lowered on a column by hydraulic means, to facilitate stacking in tiers. The columns are of a telescopic design and can be lowered to a minimum height of about 83 in. above the floor, which enables the trucks to enter elevators and freight cars with adequate clearance. The hydraulic equipment comprises cylinders for tilting the column and raising and lowering the forks, respectively; a pump supplying oil under pressure; valves for controlling the admission of oil to the cylinders, and an oil tank. The pump, which is driven from the engine, delivers oil as long as the engine is running, but ordinarily the oil returns directly from the delivery to the suction port, and is not subjected to material pressure.

Reverting to the subject of industrial trucks and tractors in general, the smaller units, designed mainly for indoor use, are equipped with solid rubber tires and with transmissions giving two or three forward speeds, the maximum speed being limited to about 7 mph. Larger units, for use in factory yards and for road haulage, sometimes are equipped with pneumatic tires and with transmissions affording either three or four forward speeds, and their maximum speeds are around 10 or 12 mph.

Industrial trucks and tractors are built in regular three- and four-wheel types, and also with caster-mounted twin steering wheels. Either the front or the rear axle may be the driving axle. Front drive has an advantage in the case of lift trucks and fork trucks, because in these the greater part of the "loaded" weight is on the front axle, and front drive therefore affords greater traction. Final drive is frequently by worm gear, because the total reduction between the engine and the drive wheels must be quite large.

Fig. 11 is a side elevation of a fork truck manufactured by the Elwell-Parker Electric Company, with the powerplant and drive units dotted in. With this truck the driver is seated on a box containing the powerplant, whereas in electric industrial trucks he often stands on a rear platform. The powerplant is of the four-cylinder type; it comprises a sliding-gear transmission giving two forward speeds and a reverse, and it is equipped with an electric generator and a starting motor. As compared with passenger-car installations, the powerplant may be said to be turned end for end, the radiator and fan being at the rear, and the transmission at the forward end. From the latter a toothed chain carries the power to a transfer case, whence it is transmitted to the front axle by a short propeller shaft with universal joints, and a worm-gear drive. The truck is steered by its single, center-mounted rear wheel, by means of a conventional steering gear, from which a drag link extends aft. As the load is carried on the forks in front of the driving axle and produces a moment around the axis

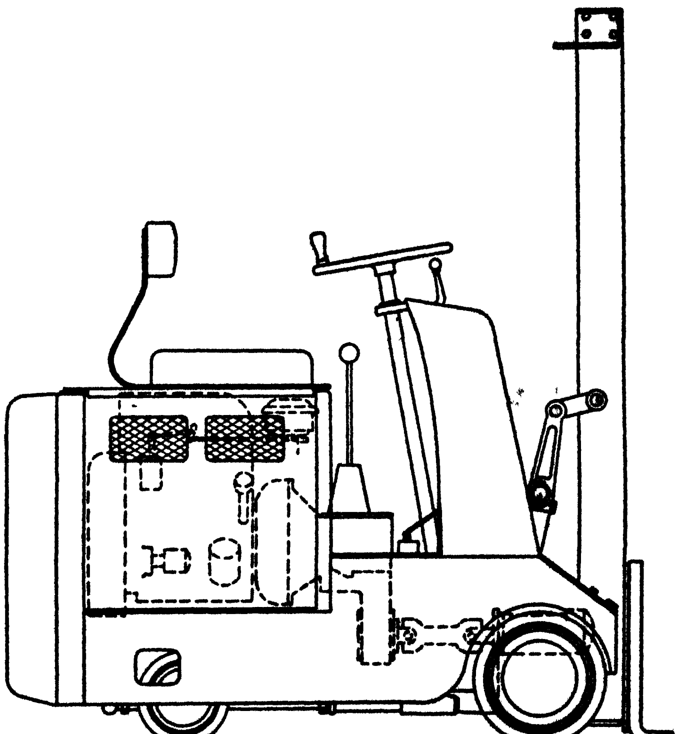


FIG. 11.—SIDE ELEVATION OF ELWELL-PARKER FORK TRUCK.

of the latter, it is necessary to provide a larger, opposite moment, so that there will be enough weight on the trailer wheel to ensure positive steering. This requirement is met by mounting a counterweight on the rear end of the truck. Solid rubber tires are fitted. With these there is no advantage in interchangeability of front and rear tires, and the trailer wheel therefore is made smaller, because it is located under the engine, where the available height is limited.

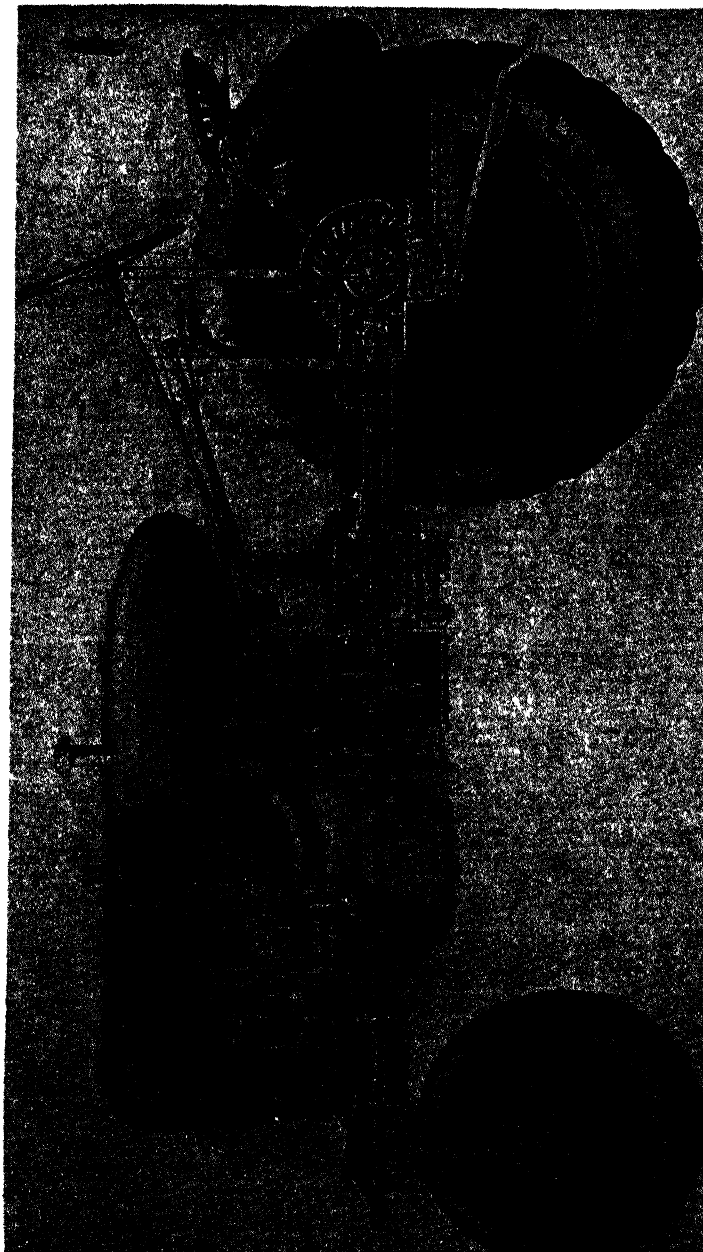


Fig. 12.—Longitudinal Vertical Section of Allis-Chalmers Model WC Farm Tractor.

Farm Tractors—The earliest farm tractors had panel-type frames extending between the front and rear axles, on which the various chassis units were mounted. Modern wheeled tractors usually are of the "backbone" type, the housings of the engine, transmission and rear axle being bolted together to form the main structural member. The prototype of the backbone or frameless tractor was a product of the Case Plow Company of Racine, Wis., now the Massey-Harris Co. It had a tapering semi-shell of boiler steel which formed the lower parts of the engine crankcase, the transmission case, and the rear-axle center housing. Ford Motor Company later improved on this design by bolting cast engine, transmission and rear-axle housings together to form the backbone.

Fig. 12 is a longitudinal vertical section of the Allis-Chalmers Model WC tractor, which has unusually high ground clearance, and therefore is specially adapted to the cultivation of certain row crops. Most farm tractors now are equipped with rubber tires, which make the tractor more comfortable for the driver, and reduce the power consumed in propelling the tractor itself. With steel tires, tractors usually have a drawbar rating substantially equal to two-thirds the belt horse-power rating, indicating that about one-third of the engine power is consumed in moving the tractor over the ground, while the remaining two-thirds is available for hauling implements. With rubber tires the drawbar rating usually is about four-fifths of the belt rating, hence only one-fifth of the engine power is required to move the tractor itself, instead of one-third. Rubber tires, moreover, enable the tractor to travel over the highway at speeds materially greater than can be used for plowing and other field operations, and rubber-tired tractors usually are equipped with four-speed transmissions and have a nominal speed in top gear approximately twice that of steel-tired tractors. In the Allis-Chalmers tractor shown, the engine crankcase, clutch housing and transmission case are flange-bolted together. A forward extension from the engine crankcase carries a vertical column enclosing the post on which the steering wheels are mounted, while a rearward extension from the transmission housing bolts to the gear carrier of the rear axle. In order to obtain the desired ground clearance, the rear axle is cranked or offset, the center portion being at a considerably higher level than the axle spindles. A double-reduction drive is used, the speed being reduced by spiral bevel gears at the center of the axle, and by spur gears at the ends. The four-speed transmission gives nominal speeds of 2.64, 3.84, 5.10 and 9.77 mph. Rear tires are 11.00/28 in.

The special pneumatic tires made for farm tractors are formed with deep integral bars on their treads, to improve the traction conditions. To further increase the traction with pneumatic tires, cast-iron weights may be secured to the driving wheels, and the tires partly filled with a non-freezing solution, to increase the load on them.

Farm tractors are made in a considerable range of sizes, with from less than 20 to more than 50 belt horse-power rating. The belt horse power is substantially equal to the net power of the engine. Because the service usually is severe and continuous, tractor engines are governed to run at moderate speeds. Piston speeds at governed engine speeds vary from about 850 to about 1150 fpm, averaging close to 1000 fpm. Owing to the considerable range of piston speeds, and the fact that different fuels and different engine cycles (Otto and Diesel) are used, the cu in. of piston displacement per belt horse power varies considerably, from about 4.5 to about 9.0, the average being close to 6.6. Four-cylinder engines predominate, but two-cylinder and six-cylinder also are being used.

Two-Wheel Tractors—Tractors can be built also with a single axle supported by two driving wheels, a construction which has the advantage that the whole weight of the tractor (and even some of that of the implement or trailer attached) can be made available for traction purposes. When power is applied to move the driving wheels forward, the axle housing and everything supported by it tends to revolve in the opposite direction, and such rotation must be prevented by a torque member extending to the rear, which either is supported by or forms part of the trailer or implement hauled. To permit of steering, there must be a vertical pivotal connection between the tractor and torque member, fairly close to the tractor axle.

A farm tractor of this type, the Moline Universal, was in large production for several years around 1920. A torque member, forked at its forward end and having a vertical pivot connection with the tractor, was supported at its rear end on the plow, in front of the operator's seat. All control members, including the steering wheel, were mounted on the rear end of the torque member. Steering was through a spur pinion carried on the torque member that meshed with a gear sector on the tractor. Torque applied to the tractor wheels transferred some of the tractor weight to the plow. Most of the tractor weight was ahead of the axle, so that without power on the driving wheels there was an upward force on the torque member, tending to lift the plow out of the ground. With a normal drawbar load enough tractor weight was transferred to convert the upward to a slight downward force on the plow, and it was claimed that under these conditions about 98 per cent of the tractor weight was available for traction purposes.

In recent years the same general principle has been applied in an earth-moving machine known as the Heiliner, a combined tractor, scraper, and bottom dump wagon. In this case a sort of goose neck formed on the dump wagon serves as the torque member, being pivoted to a vertical spindle on the tractor frame a short distance behind the axle. The entire power-plant and the driver's seat are in front of the axle, so that without power on the tractor wheels the latter carry not only all of the tractor weight but also some of that of the dump wagon. Power application naturally changes the distribution of weight between the tractor wheels and those of the dump wagon. Steering is effected by means of double-acting hydraulic cylinders, and may be assisted by applying a brake to one of the wheels, that on the inside of the turn to be made. Fig. 13 is a side view of the Heiliner tractor and part of the dump wagon, showing the hitch or pivotal connection between the two units.

This tractor is equipped with a 200-hp engine and has a low-gear speed of 2.81 mph; so, allowing for a loss of 20 per cent in transmission, it should be able to produce a maximum tractive force of more than 21,000 lb. This is just about equal to the weight of the tractor itself. The use of high-traction lug-type tires and support of the front of the dump wagon on the tractor are designed to enable the wheels to produce this tractive effort with minimum slip.

Tracklayer Tractors—A tracklayer tractor is essentially a locomotive that carries and lays its own tracks. It is propelled over these tracks in the same manner as the locomotive of a cog railway over its rails. The tracks are endless, and consist of links joined together by pins, which latter are surrounded by bushings that are engaged by the sprockets of the driving mechanism. As far as practical use is concerned, the chief difference between the tracklayer tractor and the wheeled type is that in the former the ground-contact area is relatively much larger, and the pressure per unit of contact area therefore much lower, which enables it to operate on soggy or marshy soil, and on sand, where a wheeled tractor would get mired or would not be able to secure a foothold. The average load on the chain track varies from about 5 psi in the smaller

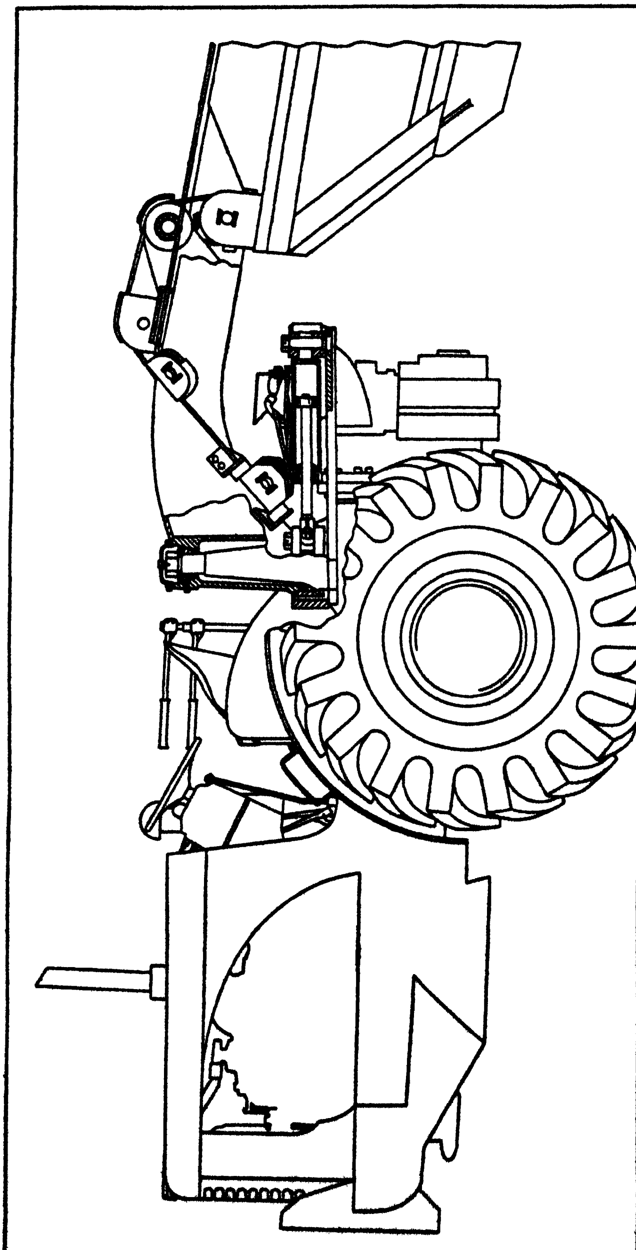


Fig. 13.—Side Elevation of Heiliner Two-Wheel Tractor and Forward Part of Dump Truck.

to nearly 10 psi in the larger models, and is of the same general order as the pressure of a man's feet on the ground.

Whereas wheeled tractors are used chiefly in farming or field operations, tracklayer tractors find their widest use in road building and other construction work.

Fig. 14 is a longitudinal vertical section of the Allis-Chalmers Model HD-14 tractor, which is equipped with a six-cylinder General Motors Diesel engine developing 150 belt hp at its governed speed of 1500 rpm. Its transmission gives six forward speeds ranging from 1.72 to 7.03 mph, and two reverse speeds of 2.0 and 3.2 mph respectively. The shipping weight of this large tractor is 28,780 lb and its maximum drawbar pull in low gear, 28,000 lb.

Referring to tracklayer tractors generally, the tractor proper is supported on the tracks by a series of rollers on each side, carried in bearings on a track-roller frame. Engine, transmission and driving gear are combined in a single unit, their housings being flange-bolted together. This unit is carried on the track-roller frame by means of a transverse pivot shaft under the transmission, and a transverse leaf spring under the rear part of the engine, the two together providing a three-point, flexible support which minimizes stresses on the engine and transmission housings when the tractor travels over rough terrain.

In the transmission of tracklayer tractors the drive usually is from a primary to a secondary shaft in all gear speeds. The secondary shaft drives through a pair of bevel or spiral bevel gears to the transverse differential shafts, and the latter to the track sprockets through spur gears. All of the gears are fully enclosed. The shafts carrying the sprocket wheels have bearings in the walls of the housing, and these bearings are provided with seals that keep the lubricant in and dust and dirt out.

The chain track runs over sprockets at the rear and over idlers at the front of the tractor. Bearings for the idlers are so mounted that they can slide lengthwise on the track-roller frame and are pressed forward by coiled cushion springs which maintain tension in the chain tracks and relieve shocks. On the larger tractors the upper part of the track is supported by track-carrier rollers on top of the track-roller frames.

Tracklayer tractors are built in a wide range of sizes, shipping weights ranging from a little over 3000 to over 30,000 lb, and piston displacements from about 120 to about 1200 cu in. The larger sizes predominate, and the average piston displacement of all tracklayer models is much higher than that of all wheeled models. About one half of all track-

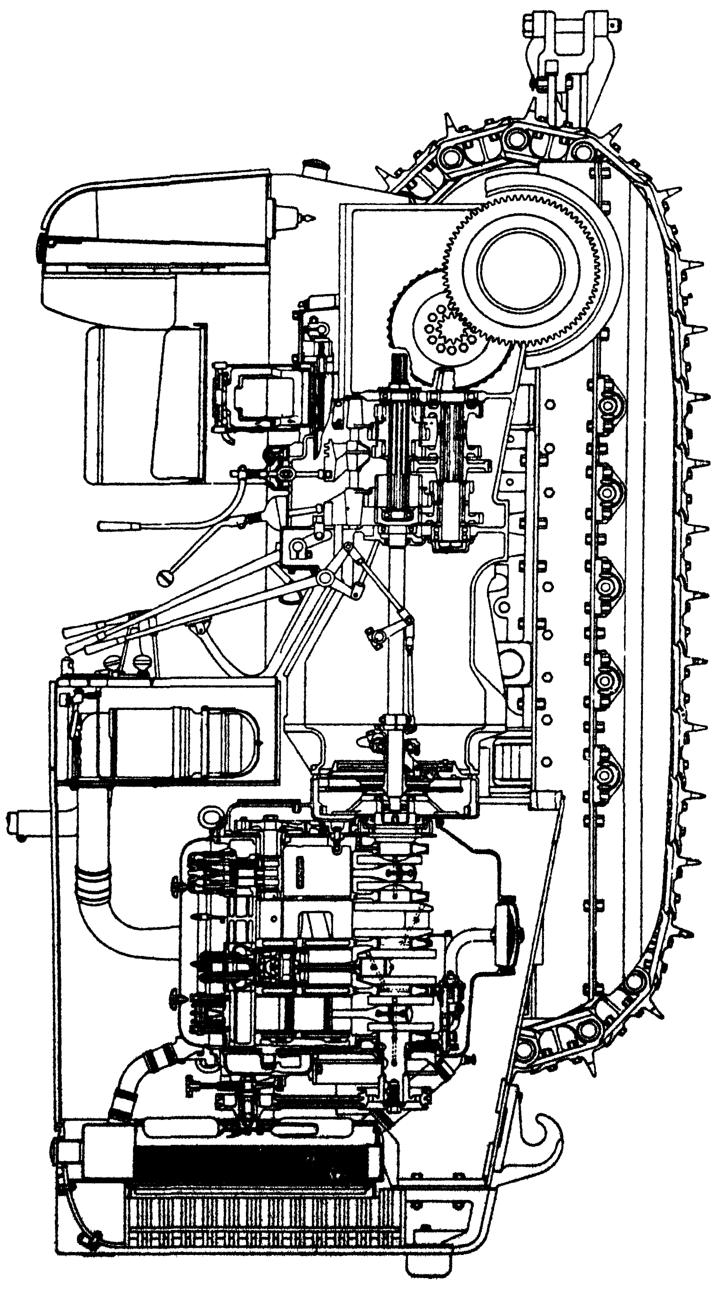


FIG. 14.—LONGITUDINAL VERTICAL SECTION OF ALLIS-CHALMERS MODEL HD-14 TRACKLAYER TRACTOR.

layer models are equipped with Diesel engines. Except in the very smallest models, the transmissions have either four, five, or six forward speeds. The first speed usually lies between 1.5 and 2 mph, and the highest between 5 and 7 mph, the average theoretical low speed being 1.6 and the average high speed, 5.2 mph. Under load the actual speed is somewhat lower than the theoretical, due to slippage of the track on the ground. Some of these tractors are provided with two reverse speeds, which are desirable in road-building and similar operations, where the tractor sometimes must be backed considerable distances.

The drawbar pull rating in low gear sometimes is equal to the shipping weight, but on the average it is just under 90 per cent of the shipping weight. Drawbar horse power, drawbar pull and speed are connected by the following equation:

$$P = \frac{375 \times \text{hp}}{V} \text{ lb},$$

where hp is the drawbar horse power and V the speed of travel in mph.

All linear dimensions of these tractors vary substantially as the cube root of the shipping weight. Treads of most tractors are adjustable, and the minimum tread width averages $2.15\sqrt[3]{W}$. The width of the shoes is about $0.66\sqrt[3]{W}$, and the length of track on the ground, $3\sqrt[3]{W}$. The ratio of shipping weight to belt horse power varies from about 150 in the smaller to 250 in the larger sizes, and averages close to 200 lb/hp. Drawbar horse power ratings of these tractors average 86 per cent of their belt horse power ratings.

Treads—The two basic dimensions of a motor-vehicle chassis or tractor are its tread and its wheelbase. The tread or track is the distance between center points of ground contact on opposite sides of the vehicle. It is often different for the front and rear wheels. The National Automobile Chamber of Commerce (predecessor of the Automobile Manufacturers Association) at one time adopted a standard of 56 in. for the tread of passenger cars, but there never has been a standard tread for either buses or trucks. Adoption of the 56-in. standard tread in the early years of the industry was due to the fact that all horse vehicles in the Northern part of the country had a tread of 56½ in., measured from outside to outside of their steel tires, and an automobile with a tread of 56 in. would run in ruts made by horse vehicles. But as the proportion of hard-surfaced roads increased, the need for a standard tread diminished. When the low-pressure tire was introduced, during the middle twenties, owing to its greater width as compared with a high-pressure tire of like capacity, it became impossible with the standard tread to provide a rear seat of sufficient width to be comfortable for three normal-sized passengers, and rear treads had to be increased. In current American passenger-car practice—neglecting the one midget-sized car on the market—front treads vary from 55 to 60 in. and average close to 57 in., while rear treads vary from 54 to 66 in. and average close to 60 in. One thing in favor of a wide tread is that it increases the stability of the car.

In Central and Western Europe, where badly rutted roads never had to be considered by designers, the majority of passenger cars have treads considerably smaller than those of American cars, and in the so-called light cars the tread sometimes is as small as 40 in. In such cars, of course, only two passengers are seated side-by-side.

In commercial vehicles front and rear treads are rarely equal. The required front tread is governed by the width of the frame, the wheel diameter, and the maximum front-wheel deflection required for steering. The rear tread is determined largely by the width of the body, and is limited by legal restrictions on maximum body or over-all widths. When wheels are fitted with dual tires, the tread is measured between points on opposite sides of the vehicle midway between the center points of ground contact on each side. In buses the front tread generally is considerably wider than the rear tread, the single front tires being made to track substantially with the outer rear tires. The majority of buses in production have front treads close to 80 and rear treads close to 70 in.

Some farm tractors have a fixed tread, while others, and especially those intended for the cultivation of different row crops, are so designed that the tread can be readily changed, to suit different spacings of the rows. Treads of tractors without means of adjustment generally fall within the range 40-70 in. Where the tread is adjustable, its minimum width ranges between 40 and 60 in., though a few tractors have even wider minimum treads. Adjustments, as a rule, are made in steps of 4 in., and in some tractors the maximum tread is 88 and even 90 in.

Wheelbases—The distance between the center points of ground contact of wheels on the same side of the car is known as the wheelbase. In six-wheeled vehicles the wheelbase is measured between the center point of ground contact of a front wheel and a point midway between the center points of ground contact of the two rear wheels on the same side.

Since the widths of different cars usually do not differ materially, the wheelbase is the principal factor determining the size of a car and its weight. With engines mounted longitudinally in front, the length of wheelbase required for a given passenger capacity depends chiefly on the number of cylinders in line. In recent years the great majority of American passenger cars have been of either six- or eight-cylinder type. Six-cylinder cars, as a rule, are built to carry six passengers (three on each seat). Wheelbases of such cars range between 111 and 125½ in. and average 119 in. Except where provision is made for extra seats, wheelbases of cars with eight-in-line engines vary from 120 to 130 in., averaging 126.5 in. Although V engines are considerably shorter than in-line engines of the same cylinder size, their wheelbases are substantially the same. A few very large eight-cylinder cars have wheelbases of between 145 and 150 in. The overall length of American passenger cars usually is substantially 1.70 times their wheelbase length.

Interurban or parlor-car buses for 25 passengers have an average wheelbase length of 190 in., and the wheelbase increases 22 in. for each additional four passengers. An increase of four in the passenger capacity implies the provision of an additional seat on each side of the center aisle. Now, the spacing of seats in motor buses is usually about 28 in.; hence, when two seats are added, the frame must be lengthened 28 in. This increase in frame length is brought about by increasing the wheelbase 22 in. and the combined overhangs of the frame over the front and rear axles 6 in. City buses designed for 24 passengers have an average wheelbase of 138 in. and the wheelbase increases 22 in. for every additional four passengers. There are naturally considerable deviations from these average values in particular cases.

Trucks generally are furnished in different lengths, to suit the particular service in which they are to be operated. The shortest are the tractor trucks for use with semi-trailers. Tractor trucks of the cab-over-engine type have wheelbases ranging between 90 and 105 in., while those of conventional design have wheelbases ranging between 130 and 170 in. Other trucks usually are furnished in several different lengths of wheelbase, the maxima for different gross vehicle weights being substantially as follows: 5000 lb, 120 in.; 10,000 lb, 145 in.; 20,000 lb, 195 in.; 30,000 lb, 225 in.; 40,000 lb, 245 in. These figures refer to four-wheeled trucks of the conventional type with the driver's cab behind the engine. The longest wheelbase for COE trucks is about 10 in. less for the same g.v.w.

In farm tractors wheelbases vary in a general way with the size of the engine. As the piston displacement varies as the cube of the linear dimensions (of the engine), one might expect the wheelbase to vary as the cube root of the piston displacement, but an analysis of tractor specifications shows that it varies less rapidly, average values being given fairly accurately by the equation

$$w = 10.7\sqrt[3]{D} + 13 \text{ in.},$$

where w is the wheelbase in in. and D the piston displacement in cu in. Of course, there is no rigid connection between the piston displacement and the wheelbase, and it happens that of two tractors listed by the same manufacturer, the one with the smallest displacement has the longest wheelbase.

CHAPTER II

Power Required for Propulsion and Acceleration

The power required to propel a vehicle is made up of two factors, propelling force and speed. The propelling force, of course, is equal to the sum of the various resistances to motion. If we know the speed of the vehicle in miles per hour, we can readily calculate the distance in feet traveled in one minute, and by then multiplying the propelling force in pounds by the distance in feet traveled in one minute, and dividing by 33,000, we have the net horse power expended in propelling the vehicle. Some power is lost in overcoming friction in the driving units, including the transmission, the propeller shaft with its universal joints, and the driving axle, and this must be taken into account when determining the required engine output.

The resistance encountered by a vehicle when traveling at constant speed over a level road is made up of two items, viz., the rolling resistance and the air resistance. For a long time the rolling resistance was assumed to be independent of the speed—a practice which had some justification, especially in the case of trucks equipped with solid rubber tires, which were limited to speeds of 12 to 15 mph. It is now well established, however, that this factor varies with the speed. Extensive tests with pneumatic-tired trucks by the Public Roads Administration have shown that for smooth concrete pavement the average value of the rolling resistance is given by the expression $7.6 + 0.09V$ lb per 1000 lb, where V is the speed in mph.

Figures for the average rolling resistance on different kinds of road at three different speeds have been published by Prof. T. R. Agg of the Iowa State College Engineering Experiment Station and are reproduced in Table I.

Air Resistance—The resistance encountered by a solid body moving through a fluid medium, such as atmospheric air, is given by the equation

$$R = c\frac{W}{2g}AV^2 \text{ lb},$$

Table I—Average Unit Rolling Resistances for Various Surface Types and Conditions

Surface Type and Condition	*Unit Rolling Resistances, lb per 1000 lb, for Speeds of* 15 Mph	25 Mph	35 Mph
Concrete—best	11.0	13.5	17.5
Concrete—fair	13.5	16.0	19.5
Concrete—rough, poor	15.0	17.5	21.0
Asphaltic concrete—best	12.5	15.0	18.5
Asphaltic concrete—average	13.5	16.0	19.5
Sheet asphalt—best	11.5	14.0	17.5
Sheet asphalt—average	15.0	17.5	21.0
Brick, bituminous filled—average	13.0	15.5	19.0
Brick, grout filled—average	15.0	19.0	22.5
Wood block, bare of filler—average	15.0	17.0	20.0
Gravel, claybound—best	17.5	20.0	23.5
Gravel—fair to poor	25.0	27.5	31.0
Gravel—poorest	27.5	30.0	32.5
Natural soil, well-graded—good	17.5	20.0	23.5
Natural soil, slightly spongy—soft	35.0	37.5	40.0
Snow—2 in., thick and well-packed	25.0	35.0	....
Snow—4 in., slightly packed	35.0	....	....

where c is a coefficient; W, the density of the medium in lb per cu ft; g, the constant of gravity (32.16); A, the forwardly-projected area of the body in sq ft, and V, the velocity of the motion, in fps. The expression $WV^2/2g$ represents what is known as the dynamic pressure of the medium corresponding to the velocity V, in lb per sq ft, and when a flat disc is moved straight on against the air, the resistance encountered by it is only little greater than the value of that expression, the coefficient c in that case ranging between 1.00 and 1.20, depending on the size of the disc and other factors. When the air resistance of automobiles or models of same is being determined experimentally in wind tunnels, the above equation generally is made use of, but in calculating the power requirements of automobiles it is convenient to figure on a standard atmosphere (of density 0.0763 lb per cu ft) and to include the value of this factor in the coefficient. It is also more convenient to insert the velocity in mph rather than in fps. Making these changes we get for the air resistance—

$$R_a = KAV^2 \text{ lb},$$

where K is a coefficient depending on the degree of streamlining of the car's exterior; A, the forwardly-projected area in sq ft; and V, the speed of travel in mph. For a disc or plane surface the value of K ranges between 0.0028 and 0.0030, and the air-resistance coefficient of early passenger cars was of the same order. Since the early thirties the forms of passenger cars have been greatly improved from the aerodynamic standpoint, and the value of K in some cases has been reduced to as low as 0.00125. A good average value of the coefficient for modern cars is 0.0017. It is of interest in this connection that the air-resistance coefficient of a perfectly streamlined body is only about 0.0006. A body can be perfectly streamlined for one particular speed only.

Motor trucks and other commercial vehicles are not streamlined, as a rule, and their air-resistance coefficients are higher. However, few experimental data on the subject have been published. Calculations based on the test results of the Public Roads Administration referred to in the foregoing gave figures ranging from 0.00242 to 0.00249, and 0.0025 is a safe value to work with.

The sum of the rolling resistance on smooth concrete pavement and the air resistance of an average car is given by the equation

$$R_t = (7.6 + 0.09V)W + 0.0017AV^2 \text{ lb.}$$

A car traveling at V mph covers a distance of $88V$ fpm; consequently, the work done per minute in keeping the car moving at this speed is

$$[(7.6 + 0.09V)W + (0.0017AV^2)]88V \text{ ft-lb,}$$

and since 1 hp is equal to 33,000 ft-lb per minute, the horse power absorbed is

$$\frac{[(7.6 + 0.09V)W + (0.0017AV^2)]88V}{33{,}000} = \frac{(7.6V + 0.09V^2)W + 0.0017AV^3}{375}.$$

Gradeability—When a vehicle is ascending a grade it encounters an additional resistance which is generally figured as equal to the product of the vehicle weight in lb by the grade in per cent, divided by 100. Actually the force required parallel to the road surface to overcome the grade resistance is equal to the product of the vehicle weight by the sine of the angle of inclination, whereas the quotient of the grade by 100 is the tangent of that angle. But for the small angles corresponding to ordinary highway grades the sines and tangents are so nearly equal that it is quite permissible to multiply the weight directly by the grade to obtain the grade resistance.

So-called "gradeability" ratings are often applied to trucks. By "gradeability" is meant the maximum road grade which the vehicle is able to negotiate with a full rated load under specified conditions. Formerly the gradeability was generally calculated by means of the following formula

$$G = \frac{T \times r \times e}{R \times W} - 1.5 \text{ per cent},$$

where T is the maximum net engine torque in lb-ft at the speed for which the gradeability is to be determined; r, the total reduction ratio between crankshaft and drive wheels; e, the efficiency of transmission in per cent; R, the effective wheel radius in ft, and W the gross vehicle weight in lb. The 1.5 per cent in the formula represents the grade equivalent of the rolling resistance.

It will be noted that the above formula takes no account of air resistance and assumes the rolling resistance to be equal to 15 lb per 1000. At the speeds of operation of modern trucks the air resistance is not negligible, and, of course, the road resistance varies with the speed, and even more with road conditions. Moreover, in mountainous territory steep grades often must be negotiated at high altitudes where, on account of the low atmospheric pressure, the engine develops less torque and power than at sea level, so that a suitable correction factor must be applied to the engine output. The loss in horse power with altitude amounts to about 3 per cent per 1000 ft. Certified horse power curves of truck engines are furnished by manufacturers in conformity with an SAE standard. When all these factors are taken into account, the maximum grade which a truck is able to ascend with full load under specified conditions can be arrived at with greater accuracy than by the so-called "gradeability formula."

Friction Horse Power—After the engine net horse power has been ascertained, it becomes necessary to estimate or calculate the power loss due to friction in the drive, or the efficiency of the transmission. Tests conducted by the Public Roads Administration have shown that for direct drives with a ratio of up to about 7, an efficiency of about 90 per cent can be figured with, while for geared drives the efficiency reduces to about 84 per cent for an overall reduction ratio of 12, and to 80 per cent for an overall reduction ratio of 20. Recently objections have been raised to this practice of assuming the frictional losses in the drive to be directly proportional to the engine output, and these seem to have some justification, in that engines of the same size are used for trucks of widely different gross vehicle weight. The axle friction, which probably accounts for more than one half the total frictional loss, is certainly more dependent on the gross vehicle weight than on the engine output.

Chief contributors to the frictional loss in the drive are the transmission and the driving axle. The frictional moment or torque of each of these units varies with its size. Now, the size of the transmission—at least for a fixed number of gear ratios—is determined by the size of the engine, as the transmission must be large enough to safely transmit the maximum engine power for extended periods, and should be no larger. On the other hand, the best measure of the size of driving axle required is undoubtedly the gross vehicle weight. In the case of a tractor, only the maximum weight carried on the tractor wheels should be figured with.

From the foregoing discussion the conclusion is arrived at that the frictional losses in the drive are dependent on both the engine size (displacement) and the gross vehicle weight, and the author has found, from limited data available, that the total friction torque of the drive, referred to the crankshaft axis, can be represented by the equation

$$T_f = c\sqrt{wD} \text{ lb-ft},$$

where w is the gross vehicle weight expressed in units of 1000 lb; D, the piston displacement in cu in., and c a coefficient. For trucks with four-speed transmissions c is equal to about 0.300 for direct drive, about 0.375 for third speed, and about 0.400 for second speed.

Gradeability Calculation—If the gradeability is to be determined for a locality of considerable altitude, we must first apply an altitude correction factor (3 per cent per 1000 ft) to the net engine power, which latter generally can be obtained from a certified curve sheet showing the variation of the engine hp with the rpm. Next we calculate, successively, the friction horse power hp_f, the rolling-resistance horse power hp_r and the air-resistance horse power hp_a corresponding to the conditions for which the gradeability is to be ascertained, and then subtract these three items from the engine net horse power. The remainder is the horse power available for grades, hp_g. For these various calculations we need to know the speed of the vehicle corresponding to a certain engine speed. This may be found from the following equation:

$$V = \frac{N \times R}{14 \times r} \text{ mph.}$$

Values of the effective rolling radius R of different sizes of truck tire are given in Table II.

Table II—Effective Radii of Truck Tires, Ft

Tire Size	*Radius*	*Tire Size*	*Radius*
6.00/20	1.32	11.00/22	1.77
6.50/20	1.36	11.00/24	1.85
7.00/20	1.43	12.00/20	1.74
7.50/20	1.47	12.00/22	1.82
8.25/20	1.51	12.00/24	1.91
9.00/20	1.59	13.00/20	1.80
10.00/20	1.64	13.00/24	1.98
10.00/22	1.71	14.00/20	1.91
10.00/24	1.80	14.00/24	2.07
11.00/20	1.69		

The equation for the friction torque T_f has been given already, and having found its value we get the friction horse power from the equation

$$hp_f = \frac{T_f \times N}{5252}.$$

The rolling resistance R_r is determined as explained in the foregoing, and the rolling-resistance horse power then is

$$hp_r = \frac{R_r \times V}{375}.$$

The air resistance horse power is

$$hp_a = \frac{0.0025 A V^3}{375},$$

and the horse power available for grades is

$$hp_g = hp_n - hp_f - hp_r - hp_a.$$

We then obtain the value of the grade which the vehicle will be able to negotiate under the specified conditions from the equation

$$g = \frac{37{,}500 \times hp_g}{W \times V} \text{ per cent.}$$

We will now apply these equations to a truck having the following specifications: Gross vehicle weight, 20,000 lb;

engine piston displacement, 315 cu in.; horse power at governed speed, 105 at 2750 rpm; torque at governed speed, 200 lb-ft; maximum torque, 240 lb-ft at 1500 rpm; rear-axle reduction ratio, 7:1; third-speed reduction ratio in the transmission, 1.75:1; overall reduction ratio in third speed, 12.25:1; tire size, 9.00/20 (effective wheel radius, 1.59 ft); frontal area of truck, 75 sq ft.

Inserting these values in the foregoing equations we get the following results:

Truck speed at governed engine speed,

$$\frac{2750 \times 1.59}{14 \times 12.25} = 25.5 \text{ mph.}$$

For the friction torque we get

$$0.375\sqrt{20 \times 315} = 29.8 \text{ lb-ft.}$$

The friction horse power then is

$$\frac{29.8 \times 2750}{5252} = 15.6 \text{ hp.}$$

The rolling resistance on concrete pavement in first class condition at 25.5 mph would be 9.9 lb per 1000. For an average concrete pavement it probably would be about 12 lb per 1000, or 240 lb total. The rolling-resistance horse power then is

$$\frac{240 \times 25.5}{375} = 16.3 \text{ hp.}$$

For the air resistance horse power we get

$$\frac{0.0025 \times 75 \times 25.5^3}{375} = 8.29 \text{ hp.}$$

This makes the horse power available for grades at sea level

$$105 - 15.6 - 16.3 - 8.29 = 62.81.$$

At a speed of 25.5 mph this makes it possible to negotiate a grade of

$$\frac{37{,}500 \times 62.81}{20{,}000 \times 25.5} = 4.62 \text{ per cent.}$$

If we carry the same calculations through for the condition where the engine is pulled down to its speed of maximum torque, 1500 rpm, we find that the net horse power then is 68.5 and the sum of the friction, rolling, and air-resistance horse powers, 18, leaving 50.5 hp for grades. At the corresponding vehicle speed of 13.9 mph this enables the truck to negotiate a grade of 6.8 per cent.

Acceleration—More power is required to get a vehicle up to speed than to maintain it at a uniform speed under otherwise similar conditions. A car in motion has a certain amount of kinetic energy stored up, and this energy was acquired by it during the period of acceleration. The force necessary to produce (linear) acceleration is given by the equation

$$F = \frac{W}{g} a \text{ lb}$$

where W is the weight of the mass being accelerated, in lb, and a, the acceleration, in ft per sec^2. In the case of motor vehicles, complications arise from the fact that any linear acceleration of the vehicle as a whole is accompanied by angular acceleration of the road wheels, axle shafts, differential gear, propeller shaft, transmission gears, clutch, flywheel and other engine rotating parts. The rotary speeds of these parts increase and decrease with the car speed, and additional engine torque is required to overcome the inertia of these rotating parts. It will be seen from the above equation that the force necessary to produce a certain linear acceleration varies directly as the weight being accelerated. A body is accelerated angularly by applying a moment (or a torque) to it, and the moment required to produce a certain angular acceleration α varies directly as the moment of inertia of the body. The angular velocity ω is expressed in radians per second (one radian per second is equal to one revolution per second $\div 2\pi$), and the moment or torque required to produce an angular acceleration α is given by the equation

$$T = \frac{J}{g} \alpha,$$

where J is the moment of inertia of the body around its axis, in lb-ft^2; g, the acceleration of gravity, and α, the angular acceleration in radians per second^2.

Of the varions rotating masses of a motor vehicle listed in the foregoing, those ahead of the rear axle turn at a higher angular velocity than the axle, and in determining the effect of the rotating parts on the force or torque required to produce vehicle acceleration, it is convenient to "reduce" the moments of inertia of parts ahead of the axle to equivalent moments of inertia of parts rotating with the axle shafts or the road wheels. This necessitates multiplying the moments of inertia of the faster-rotating parts by the square of the axle ratio. Thus if the moment of inertia in lb-ft² of all parts rotating with the engine crankshaft is designated by J_e, that of parts rotating with the axle or road wheels by J_a, and the axle ratio by r, the equivalent moment of inertia of all rotating parts reduced to the road wheels is $r^2J_e + J_a$. This is the moment of inertia of a mass of $(r^2J_e + J_a)$ lb, located on a road wheel at a distance of 1 ft from the axis of the wheel, or of a mass of $(r^2J_e + J_a)/R^2$ lb located on the circumference of the road wheel, where R is the effective radius of the wheel in ft. As the peripheral velocity of the wheel is the same as the linear velocity of the car, the effect of the rotating parts on car acceleration is the same as if a weight $(r^2J_e + J_a)/R^2$ lb were added to the car weight.

The various moments of inertia can be determined by the pendulum method, or in the case of relatively simple parts, such as shafts and the flywheel, by calculation, but the procedure in any case is a rather involved one. The author has complete data on moments of inertia of rotating parts for a number of foreign cars of rather small size. In one, weighing 2860 lb, the moment of inertia of the rotating parts is equivalent to 11.2 per cent additional car weight as regards acceleration in direct drive, while in another, of the same weight, it is equivalent to 18.75 per cent. Both of these cars have axle ratios of 5.7, which makes the equivalent chassis weight of parts rotating with the crankshaft rather large. In the case of an American car with 4.5 axle ratio, for which some of the moments of inertia are known while others were assumed, the rotating parts were found to be equivalent to 7.5 per cent additional car weight with respect to acceleration in direct drive. For acceleration in any of the lower gears, the moments of inertia of parts rotating with the crankshaft, in addition to being multiplied by the square of the axle ratio, must be multiplied also by the square of the reduction ratio of the particular gear.

The method of determining the equivalent additional car weights of rotating parts for acceleration in first and second gear and in direct drive will be illustrated by an example. We will assume a car weighing 3500 lb having parts rotating with the crankshaft, with a moment of inertia of 12.5 lb-ft². These parts include the crankshaft, connecting-rod big ends, flywheel, clutch, and that part of the transmission which either revolves at crankshaft speed or at a fixed ratio thereto. We will also assume that the parts which revolve at axle speed (driving wheels, axle shafts and differential gear) have a moment of inertia of 160 lb-ft^2. There are some parts which, while in permanent driving connection with the rear axle, rotate at an increased speed, viz., the main drive shaft of the transmission with the gears thereon, the propeller shaft and the universal joints, but these have comparatively small moments of inertia and may be neglected.

We will assume that the axle ratio is 4.5, the ratio of the intermediate gear in the transmission, 1.62, and that of the low gear, 2.75. We will further assume that the driving wheels have an effective radius of 13.75 in. (=1.15 ft). Then for acceleration in high gear, the moment of inertia of parts rotating with the crankshaft, reduced to axle speed, is

$$12.5 \times 4.5 \times 4.5 = 253 \text{ lb-ft}^2.$$

Adding to this the moment of inertia of parts rotating at axle speed,

$$253 + 160 = 413 \text{ lb-ft}^2.$$

To convert this into equivalent car weight we divide it by the square of the effective wheel radius in ft—

$$413/(1.15 \times 1.15) = 312 \text{ lb},$$

which is equal to

$$(312 \times 100)/3500 = 9 \text{ (appr.)}$$

per cent of the car weight.

In intermediate gear the moment of inertia of parts rotating with the crankshaft, reduced to axle speed, is

$$12.5 \times 4.5 \times 4.5 \times 1.62 \times 1.62 = 664 \text{ lb-ft}^2$$

Adding the moment of inertia of parts rotating with the axle—

$$664 + 160 = 824 \text{ lb-ft}^2,$$

and dividing by the square of the effective wheel radius,

$$824/(1.15 \times 1.15) = 623 \text{ lb},$$

which is equal to

$$(623 \times 100)/3500 = 17.8 \text{ per cent of the car weight.}$$

In the same manner we find that for acceleration in low gear the rotating parts are equivalent to approximately 45 per cent additional car weight.

Engine Torque Required for Acceleration—American passenger cars are capable of a maximum acceleration of about 2.5 miles per hour per second ($=3.67$ fps^2). When accelerating in direct drive, the car considered in the foregoing example will have an equivalent weight of

$$1.09 \times 3500 = 3815 \text{ lb},$$

and the force necessary to produce an acceleration of 3.67 ft per sec^2 is

$$\frac{3815}{32.16} \times 3.67 = 435 \text{ lb}.$$

This force on the circumference of driving wheels with an effective radius of 1.15 ft corresponds to an axle torque of 500 lb-ft. As the axle ratio is 4.5, the corresponding engine torque is

$$500/4.5 = 111 \text{ lb-ft}.$$

To this, however, should be added 10 per cent to take account of frictional losses in the transmission and drive, hence the engine should produce 122 lb-ft above the torque required to overcome rolling and air resistance. Assuming the car to have a frontal area of 28 sq ft, and making an allowance of 500 lb for supplies and passengers, at 25 mph the combined rolling and air resistance would be

$$0.015 \times 4000 + 0.0017 \times 28 \times 25 \times 25 = 90 \text{ lb},$$

which corresponds to

$$(90 \times 1.15)/4.5 = 23 \text{ lb-ft engine torque}.$$

Thus the engine should be capable of delivering 145 lb-ft torque at the speed corresponding to 25 mph car speed (1370 rpm). The engine, of course, also must drive the accessories, such as the fan and generator, and its gross torque must be correspondingly higher.

Grade Equivalents of Acceleration Rates—As far as propulsive effort required is concerned, accelerating the car is equivalent to climbing a grade. If linear acceleration alone is considered, accelerating at the rate of one mile per hour per second calls for the same propelling force and the same axle torque as climbing a 4.56 per cent grade. Of course, the angular acceleration of rotating parts also must be taken into account, and if the moments of inertia and angular velocities of these parts bear the same relations to the weight and linear velocity of the car as in the foregoing example, an acceleration of one mile per hour per second in high gear calls for the same axle torque as climbing a 5 per cent grade; in intermediate gear, as climbing a 5.37 per cent grade, and in low gear, as climbing a 6.6 per cent grade. But whereas the *axle* torque required to produce a certain car acceleration is greater in low gear than in high, the *engine* torque required is less in low gear than in high. With the same transmission gear ratios as in the example, a car acceleration of one mile per hour per second in high gear calls for the same engine torque as climbing a 5 per cent grade; in intermediate gear, as climbing a 3.3 per cent grade, and in low gear, as climbing a 2.4 per cent grade.

Displacement Factor—Whereas with heavy trucks and truck-trailer combinations the ability to climb severe grades under full load is the most important performance characteristic, in the case of passenger cars the maximum acceleration in high gear is more significant. If a curve of engine net torque vs rpm is available, this can be calculated by the method outlined in the foregoing, the calculation being carried through for different car speeds in succession and a curve of maximum acceleration vs car speed then drawn. However, the acceleration of passenger cars (frequently referred to as "performance") is generally judged by the displacement factor, which is given by the equation

$$\text{Displacement factor} = 1912\,\frac{Dr}{RW}\ \text{cu in. per ton ft}$$

where D is the engine displacement in cu in.; r, the axle ratio; R, the effective rolling radius of the drive wheels, in in., and W, the loaded weight of the car (dry weight plus 500 lb). Modern American passenger cars have a displacement factor of about 38. The higher this factor the greater the acceleration and the "livelier" the car. Another unit for displacement factor is the cu ft per ton-mile. One cu in. per ton-foot is equal to 3.05 cu ft per ton-mile.

Modern automotive engines develop approximately 0.8 lb-ft maximum torque per cu in. displacement. A net torque of 0.7 lb-ft per cu in. can be figured with in making calculations of engine requirements.

Power Required by Farm Tractors—The relations between engine power, tractor weight and drawbar pull were briefly discussed in Chapter I. Farm tractors are used for a variety of purposes, and it is impossible to give data on power requirements or drawbar-pull requirements in all of these operations. Their principal use generally is in plowing, and a few figures relating to power requirements in that operation will be given.

In the Corn Belt of the Middle West the resistance to plowing varies between 5.5 and 7 psi, hence with a 14-in. bottom plowing to a depth of 6 in., a draft of from 450 to 600 lb is required per furrow. In other soils, however, the draft varies up to 20 psi. The following table gives the average draft in lb for a 14-in. plow and various depths of plowing:

Depth	6 in.	7 in.	8 in.
Sandy	250	300	340
Clover sod	590	690	785
Clay	675	785	900
Prairie sod *	1260	1470	1680
Gumbo	1680	1960	2240

* Prairie sod is seldom plowed as deep as 6 in. and the drawbar pull required is therefore seldom as much as here given.

"Rearing" Tendency of Tractors—A problem in connection with tractor operation on which there was much discussion during the early years of the industry was that of the tendency of the tractor front wheels to rise when power is applied to the drivers—the so-called "rearing" propensity of the tractor. A number of fatal accidents occurred during that period, in which tractors turned over backward. It is practically impossible for a tractor to "rear" when pulling a load through the drawbar, as the latter generally is attached below the center of the driving axle, and the drawbar pull then creates a moment around the axle, which tends to keep the front wheels down. "Rearing" usually occurred when the tractor was being moved around without load and had become stalled due to the driving wheels having gotten into a ditch or pot holes. It then sometimes occurred to the driver that by speeding up the engine to its limit and engaging the clutch quickly, an impulse could be imparted which might lift the wheels out of the depression or over the obstacle. This may have worked at times, but in other cases less torque was required to lift the front end of the tractor than to cause the drivers to turn forward, and the tractor then turned over backward.

A torque T on the driving wheels is accompanied by a torque reaction of the same magnitude which tends to turn the entire tractor around the driving axle in the direction opposite to that in which the drivers tend to turn. This torque is opposed by the moment W_1L, where W_1 is the normal load on the front wheels and L the length of wheelbase. The question now arises: What is the maximum torque that can be impressed on the drivers? The maximum continuous or steady torque evidently is given by the expression *tre*, where *t* is the maximum engine torque, in lb-in.; *r*, the reduction ratio between the engine crankshaft and the driving wheels, and *e*, the efficiency of the transmission. However, if the engine is speeded up and the clutch engaged suddenly, a much higher momentary torque can be produced by drawing on the store of kinetic energy of the flywheel and other rotating parts. In that case, what limits the torque on the drivers is the holding power of the friction clutch. Tractor clutches usually are so built or adjusted that they will begin to slip only under a torque which is substantially twice the maximum which the engine can produce continuously. Therefore, if the expedient described is resorted to, the torque on the driving wheels may reach a momentary value of $2tre$ lb-in., and if this is greater than W_1L, the tractor will "rear."

Little has been heard of tractor "rearing" accidents now for a long time. Modern tractor engines generally are provided with governors which make it impossible to "race" them, and many tractors have a platform extending back of the rear axle which, in case of "rearing," contacts the ground and prevents the tractor from turning over completely. Moreover, under conditions otherwise favorable to "rearing," rubber tires are more likely to slip than steel wheels with angle-iron lugs, and, finally, tractor operators generally are familiar with the hazard and refrain from practices that might induce "rearing."

CHAPTER III

Frames and Their Brackets

The frame of a motor-vehicle chassis is the foundation on which the powerplant and the body are carried, and which in turn is supported on the road wheels through the intermediary of axles and springs. During the early years of the industry frames were made of tubular steel, rolled steel sections, wood, and "armored wood," that is, wood sills reinforced with steel flitch plates. Pressed-steel frames were introduced on the first Mercedes model, in 1900, and their use in pasenger cars soon became general, but rolled-steel frames continued to be used for trucks for many years. The material of rolled sections actually has somewhat better physical properties than the mild steel used for pressed steel frames, but since a rolled steel frame rail necessarily has the same section throughout, whereas the bending moment it must withstand varies along its length, there is a serious waste of material in such frames, and they are rather heavy.

Historical Developments—Frame requirements for passenger cars have been greatly affected by two developments in general design. Practically all of the earlier cars were equipped with open bodies which, on account of the door openings, had little transverse and torsional rigidity. In the car the body is securely mounted on the chassis frame, and under load the two deflect as a unit, both transversely and in torsion. The open body contributed little to the stiffness of the assembly, and if the frame of an open car is insufficiently stiff, the doors are likely to jam, and the hood will not fit its ledge. Many of the earlier cars had these defects. The advent of the closed car brought about a change in this respect, because closed bodies are much more rigid, especially now that they are built with steel roofs.

The other development referred to was the adoption of flexible engine mountings. In the early cars the crankcase usually was cast with four integral supporting arms, and these were rigidly bolted to the frame rails, so that the engine formed a sturdy cross member, which made the forward end of the frame particularly resistant to torsion. Later, when the three-point support was adopted and flexible engine mountings were introduced (to prevent the transmission of engine vibration and noises to the body), this stiffening effect was lost, and trouble began to be experienced from "front-end shake." That part of the frame forward of the body had little torsional rigidity, and was prone to vibrate torsionally under the influence of road shocks. In recent years much effort has been devoted to increasing the torsional rigidity of passenger-car frames without unduly increasing their weight.

Increased torsional stiffness can be secured either by using frame members which possess considerable torsional stiffness in themselves, or by so arranging the frame members that shocks imparted to the frame through the springs are translated chiefly into bending moments, and by endowing the parts on which these moments act with adequate beam strength. Tubular members have the greatest torsional rigidity for a certain weight and "fineness ratio," while channels and I beams have the greatest beam strength. Box sections —often made up of two channels fitted one within the other and welded together—combine both torsional rigidity and beam strength in a fair degree.

General Form and Dimensions of Frames—The motor-vehicle frame comprises two side rails, a variable number of cross members, and sometimes also an X member. What may be regarded as the basic type of frame side rail has a channel section, with the open side turned inward. Side rails are made with a maximum depth at and near the middle of their length, and tapering toward their ends. Whereas in truck frames the top of each side rail usually is straight from end to end, passenger-car frames are made with a lower central section and higher front and rear sections, such frames being referred to as the double-drop type. This type of frame makes it possible to place the floor of the body close to the ground and still provide ample clearance over the axle. Sometimes frames are made with a "kick-up" over the rear axle to provide the necessary clearance for spring action.

Motor-car frames must be narrow in front, to permit of sufficient steering lock. The frames of passenger cars must be wide at the rear, because the body sills usually are bolted to brackets riveted to the outside of the frame rails, and the body is made as wide as the space between wheels permits. Front-end widths of passenger-car frames range between 32 and 36 in., this dimension being limited by the front tread, the diameter of the front wheels, and the maximum wheel deflection required for steering. Rear-end widths of passenger-car frames vary between 47 and 50 in. Truck frames are made with parallel side rails, and the width of such frames has been standardized at 34 in. by the S.A.E. Truck manufacturers usually offer their trucks in different wheelbase lengths. While there is no standard for frame lengths, what is known as the *CA* dimension, the distance from the back of the cab to the rear-axle center, has been standardized at 39, 48, 60, 72, 84, 96, 108, 120, 132, 144 and 156 in. Similar dimensions apply to six-wheel trucks, but in that case the measurement is taken from the back of the cab to a line midway between the two rear axles. For any given truck model, the frame length naturally varies with the *CA* dimension.

In earlier passenger cars the frames usually had an "insweep" immediately back of the front springs (at the dash), the narrow forward part and the wider rear part both having the side rails parallel. This had the advantage that it facilitated fitting of such attachments as spring brackets, but was open to the objection that the offset in the side rail greatly reduced its beam strength. In any portion of the side rail that makes a considerable angle with a straight line between supports, the load has an appreciable torsional component, and a channel section has little resistance to torsion. At present the side rails are made parallel only at the rear; they approach each other over a considerable portion of the frame length, and the weakening effect of this "insweep" is compensated for either by providing the frame with an X member and reinforcements, or by using a box section for the forward part of the side rail.

Frame Materials—Three grades of mild steel are being used for pressed-steel frames, the most widely used being a low-carbon steel of either 0.18 or 0.20 per cent carbon content. A steel with a slightly higher carbon content, about 0.25 per cent, which is responsive to heat treatment, is also being used to a certain extent, while alloy steels (nickel-chromium) responsive to heat treatment are used for truck frames of the better grade. All or practically all passenger-car frames are made of the low-carbon steel. This is due to the fact that with high-speed vehicles there usually is greater likelihood of trouble from excessive front-end shake (and sometimes also rear-end shake) than from frame breakage. In other words, frames are more likely to have insufficient rigidity than insufficient strength, and since the rigidity of a steel part is practically independent of the grade of steel of which it is made, the considerably higher cost of alloy-steel frames is not warranted in the case of passenger cars. Physical properties of the three grades of frame steel are given in Table I.

Table I—Physical Properties of Frame Steels

Material		*Elastic Limit, Psi*	*Reduction of Area, Per Cent*	*Elongation in 2 In., Per Cent*
S.A.E. steel No. 1020 (natural)		40,000	60	30
S.A.E. steel No. 1025	natural	45,000	55	30
	heat treated	60,000	64	24
S.A.E. steel No. 3230 (heat treated)		85,000	65	22

Frame Stresses—When a motor vehicle is at rest, the stresses to which its frame members are subjected are due chiefly to vertical bending moments. In service these stresses are modified and considerably increased as a result of road shocks. Bending moments reach their maximum values when either both front or both rear wheels pass over an obstacle together, in which case inertia or dynamic loads are superimposed on the static loads. When one wheel passes over an obstacle at speed, a shock with a large vertical component is imparted to the adjacent corner of the frame. This causes the frame to twist or deflect torsionally, and insufficient resistance to such deflecting moments has been a weakness of a good many frames in the past. When one of the front wheels runs up against a major obstruction, a nearly horizontal shock may be imparted to it, and transmitted through the axle and spring to the side rail on that side. The frame then has a tendency to "weave" or assume a rhomboidal form. This effect, of course, is even more pronounced in case of a collision where the impact is approximately in line with one of the side rails. Modern passenger-car frames with substantial X members are well able to take shocks of this kind, but, of course, no frame can be expected to withstand a major collision without damage. In truck frames proper resistance to "weaving" is ensured by substantial gussets at the junctions between side rails and cross members, and occasionally also by diagonal braces or tension members in one panel.

Properties of Frame-Rail Sections—Most frame side rails have either a channel or a box section. With the vehicle at rest, each side rail is subjected to vertical loads and acts as a beam. The stress to which any section of the rail is subjected under static conditions is directly proportional to the bending moment at that section, and inversely proportional to the section modulus. A channel section of conventional proportions is shown in Fig. 1, and with the notation there used the section modulus is

$$Z = \frac{CB^3 - [(C - t)(B - 2t)^3]}{6B} \text{ in.}^3$$

The bending moment at any section can be calculated from the various loads on the frame and their distances from the section under consideration, but such calculations are rarely made for passenger-car frames. In these cars the distribution of load with respect to the axles or points of frame support is more or less constant, and a preliminary design can be based on frame data on successful cars of similar specifications. An empirical rule for the required side-rail section modulus

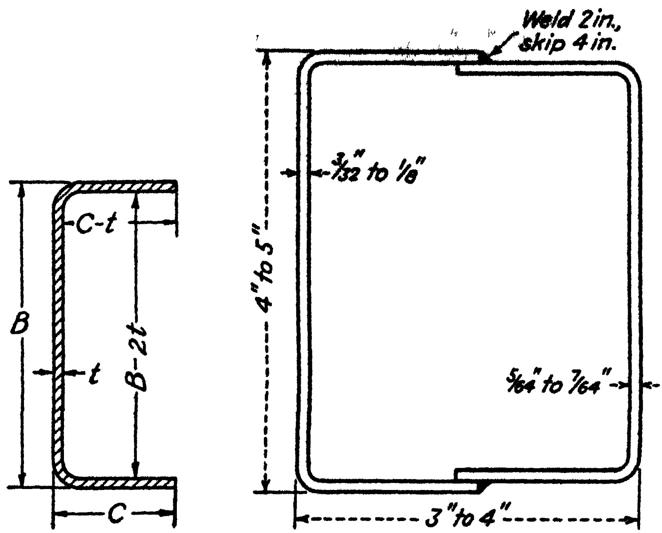


FIG. 1 (*left*).—CHANNEL SECTION OF FRAME.
FIG. 2 (*right*).—BOX SECTION OF PASSENGER-CAR FRAME.

may be derived as follows: Let l be the wheelbase in in. and W the weight of the complete car in lb. As the weight carried by the frame will be substantially proportional to W, it follows from the theory of beams that the section modulus should be proportional to the product Wl and inversely proportional to the elastic limit of the frame material. This leads to the equation

$$Z = c\frac{Wl}{E} \text{ in.}^3,$$

where c is a constant and E the elastic limit in psi. An analysis of a number of passenger-car frames with substantial X members, made some years ago, showed that c has an average value of 0.22.

In addition to channel sections, box sections are used extensively for passenger-car frames, and Fig. 2 gives the limiting dimensions for these sections. The minimum section ($4 \times 3 \times \frac{3}{32}$-$\frac{5}{64}$) has a section modulus of about 1.34 in.^{3}, and the maximum section, one of about 3.26, so that the mean value is 2.30 in.^{3} The mean value of the expression Wl for small and large passenger cars is approximately 460,000, and inserting these values in the equation for the required section modulus we get a value of 0.20 for the constant c.

In motor trucks the weight distribution with respect to the axles is quite different, and the constant in the equation for the section modulus of the side-rail therefore must be given a different value. In a fully-loaded truck the rear axle usually carries about 75 per cent of the total weight, the pay load being nearly centered over this axle. The maximum bending moment on the frame therefore is not nearly as great as it would be if all of the load were carried between front and rear axles, and the constant c in the equation can be made smaller. Its value usually ranges between 0.14 and 0.16. Frames for dump trucks are made with heavier sections, on account of the concentration of load when the dumping mechanism is operated, and the shocks received by the frame when power shovels are discharged onto the truck. In the case of trucks and trailers it is advisable to study the bending moment and the shearing force along the length of the frame, and to design the side rails accordingly. The resulting diagrams are valuable in that they show the point of maximum bending moment, and also which flange of the channel is subject to tension and which to compression.

Proportions of Channel Sections—In the design of channel-section frame rails the question of the best proportion between depth, width of flange, and thickness of stock arises. If the section modulus in itself were a safe guide to the value of the design, very light stock would be chosen, as that would reduce the weight for a given theoretical stiffness and strength. However, if the gauge is too light, the section becomes unstable. Extensive studies of actual frame failures obviously due to inadequate sections revealed that the direct cause of the failure usually was a fatigue crack in the upper flange, which is under compression. As in a symmetrical section the stresses in both flanges should be equal, and as all materials have greater compressive than tensile strengths, one would naturally expect failure to occur in the lower flange. The explanation of the seeming discrepancy is that owing to the lightness of the stock, the flange under compression buckles, and the buckling—which constantly changes as a result of road shocks—induces additional stresses of a repetitive nature, which eventually result in fatigue failure.

When a part is found to be too light for the loads it must sustain, a convenient remedy often consists in changing to a material of higher grade, which in the case of frame members would mean heat-treated alloy steel. This, however, would not reduce the buckling, since the modulus of elasticity would be no greater, though, owing to its higher elastic limit, the material probably would be able to withstand the buckling somewhat longer. A more effective and less expensive remedy is to use stock of heavier gauge, reducing the flange width to keep down the weight, thereby eliminating the buckling. When frames fail in service, a careful study of the nature of the fracture often yields information which makes it possible to eliminate the weakness without adding weight.

Passenger-Car Frames—Plans and side views of the two predominating types of passenger-car frame, the X-member type and the box-section type, are shown in Figs. 3 and 4, respectively, both being Plymouth designs. As usual in engineering matters, both types have their advantages and disadvantages. The X type excels with respect to torsional rigidity, while the box type has an advantage from the standpoint of beam strength. A box-section side rail provides a better anchorage or base for relatively long body brackets. The box section need not be as deep as an equivalent channel, and a box-section frame therefore permits of a lower floor height and a lower center of gravity. The absence of an X member in the box-section type allows the designer greater freedom in the layout of the underbody and running gear. This type of frame has gained in popularity since the introduction of full-width bodies. Manufacturers employing it usually add an X member in frames for convertible models, the bodies of which contribute little to the torsional rigidity of the assembly. As shown in Fig. 2, the two channels are welded together near the center, to minimize distortion due to welding. The weld, moreover, is not continuous, welds 2 in. long being separated by 4-in. intervals.

X Members—X members have been used in motor-car frames for many years, but originally they were essentially cross members, consisting of two channels placed back-to-back and splayed out at the ends, the primary object evidently being to prevent weaving of the frame. At present the X

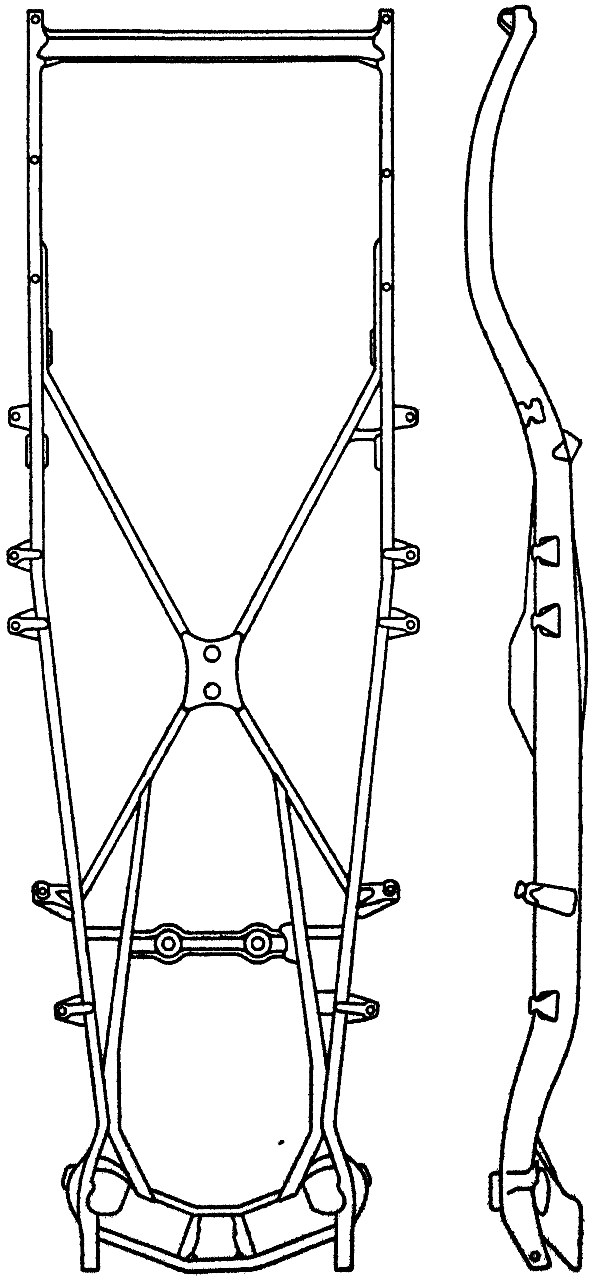


Fig. 3.—X Member Type Passenger-Car Frame (Plymouth).

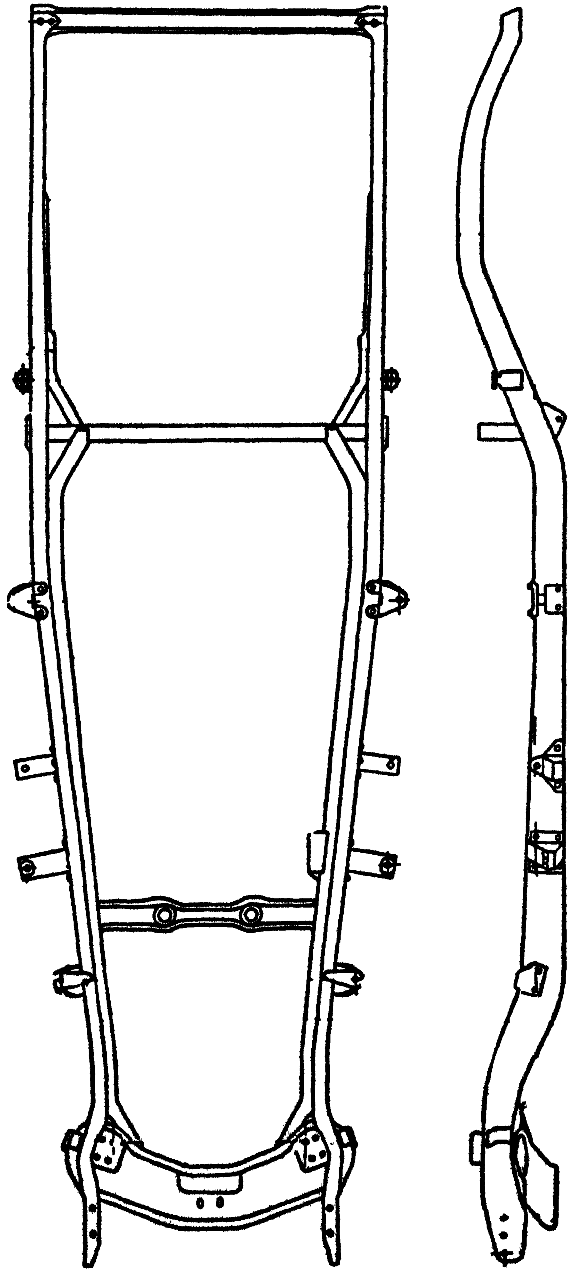


FIG. 4.—BOX-SECTION TYPE PASSENGER-CAR FRAME (PLYMOUTH).

member of a passenger-car frame is more nearly a longitudinal than a cross member, extending over substantially one-third the whole length of the frame. The A. O. Smith Company, a leading manufacturer of pressed-steel frames, developed the X member illustrated in Fig. 5. Each of the two crossed beams is an I section composed of two rolled T sections forming the flanges, and a web of flat stock. The latter, after being blanked to shape, is arc-welded to the flanges. At the center the flanges of the separate arms are arc-welded together in both the fore-and-aft and transverse directions. Diamond-shaped pressings are then placed over the joints, and arc-welded to the flanges along their edges. Round holes are punched through the webs at the center, to form an opening

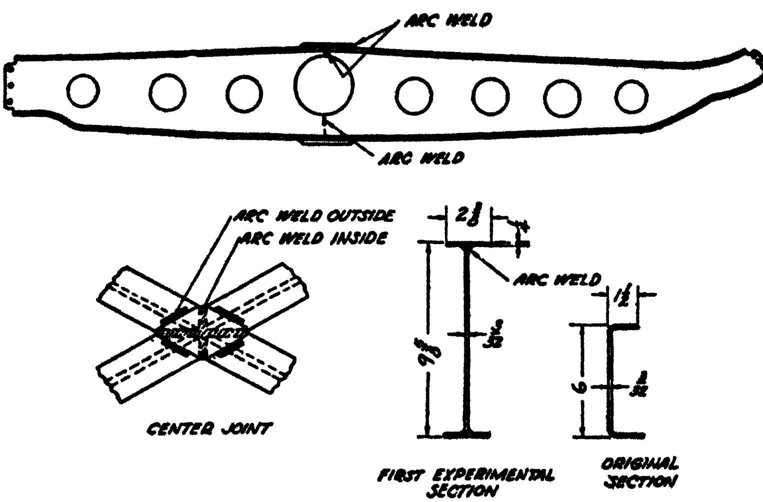


FIG. 5.—X MEMBER DESIGNED FOR MAXIMUM STIFFNESS.

through which the propeller shaft can pass, and other, smaller holes are punched in the webs along their length. To provide the greatest possible beam strength for a given weight, the section is made as deep as ground-clearance requirements will permit, and the web quite thin. For instance, for a frame having $7\frac{3}{16}$-in. side rails, the X member is 10 in. deep, and has $2\frac{3}{8}$ by $\frac{3}{16}$-in. flanges and a $\frac{5}{32}$-in. web. Each arm of such an X member may be regarded as a cantilever beam which is held rigidly at the center by the half of the X member crossing it, which latter is secured to the side rails at or near points where the body and powerplant are supported.

Frame Tests—Preliminary designs for passenger-car frames usually are based on experience with cars of similar specifications. After experimental frames have been built, road and laboratory tests are made, and if any weaknesses show up, the design is suitably modified. This is followed by the final proof-testing. Stiffness or rigidity tests may be made on the frame alone, on the body alone, and on the complete car. These tests serve to determine the linear deflection of the frame etc. under bending loads, and the angular deflection under torsion. In the bending test (as carried out in the A. O. Smith Laboratory) the frame is anchored in the vertical plane through the rear-wheel axis, its side members being clamped to pedestals with C-clamps. Half-round files are placed between the side rail and the pedestal, to permit of rotary but not of translatory motion. In bending tests of passenger-car frames the load consists of three 600-lb weights.

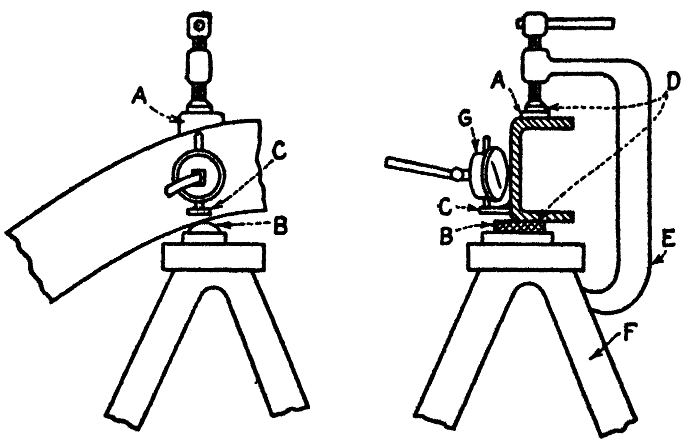


FIG. 6.—METHOD OF ANCHORING REAR END OF FRAME FOR BENDING TEST.

Two of these, representing passenger weight, are supported by adjustable bars clamped to the side-rail flanges at the locations of the seats, while the third, representing the weight of the powerplant, is placed on a triangular support, with its center of gravity in the same location as that of the powerplant. The front end of the frame is supported on rollers in the vertical plane through the front-wheel axis. Fig. 6 shows the method of anchoring the rear end, and Fig. 7 that of supporting the front end of the frame in this test.

Dial indicators showing vertical movements of the side rails are installed at substantially equal distances (12 to 18

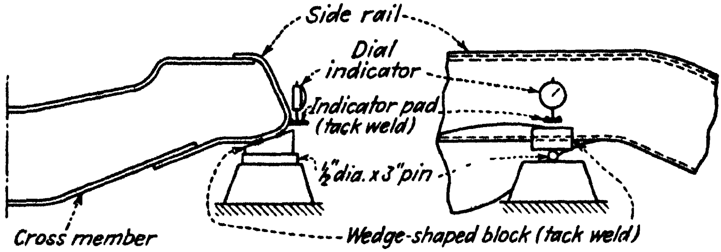


FIG. 7.—METHOD OF SUPPORTING FRONT END OF FRAME FOR BENDING TEST.

in.) between supports, and there are such indicators also at the supports. The dial stems are placed in contact with the bottom flange of the side rail, as close to the web as the rounding of the channel permits, to eliminate any influence of flange buckling on the readings.

In making a bending test, after the setup is completed, the maximum load is applied and removed, and all gauges are then set to zero, to eliminate lost motion in the setup. Next the load is applied again, and deflections at all points are recorded. In both the bending and the torsion test corrections are made for deflections of the supports and for "set." Deflec-

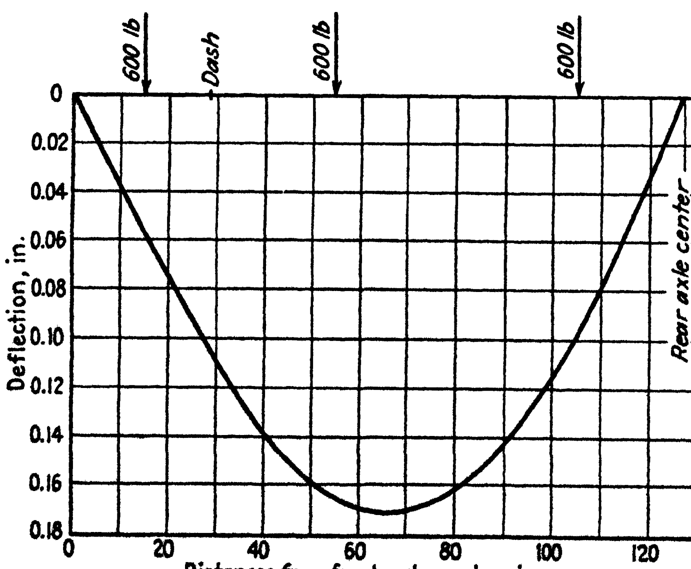


FIG. 8.—CURVE OF DEFLECTIONS IN BENDING TEST.

tions of the supports are given by readings of the dial indicators located there, when the frame is under load, while the "set" at any measuring point is the dial reading at that point after the load has been applied and then removed. Both of these values are subtracted from the frame-deflection readings under load.

Readings at corresponding measuring points on opposite sides of the frame are averaged, and from the results a curve may be plotted in which the abscissas are distances between measuring points and the ordinates deflections. Fig. 8 is such a curve.

Torsion Tests—In a test for torsional rigidity the frame is anchored to pedestals in the vertical plane through the rear-wheel axis, being clamped to the pedestals with half-round files between. The front end is supported by a roller resting on a level, flat surface (Fig. 9). In the case of a frame with a relatively rigid front cross member, such as that to which

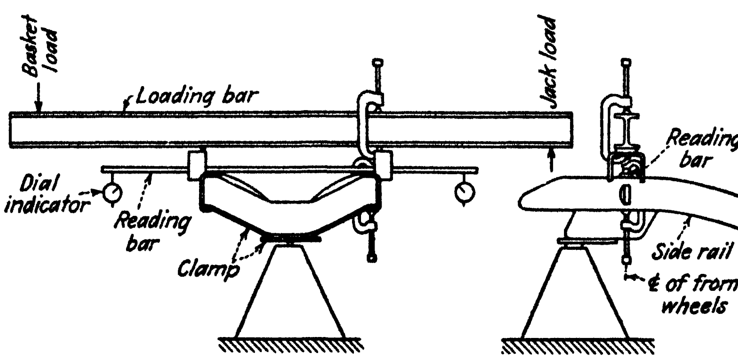


Fig. 9.—Set-Up for Torsion Test.

the bearings for the front-wheel "wishbones" are secured, the roller may support the frame at the center of that member. In other cases a rigid cross bar is clamped to the side rails to serve the same purpose. A load bar is clamped to the frame in the vertical plane through the front-wheel axis. It is made to project equally on both sides, so as not to create a torsion moment, and it is rigidly clamped to one side rail only, to prevent stiffening of the frame.

Reading bars are placed crosswise on the frame and secured to it. Dial gauges are placed under both ends of each bar, 57.29 in. apart and substantially at equal distances from the frame axis. This particular distance between gauges is chosen because (for small deflections) the difference in level between the two measuring points (the sum of the two readings) in in., is then equal to the deflection of the bar in deg. Bearing points for dial gauges are located equal distances apart along the length of the frame, including all points of support. A couple is applied to the front end of the frame by means of a weight basket on one arm of the load bar and a jack on a platform scale under the other arm. The weight in the basket is adjusted until the deflection of one side of the frame is between 0.200 and 0.250 in., and an equal moment is then applied to the other arm by means of the jack. Applied torques or couples can be varied by moving the weight basket along the load arm, to points 2.25, 2.75, 3.25 and 3.75 ft from

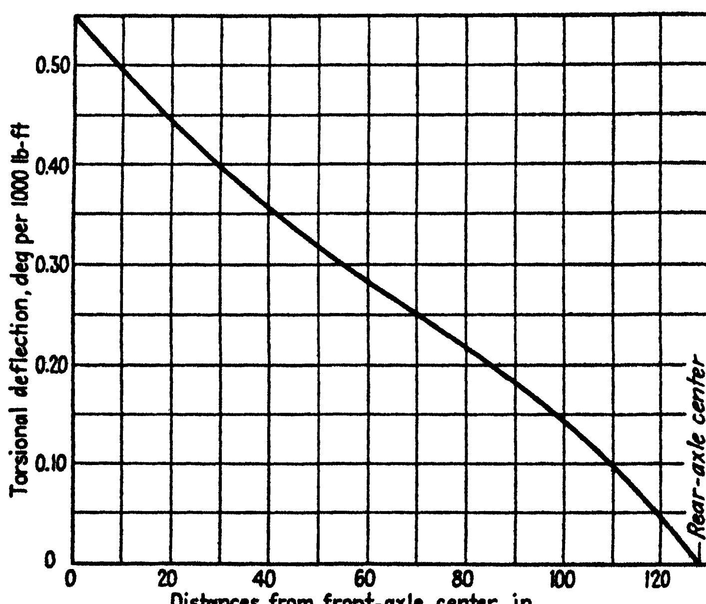


FIG. 10.—CURVE OF DEFLECTIONS UNDER TORSION.

the axis. The jack remains in the same position, 3.75 ft from the axis, and is adjusted until its moment is the same as that of the weight. In this test the load is applied in increments. The frame is subjected to torque first in one direction and then in the other, and readings for the same torque load and the same measuring point are averaged. Deflections for any given torque at any measuring point are converted to equivalent deflections under a torque of 1000 lb-ft, and the results are plotted as ordinates with the wheelbase as a base (Fig. 10).

From this graph it can be seen that the deflection increases most rapidly at the ends, where the torsional rigidity is smallest, because the side rails are of less depth there and the effect of the X member is missing.

Practically the same equipment as that here described is used to make bending and torsion tests on automobile bodies and complete cars.

Torsional-Stiffness Data—Table II, which is based on a similar table in an S.A.E. paper by D. W. Sherman of the A. O. Smith Corporation, gives stiffness data for a variety of frames, the figures applying to the section between the front and rear attachment points of the X member, and to that between the front axle and the X member. From these figures the great stiffening effect of the deep I-beam type X member is apparent. In the paper referred to it was brought out that the average torsional stiffness of American 1939 sedans between the planes through the front- and rear-wheel axes was 0.286 deg per 1000 lb-ft.

Table II—Torsional Rigidity Data for Passenger-Car Frames

No.	*Name*	*Weight*	*Deg per 1000 Lb-Ft Torque*	
			Rear of "X" to Front of "X"	*Front of "X" to Front Axle Center Line*
1	Standard X member type frame	265	0.816	0.736
2	Same as No. 1, but holes in front sub-channels closed	271	0.801	0.695
3	Same as No. 2, but front member boxed	276	0.785	0.636
4	Same as No. 1, but channel X member replaced with I-beam X member	302	0.082	0.247
5	Replacement frame for No. 1, I-beam type X member	246	0.142	0.295
6	Standard X member type frame	212.5	2.270	1.315
7	Replacement frame for No. 6, I-beam type X member	173	0.268	0.592
8	Side rails and front member from No. 6, I-beam type X member	145	0.264	0.480
9	Standard frame, tubular type	170	1.660	1.207
10	Replacement frame for No. 9, I-beam type X member	160	0.408	0.412

Calculation of Bending Moments—For a detailed study of the stresses in frame side rails it is necessary to draw a curve of bending moments for these members. Shearing forces on the side rails usually are calculated at the same time, and a graph of shearing forces is combined with the curve of bending moments on the same sheet. We will illus-

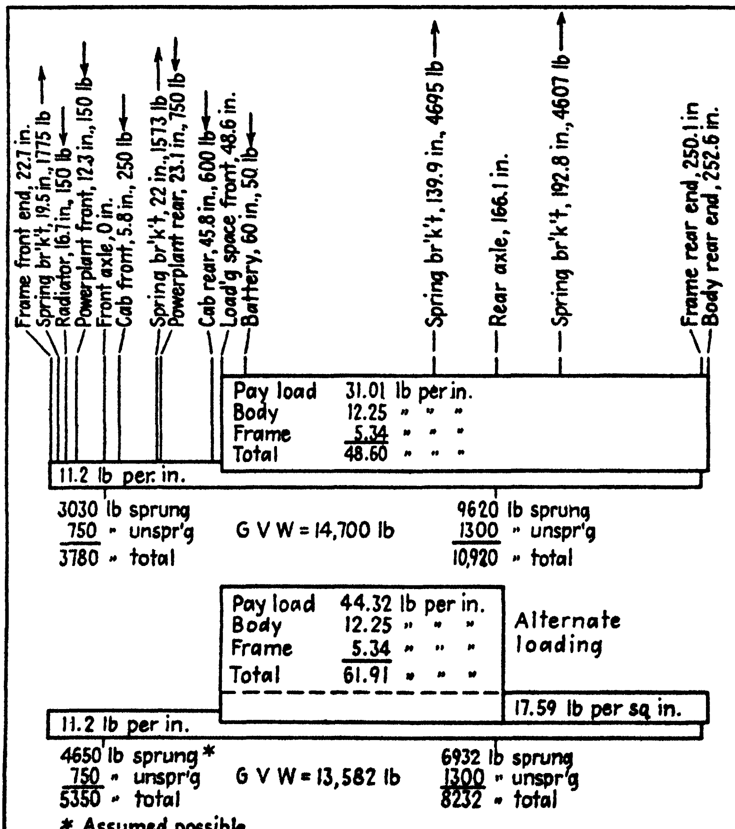


FIG. 11.—DISTRIBUTION OF FRAME LOADS FOR TWO DIFFERENT TRUCK LOADINGS.

trate the determination of bending moments and shearing forces by the example of a truck frame. A careful determination of frame stresses is more important in the case of trucks than in that of passenger cars, first because the range of loads is greater and, secondly, because weight distribution with respect to the axles is less uniform.

The truck for which the calculations will be carried through has a wheelbase of 166.1 in. and a gross vehicle weight rating of 14,700 lb. Its front springs, which are 41.5 in. long, extend 19.5 in. ahead of the axle center, while the rear springs, which are 52.9 in. long, extend 26.2 in. forward. Under full load the sprung weight on the front axle is 3348 lb and that on the rear axle 9302 lb. Unsprung weights on the two axles are 750 and 1300 lb, respectively. That part of the frame ahead of the loading space weighs 11.2 per in. of length. The total pay load of 6325 lb and the body weight of 2500 lb are assumed to be equally distributed over the length of the loading space. In the upper part of Fig. 11 is shown a diagram of loads and reactions on the frame under this loading condition, together with the values of the concentrated loads and their locations.

Bending moments are calculated for all points at which there is a concentrated load or reaction, or a change in the distributed load. The total bending moment on any section is the algebraic sum of the moments of all distributed loads and all concentrated loads and reactions to the left of that section. We will here calculate the total bending moment for one section of the frame only, that at the forward end of the loading space. The frame extends 71.3 in. ahead of this point, and as it weighs 11.2 lb per in., its weight forward of that point is 798 lb. The mean distance of this load from the section under consideration (its moment arm) is 35.6 in. We can now tabulate the various loads and reactions, and their moment arms:

Item	*Load, lb*	*Moment Arm, in.*
Frame	798	35.6
Front-spring forward eye	1775	67.1
Front-spring rear eye	1573	26.6
Radiator	150	65.3
Powerplant forward support	150	60.9
Cab forward support	250	42.8
Powerplant rear support	750	25.5
Cab rear support	600	2.8

Individual moments at the forward end of the loading space therefore are as follows:

Positive	*Negative*
$1775 \times 67.1 = 119{,}100$	$798 \times 35.6 = 28{,}400$
$1573 \times 26.6 = 41{,}850$	$150 \times 65.3 = 9{,}790$
	$150 \times 60.9 = 9{,}130$
Total 160,950	$250 \times 42.8 = 10{,}700$
	$750 \times 25.5 = 19{,}120$
	$600 \times 2.8 = 1{,}680$
	Total 78,820

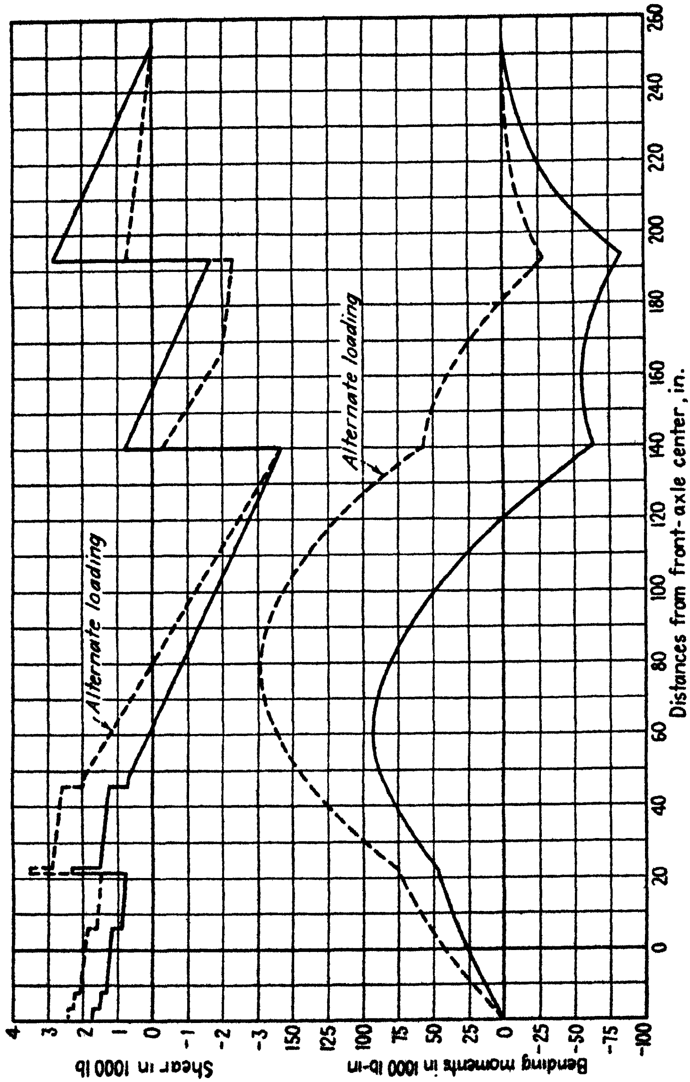


FIG. 12.—SHEARING FORCES AND BENDING MOMENTS ON TRUCK FRAME FOR TWO DIFFERENT TRUCK LOADINGS.

The net moment on the section at the forward end of the loading space, 48.6 in. back of the front-axle center, therefore is

$$160{,}950 - 78{,}820 = 82{,}130 \text{ lb-in.}$$

The bending moment at that point is positive, which means that the upper flange of the section is in compression and the lower flange in tension. The solid curve in Fig. 12 shows that under this condition of loading the bending moment is positive over approximately the forward half of the frame, and negative over the rear half. Maximum positive and negative moments are nearly equal. In calculating the data for this curve the load was assumed to be uniformly distributed over the entire length of the loading space. This is the most favorable loading condition so far as frame stresses are concerned. Sometimes a truck may carry a concentrated load (such as a heavy machine tool) substantially equal to its full rated load. Such a condition is indicated by the lower diagram in Fig. 11, where a pay load of 5207 lb is carried ahead of the rear axle. This naturally increases the load on the front axle and reduces that on the rear one.

From Fig. 12 it can be seen that the maximum bending moment on the frame under the less favorable loading condition (dashed curve) is about 174,000 lb-in. The moment on each side rail is one-half of this, or 87,000 lb-in. The frame material will have an elastic limit of about 40,000 psi, but a liberal safety factor must be allowed, because in service indeterminable dynamic loads due to road shocks are added to the static loads. If we figure on a safety factor of 4, the allowable stress is 10,000 psi, and the section modulus of the side rail required then is

$$Z = \frac{M}{S} = \frac{87{,}000}{10{,}000} = 8.7 \text{ in.}^3$$

A channel of ¼-in. stock, 9 in. deep and 3 in. wide has a section modulus of 9.32 in.^{3} and therefore should be amply strong.

Effect of Brake Application on Frame Stresses—In the foregoing calculation only the static loads on the frame were considered. In service these are added to by dynamic loads due to road shocks, and by brake application. Loads due to road shocks do not lend themselves to calculation, but the maximum loads due to brake application can be calculated with a fair degree of accuracy. For the purpose of such a calculation the friction coefficient between tire and road may be assumed to have a maximum value of 0.7. The maximum retarding force on the wheel produced by brake application is then equal to 0.7 times the wheel load (which is modified by brake application). By multiplying this retarding force by the wheel radius in in. we get the retarding moment, which latter is equal to the moment of the frame reaction. In the case of rear-wheel brakes this is taken up either on a torque tube or on the rear springs. Where there is a torque tube, the reaction occurs at the point of support of the tube on the frame, and is equal to the quotient of the retarding moment on both rear wheels by the distance of the point of support from the rear-wheel axis. In this case the force on the frame is downward, the same as that due to a load carried on it. If the torque is taken on the springs, or on so-called "wishbones" in the case of vehicles with independent front suspension, the effect of brake application is to increase the reaction at the forward spring eye (or wishbone bearing) and to reduce that at the rear.

Shearing Forces—It remains to determine the shearing forces. The shearing force at any point along a loaded beam is equal to the sum of the forces and reactions on either side of that point. This force decreases gradually for distributed loads and abruptly for concentrated loads, and it increases abruptly at points of reaction. In our example, at a point immediately ahead of the front-spring forward eye, the only load producing a shearing force is that due to the weight of a frame length of 3.2 in., which amounts to $3.2 \times 11.2 = 36$ lb. This shearing force is considered negative and plotted from the base line down. At the center of the spring bracket there is a reaction of 1775 lb, and at that point the shearing force becomes positive and attains a value of $1775 - 36 = 1739$ lb. Between this point and the center of the radiator the force is reduced by $2.8 \times 11.2 = 31$ lb, due to frame weight, and at the end of this section it is reduced by 150 lb, the weight of the radiator. Shearing forces along the length of the frame for both loading conditions are plotted in the upper part of Fig. 12. At points where the diagram shows large shearing forces it is well to check the design as to the ability of the section to withstand these forces.

Riveting Practice—There are two methods of riveting—hot and cold. An interesting difference between the results obtained with these methods is brought out by subjecting the riveted joints to a pulling test, which places the rivets under shear. With hot riveting, after the load has reached a certain definite value, there is a sudden increase in the extension, which is due to the fact that contraction of the rivet while cooling draws the surfaces of the members together with great force, and also results in the rivet hole being incompletely filled, so that the pulling force at first is resisted only by friction. With cold riveting the hole is completely filled, but the parts are not drawn together with such great force, hence the yield of the joint increases gradually throughout the load range. Owing to the lightness of the stock usually employed for frame members, which favors what is known as "oil-canning" if rivets are under tension, frames should be so designed that the rivets are subjected to practically pure shear, and under such conditions the cold-riveted joint is considered fully equal to the hot-riveted one. Cold riveting is used almost exclusively in the production of frames for American passenger cars, except where the rivets are so located that there is not enough room for the application of the somewhat bulkier equipment needed to set rivets cold. It predominates also in the production of commercial-vehicle frames, at least for rivets up to $\frac{7}{16}$ in. diameter, which is about the largest size for which cold-riveting equipment is generally available.

Channel pressings are made with inside corner radii of $\frac{3}{16}$ in. for sections less than 5 in. deep, and $\frac{1}{4}$ in. for sections of 5 in. depth and over. Rivets of $\frac{5}{16}$ and $\frac{3}{8}$ in. are used for passenger-car frames, and punched holes for these rivets are made $\frac{11}{32}$ and $\frac{13}{32}$ in. in diameter. Rivet or bolt holes should not be located in that flange of a channel section which is normally in tension, if it can possibly be avoided.

Cross Members—Most modern passenger cars have independent suspension at the front, and the front cross member of the frame then takes the place of the front axle to a certain extent. The most widely used type of front independent suspension is that employing a double wishbone linkage between the frame and the steering head, and coil springs. Wishbones on opposite sides usually make an obtuse angle with each other, which is the reason the front cross member in such cars, instead of being straight, curves forward. The front cross member must provide seats or sockets for the coil springs, and carry bearings or pivots for the lower wishbones. The upper wishbones usually form the arms of the shock absorbers, which latter are mounted on top of the cross member. Finally, this cross member must support the powerplant forward end and the radiator, and it also may have to provide a pivot support for an arm of the steering linkage. Wherever parts are mounted that are subjected to large forces or moments, the cross member must be suitably reinforced. In some designs, mounting bolts for the shock absorbers pass through both the cross member and the side rails, the junction of the two being further reinforced.

In early designs the seats for the coil springs were placed at the extreme ends of the cross member, outside the frame proper. This created an undesirable twisting moment on the side rails, and in more recent designs each side rail swings outward at the forward end, and the spring seat is located on the cross member between the side rail and a diagonal brace from the latter to it. The box-section front cross member usually has very liberal dimensions. As shown in Fig. 13, it is made in two parts, an inverted channel with the edges of the flanges turned outward, and a lower member of relatively light stock, riveted to the channel to form the box section. In designs in which the forward end of the side rail swings outward, the frame usually is narrowed slightly back

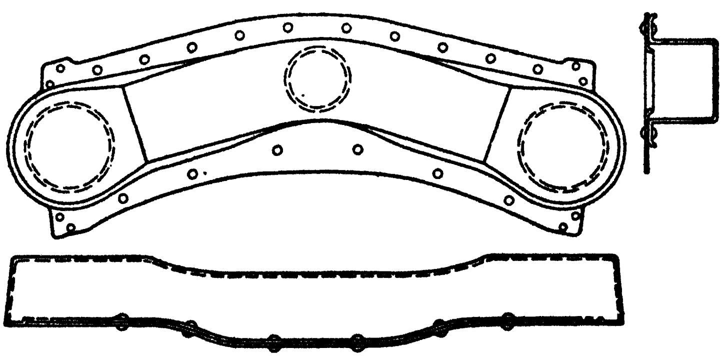


Fig. 13.—Front Cross Member for Car with Suspension on Coil Springs.

of the cross member, in order to afford the necessary steering lock. The rear of the powerplant may be supported by a cross member secured to the two forward arms of the X member, as shown in Fig. 3. At the rear end of the frame there are two cross members, as a rule, one immediately back of the rear axle, the other at the extreme rear. From these the fuel tank is suspended.

An excellent design of intermediate cross member, especially for frames without the X member, is the alligator type illustrated in Fig. 14. It is of inverted-channel section, with the edges of the flanges turned at right angles, and it offers three distinct advantages: Owing to the turned flanges it possesses great vertical stiffness and will not collapse; the large integral gussets provide good diagonal bracing, and the member can be economically produced from strip without scrap.

Frame Details—A number of frame details are illustrated in Figs. 15 to 18, relating to the frame of the Studebaker Champion. Fig 15 shows the box-section cross member on which the forward end of the engine is supported, and which has the semi-elliptic transverse front spring secured to its under side. Fig. 16 is a cutaway view of the box-section side rail immediately forward of the X member. The upwardly extending brackets on the side rail shown in Figs. 15 and 16 are for the shock absorber and the rear arm of the upper "wishbone," respectively. These parts naturally are sub-

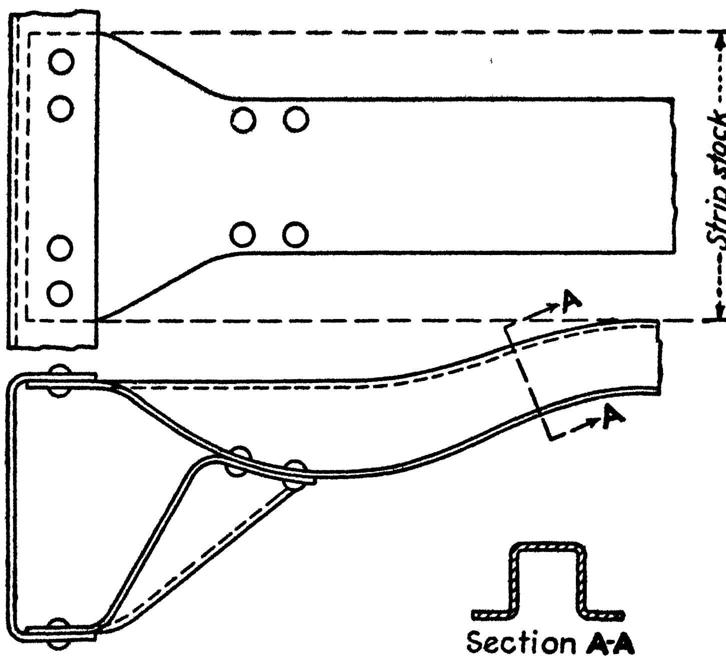


Fig. 14.—Cross Member Designed for the Economical Use of Material.

jected to considerable forces, and Fig. 16 shows how the frame section is reinforced where the bracket is mounted. Note that rivets at this point are confined to the web of the channel. This view also shows one of the body-mounting brackets on the outside of the channel. As regards the brackets for the suspension wishbone, steel tubes are welded into them, and these are provided with rubber bushings which take care of any slight misalignment. Fig. 17 shows the center section of

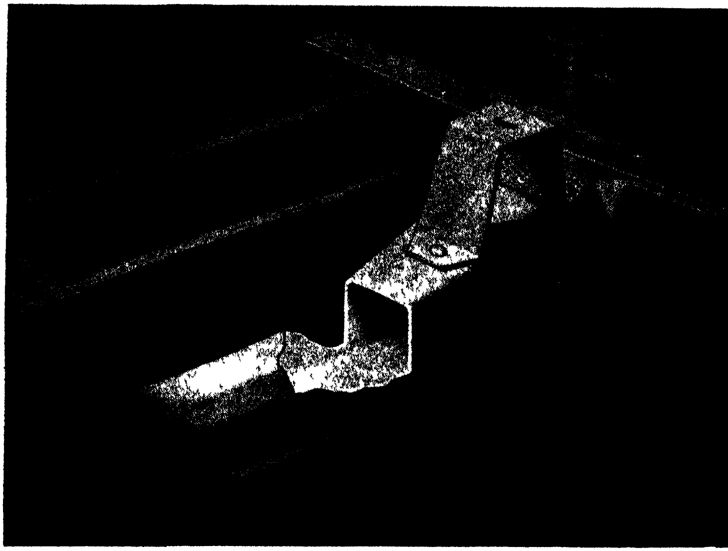


FIG. 15.—BOX-SECTION FRONT CROSS MEMBER WITH ENGINE-SUPPORT BRACKETS.

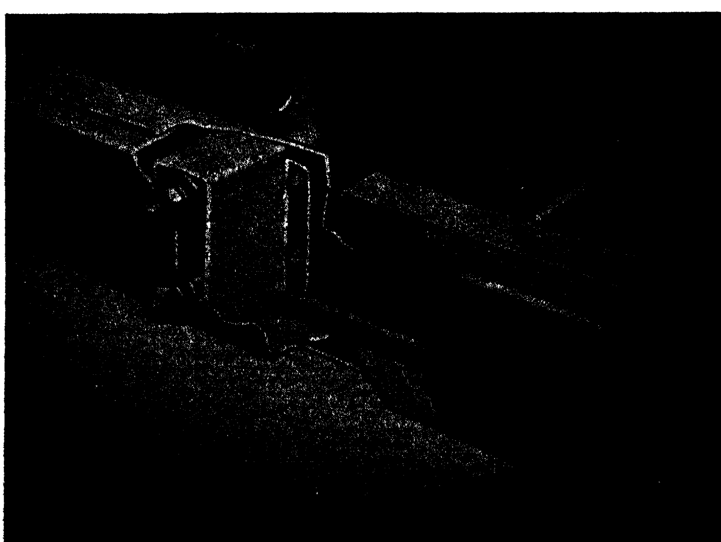


FIG. 16.—CUTAWAY VIEW OF PART OF SIDE RAIL, SHOWING BODY AND STEERING WISHBONE BRACKETS.

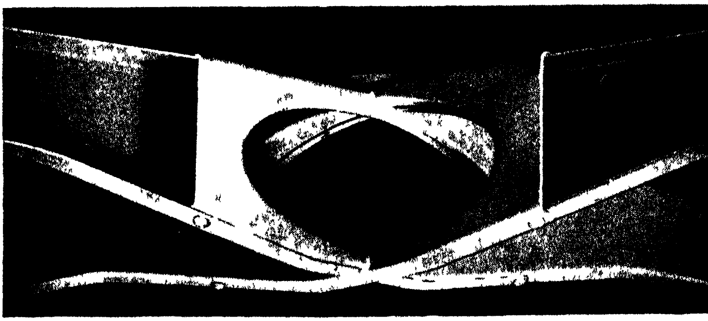


FIG. 17.—CENTER SECTION OF X MEMBER.

the X member which, while made up of channel sections exclusively, is designed to afford maximum rigidity in the vertical plane. Fig. 18 is another view of the X member center section, and shows particularly how the different parts are assembled by riveting and intermittent welding This

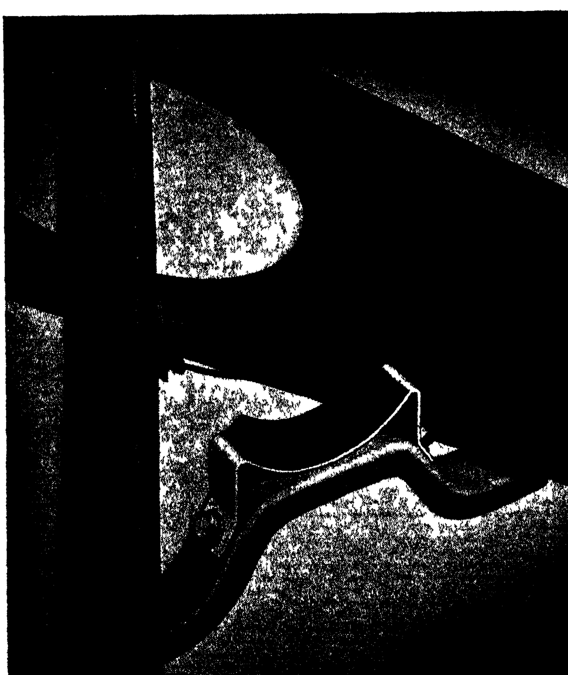


FIG. 18.—ANOTHER VIEW OF X MEMBER, SHOWING POWERPLANT MOUNTING BRACKET.

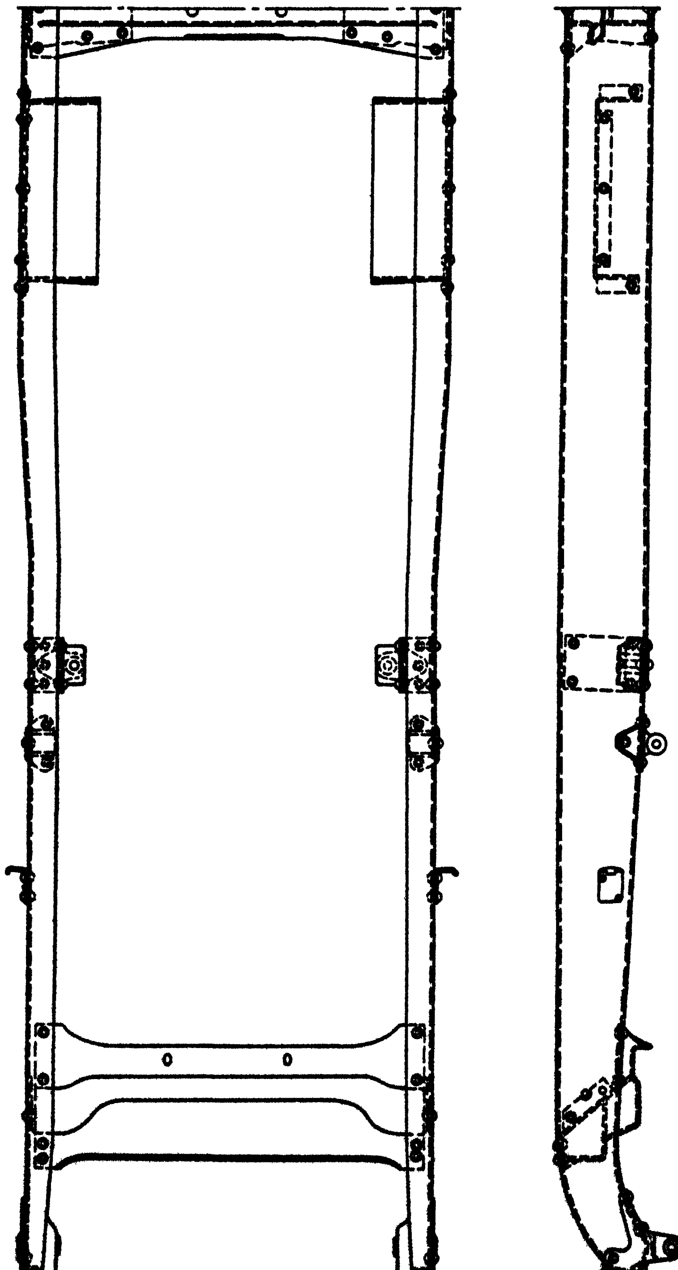


Fig. 19A.—Motor-Truck Frame, Forward Half.

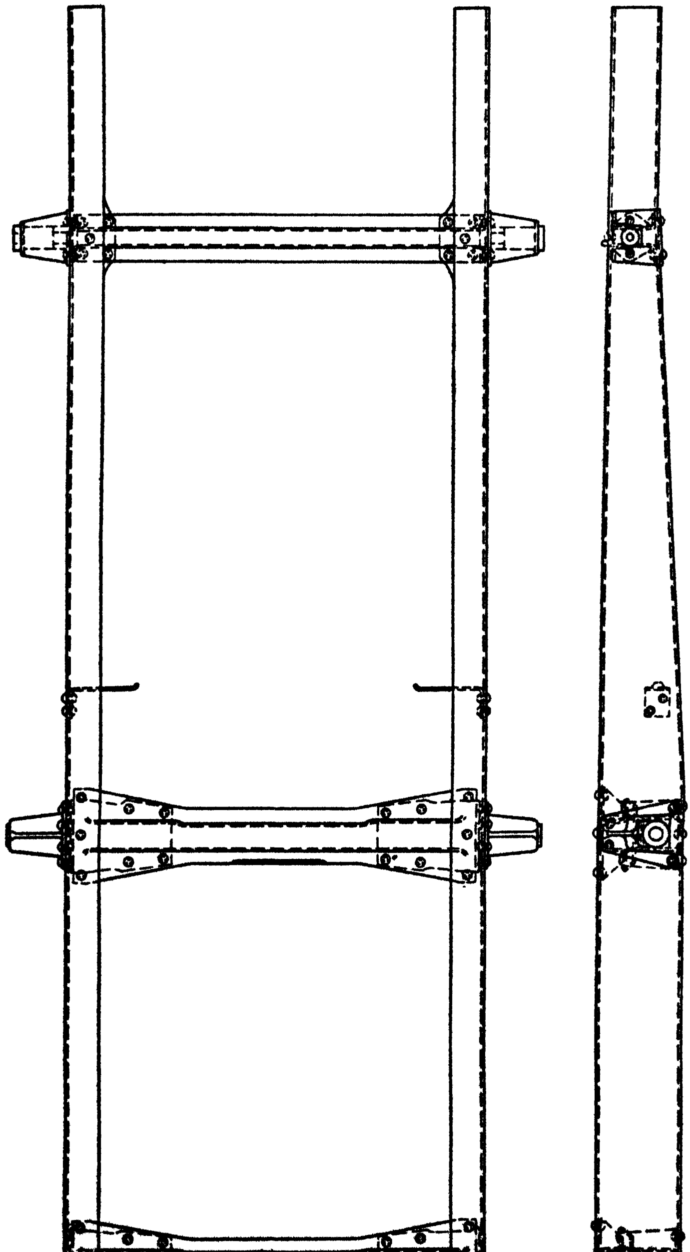


Fig. 19B.—Motor-Truck Frame, Rear Half.

view also shows the channel extending between the forward arms of the X member on which the rear of the powerplant is supported through a rubber pad.

Truck Frames—Truck frames, as already stated, are made of uniform width from end to end. The side rails and most of the cross members usually are of channel section, the depth of the side-rail section decreasing toward the ends. As the springs are located outside the frame, and therefore impose a twisting moment on the side rails through their brackets, substantial cross members are required where these brackets are located. Cross members usually are located under the radiator, at the rear of the powerplant, at the middle of the frame to support a bearing for the propeller shaft, at the rear-spring front and rear brackets, and sometimes also at the extreme rear of the frame.

A plan and a side view of a modern truck frame are shown in Figs. 19A and 19B. The two sheet-metal brackets shown secured to the side rails near the central cross member serve to support the fuel tank.

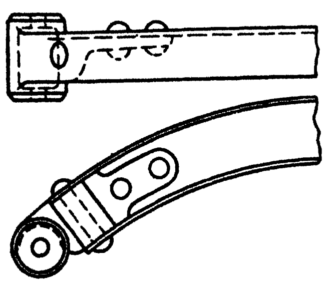


FIG. 20.—FRONT SPRING HORN.

Spring Brackets—The frame is supported by the springs through the intermediary of spring brackets. Commercial vehicles usually have semi-elliptic springs at the front, which require a bracket at each end. Formerly, when the forward end of springs and frame was exposed, the frame side rail usually curved downward at the forward end, and a spring horn of the form shown in Fig. 20 was used. At present the forward end of the chassis usually is hidden by sheet metal, and the side rails are made straight in front, hence a plain forked bracket is used, which is riveted to the web and lower flange of the side rail, being located directly under it.

The rear ends of semi-elliptic front springs are connected to the frame brackets by shackles; these shackles may be either under compression or under tension, and the brackets are designed accordingly. Simplicity of construction is in favor of shackles under compression, and these are now generally

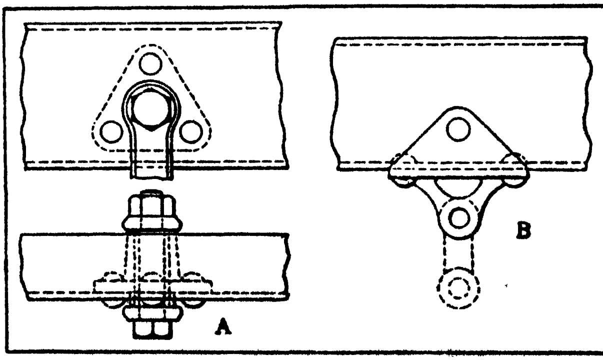


FIG. 21.—FRONT-SPRING REAR BRACKETS.

used. Fig. 21 shows two forms of front-spring rear brackets for shackles working under compression. That shown at A does not call for rivets through the flange of the channel, and therefore may be considered the better design, although at this point the stress in the side rail usually is not very high.

The rear springs at their forward ends are either pivoted or shackled to their brackets, depending on the type of drive employed. Two types of bracket for this part of the car are shown in Fig. 22. The one shown at *A* is a plain fork, and is secured to the frame by four rivets, two of which pass through the lower flange. The design shown at *B* is of the shrouded

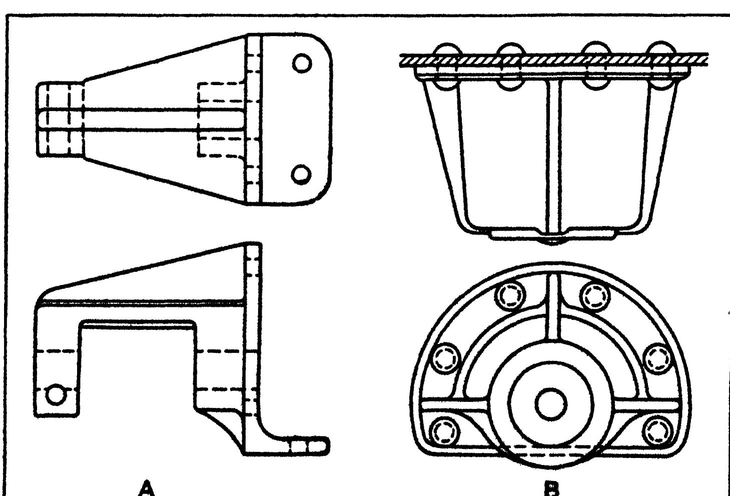


FIG. 22.—REAR-SPRING FRONT BRACKETS.

type, which is of somewhat neater appearance. The shroud tapers down from the pad or base outward, and additional rigidity is provided by ribs, one vertical and two horizontal. Spring brackets of heavy commercial vehicles occasionally are subjected to severe shocks, especially if the driving thrust and braking drag are transmitted through them, and care must be taken to ensure that they have sufficient rigidity in both the horizontal and the vertical center plane. The spring bolt can be held in position in the bracket by various means. A common practice, illustrated at *A* in Fig. 22, is to provide a transverse boss on the split outer boss for the spring bolt or pin, for a clamping bolt. With the type of bracket shown at *B* in Fig. 22, the spring pin is drilled out axially, and a retaining bolt with large washers on it is passed through.

Most spring brackets are made of malleable iron. However, steel, welded brackets also are being used, and Fig. 23

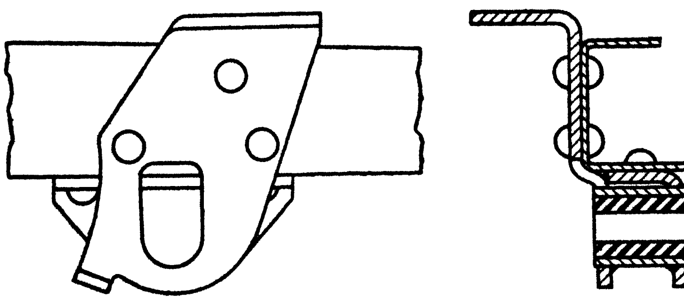


Fig. 23.—Welded-Steel, Rubber-Bushed Spring Bracket.

illustrates one such bracket designed for use on a passenger car with rubber bushings on the spring bolts. It is made of two stampings, each comprising one prong of the fork, which are welded together. Both prongs have large holes punched in them, and a steel tube designed to take the rubber bushing is welded into these holes. The bracket is riveted to both the web and the lower flange of the side rail, which at this point, near its rear end, is of comparatively small depth.

Step Hangers and Running-Board Brackets.—Step hangers for trucks are generally made of pressed steel, of the form shown in Fig. 24. When pressed steel first replaced forgings for this purpose, the hangers were given a plain channel section, with the open side down. The flanges then were subjected to compression, and as the stock used was relatively light, they had a tendency to buckle under load. To prevent buckling and make the hanger more rigid, the flanges now are

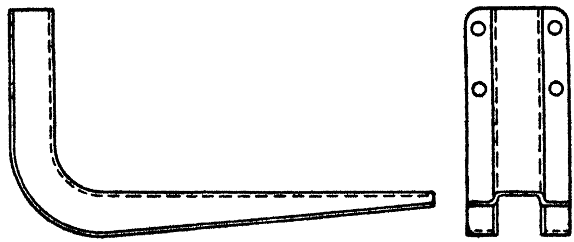


FIG. 24.—TRUCK-TYPE PRESSED-STEEL STEP HANGER.

turned over at right angles. In an alternate design the open side of the channel is turned up, and the step or running board rests on the turned-over portion of the flanges. The bending moment on the hanger is greatest at the bend, and this is sometimes made wider than the horizontal portion.

Commercial-Vehicle Bumpers—While bumpers for passenger cars and light commercial vehicles are manufactured by specialists, those for heavier trucks often are made by the truck manufacturers themselves, or by the frame makers. The object of bumpers, of course, is to take the shock of collision and transfer it directly to the frame, thus protecting frail parts at the front of the vehicle, such as lamps, radiator, radiator guard, etc. There is therefore greater variation in the design of commercial-vehicle bumpers than in that of bumpers for passenger cars. A neat and simple design of truck front bumper consists of a pressed-steel channel with the open side toward the rear, which is riveted to the ends of the side rails. (Fig. 25.)

Spare-Wheel Carrier—Road vehicles now always carry a spare wheel with an inflated tire mounted thereon. On passenger cars the spare wheel now is generally carried on the inside, where it is protected from sunlight and other deleterious influences, and where it does not add to the air resistance of the vehicle. In the case of trucks it is the usual practice to carry the spare or spares underneath the frame at the rear, a carrier of the type illustrated in Fig. 26 being provided. The wheel is held in place on the carrier by bolts passing

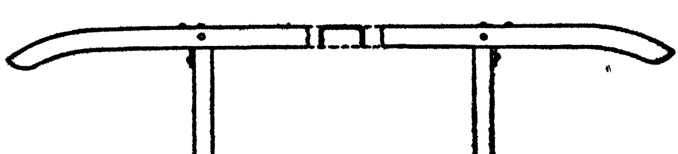


FIG. 25.—MOTOR TRUCK FRONT BUMPER.

through the same holes through which the retaining bolts pass when it is in service, by means of which it is clamped to the plate at the bottom of the carrier.

Fifth Wheels—While the fifth wheel used to couple semi-trailers to truck tractors is not a part of the frame, it is closely related thereto, one of its principal parts being supported by a bracket bolted to the tractor frame, and the other bolted or otherwise secured to the trailer frame. A fifth wheel of this type comprises two plates, generally oval in shape, the

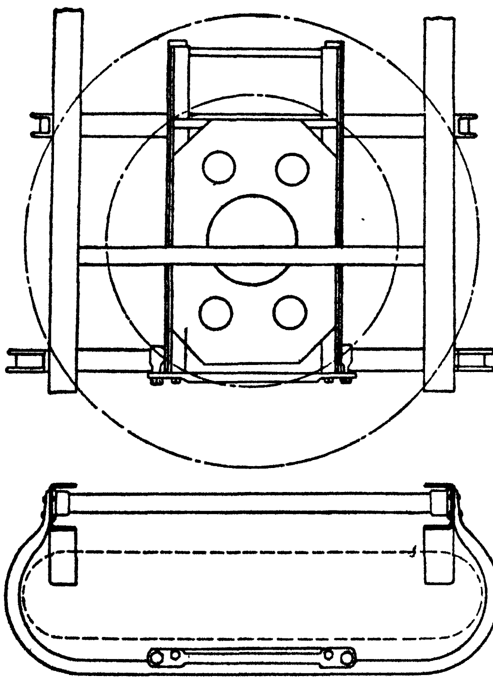


FIG. 26.—SPARE-WHEEL CARRIER FOR TRUCKS AND BUSES.

two being referred to as the lower and upper plate, respectively. One of these plates—generally the upper one—carries a kingpin at its center, while the other, which has a rocking support on the unit on which it is mounted, carries a pair of swiveled jaws adapted to grip the kingpin. The kingpin is of such design that it cannot slip out of the jaws when the latter are closed. In some designs the rocking plate is carried on its brackets through the intermediary of rubber cushions, to relieve shocks in coupling, as well as road shocks. Where such cushions are used, it is necessary to connect the metallic parts separated by the rubber by a flexible electric conductor, in order to be able to use the trailer frame as a ground return for the trailer lighting circuits. The plate which has a rocking mount is made with two pickup prongs, which provide an "inclined-plane guide" for the other plate when coupling, and a throat or "V-slot guide" for the kingpin. Most fifth wheels are made of cast steel, but pressed steel construction is also used. To provide the necessary rigidity without making the parts unduly heavy, the plates are deeply flanged and ribbed.

Fig. 27 shows a cutaway view of the under side of a fifth-wheel lower plate manufactured by the Dayton Steel Foundry Company, and Fig. 28 shows two assembly drawings of the same unit. In this design the kingpin on the upper plate (not

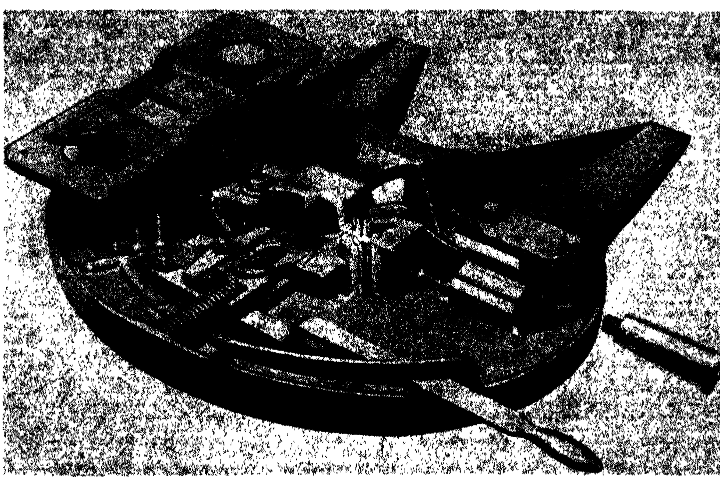


FIG. 27.—CUTAWAY BOTTOM VIEW OF DAYTON FIFTH WHEEL.

shown) is gripped between coupler jaws *A,A*, which swivel around pins *B,B*. When the trailer is coupled, the plunger *C*, under the influence of a spring clearly shown in the cutaway view, locks the jaws in position. If it is desired to uncouple the trailer, the spring is compressed and the plunger withdrawn by means of an operating lever *D* whose handle is within reach from the side of the tractor. The plunger is held in the uncoupled position by a latch *E*, which is under the influence of a tension spring and engages a projection *F* on the plunger. When the trailer is being coupled, a pin on one of the coupler jaws causes the latch to turn on its pivot, against the force of its spring, releasing the plunger and allowing it to be forced into the locking position by its spring. During the coupling operation, when the kingpin first comes in contact with the jaws, it forces them apart, and then, after it has fully entered, it causes them to close.

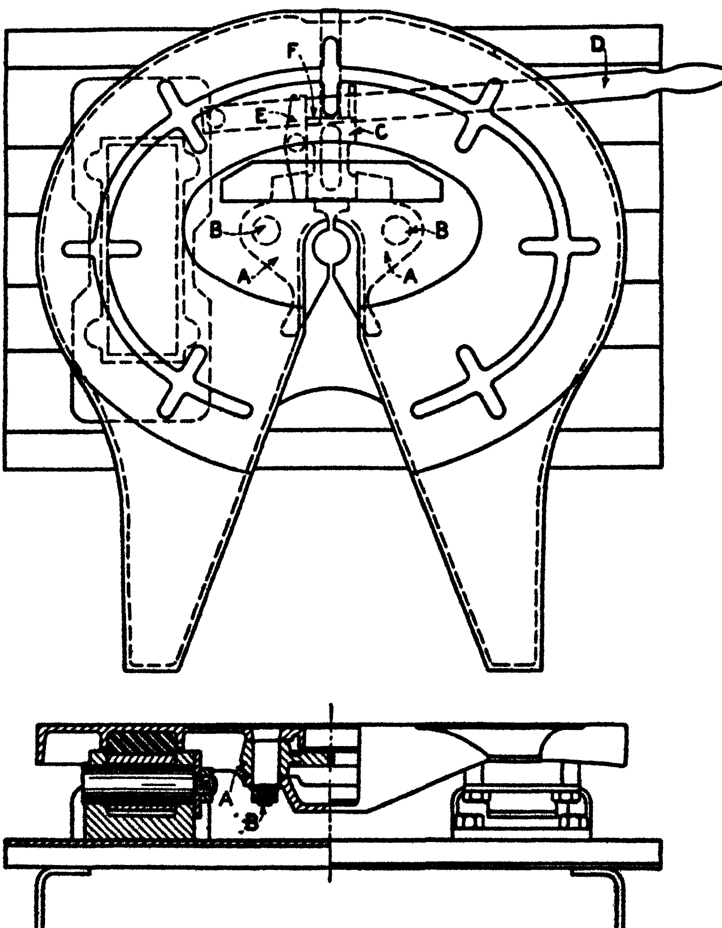


FIG. 28.—TWO VIEWS OF MARTIN-TYPE FIFTH WHEEL (DAYTON STEEL FOUNDRY COMPANY).

CHAPTER IV

Front Axles

Up to about 1930 all four-wheel motor vehicles had front axles, but since the advent of front independent suspension such axles are no longer being used for passenger cars, at least not to any extent. However, practically all commercial vehicles and four-wheel tractors and trailers have front axles.

The conventional front axle is of the divided type, comprising an axle center and two steering knuckles pivotally connected to the center. In most of the earliest motor vehicles the front axles were straight, the center portion being in line with the spindles of the steering knuckles. Later, when the engine was located in front, either it or a drop-type cross member of the frame came directly over the axle, and this necessitated "dropping" the axle at the center to prevent interference. Still later, when the center of gravity of road vehicles had to be lowered, for the sake of greater stability and safety at high speeds, the entire center portion of the axle center, including the spring pads, was dropped. These three stages in front-axle development are illustrated in Fig. 1. Axles without drop, similar to the one shown at the top in Fig. 1, are still being used where high ground clearance is required, as in farm tractors. The lowest of the three views is typical of the front axles of modern road vehicles.

Loads on Front Axles—Front axles of road vehicles are subjected to two principal types of stress—that due to the load supported, and that due to the braking moment or torque. With the vehicle at rest, the axle acts as a beam supported near its ends, directly over the center points of tire contact on the ground, and loaded at some distance from the supports—at the centers of the spring pads, or at the centers of the spring bolts in the case of cars with a transverse front spring. The load subjects the axle to a vertical bending moment which, starting at zero at each point of support, increases uniformly to the center of the spring pad or spring connection, and remains constant between spring pads, if the load due to the weight of the axle itself is neglected. When the vehicle is in motion, road shocks increase the bending moments, but the additional moments cannot be calculated directly and must be allowed for in the factor of safety.

With the vehicle in motion, the resistance encountered by the front wheels produces a small horizontal bending moment on the axle. This also starts at zero at each point of support, increases uniformly up to the center of the spring pad, or to the radius-rod attachment, and remains constant between these two points on opposite sides. Under ordinary conditions, since the traction resistance is only about 15 lb per 1000, this horizontal bending moment is only between 1 and 2 per

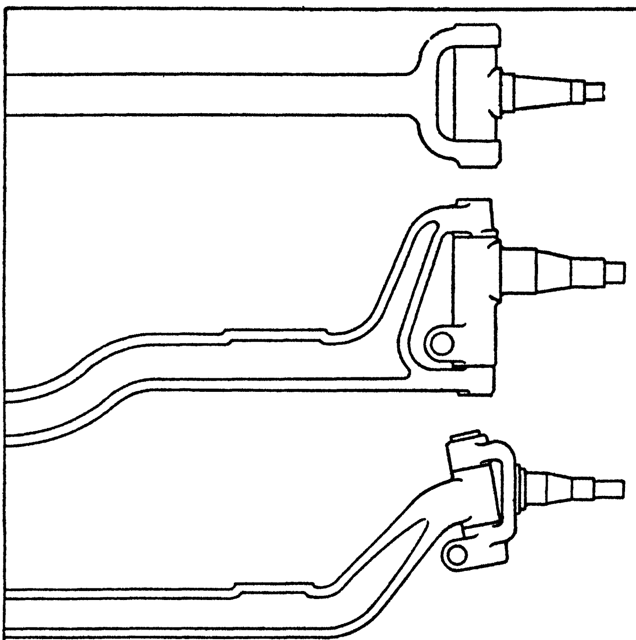


FIG. 1.—THREE FRONT-AXLE TYPES.

cent of the vertical bending moment, and is therefore negligible. It may assume considerable importance, however, if the front wheels run up against a major obstruction, such as a curb, or drop into a gulley.

In a drop-type front axle the axle center forms a sort of crank, and that portion of it directly outside the spring pad is subjected to combined bending and torsional (shearing) stresses by the resistance to motion. As shown in Fig. 2, if the resistance to motion is denoted by F and the drop of the

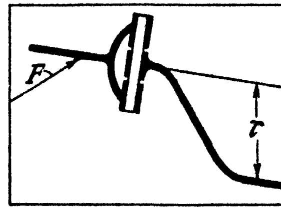


FIG. 2.—DIAGRAM OF TORSIONAL MOMENT ON DROP AXLE DUE TO TRACTION RESISTANCE.

axle by r, the torsion moment on the section of the axle just outside the spring pad is Fr.

Front-Wheel Braking Torque—When a front wheel is locked (or practically locked) by its brake, the resistance to its motion under optimum braking conditions (dry, hard road surface) is about $0.6W$, where W is the weight on the wheel. The torsion moment M on the axle center close to the steering head (Fig. 3) is then $0.6WR$, where R is the effective wheel radius. This usually is much greater than the bending moment Wl_1 at the same section, hence that part of the axle center near the steering head should be of a section offering considerable resistance to torsion.

At a section just outside the spring pad the torsional moment is less, viz., $0.6W(R - r)$, where r is the drop from the spindle axis to the center of that section. The bending moment, on the other hand, is much greater than at the steering head, and it predominates at this point. Therefore, it is advisable to change gradually from the I-section most suitable where the bending moment predominates (at the spring pad)

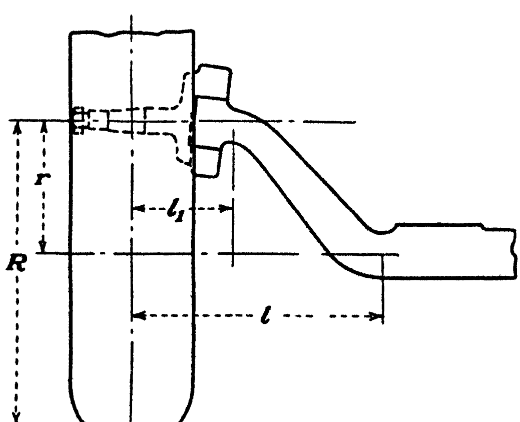


FIG. 3.—DIAGRAM OF BENDING AND TORSIONAL MOMENTS ON AXLE CENTER OUTSIDE SPRING SEAT.

to a full round, oval or rectangular section at the steering head, where the torsion moment is greatest. One way to effect this change in section is to gradually thicken the web of the I-section from the spring pad outward, so that at the steering head it becomes equal to the width of the flanges. Some designers use a full section all the way from the spring pad to the steering head, while others let the section become full at an intermediate point, as indicated in the lower view of Fig. 1.

Center Sections—Nearly all center sections of front axles are I-section drop forgings. Where vehicles are produced in small numbers, the axle centers sometimes are cast of steel or malleable iron. I-section drop-forged axles are forged from medium carbon steel—S.A.E. steel 1035 or 1040. These steels, when suitably heat-treated, have an elastic limit of 60,000 to 75,000 psi. In designing axle centers of this material, a stress of 20,000 psi can be allowed in the case of trucks for service

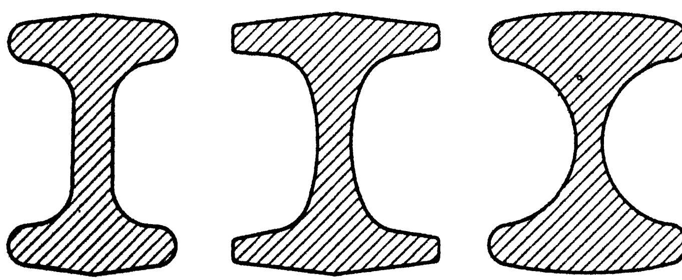


FIG. 4.—THREE SECTIONS FOR I-BEAM AXLES.

on improved highways, and 15,000 psi in the case of "off-the-road" trucks. The stress is calculated on the basis of the wheel load corresponding to the rated gross vehicle weight.

I-Sections—The form of I-section employed varies somewhat. Fig. 4 shows three different sections, all reduced to the same height. The flanges must be given about 7 degree draft so that the forging can be withdrawn from the dies. It is interesting to mention in this connection that some I-section axle centers are being produced partly by rolling. The process starts with a billet of square section, which is rolled down to an I-section. Then this part is "broken down" to obtain the necessary center drop, and finally the ends are upset and forged to shape in dies, one at a time. Rolling subjects the material to mechanical working, and is claimed to increase its strength.

In calculating the strength of I-section axle centers it is convenient to figure with an equivalent I-section with rectangular web and flanges. In Fig. 5 such an equivalent section is superimposed on the section shown at the left in Fig. 4. This gives fairly accurate results, but if a higher degree of accuracy is desired, use can be made of the method of finding the moments of inertia of complicated geometrical areas explained in textbooks on mechanics.

Referring to Fig. 5, denoting the thickness of the web by a, the width of the section is $4.25a$ and the height, slightly more than $6a$. The ends of the flanges are semi-circular, of radius $a/2$, and the fillet between web and flange has a radius a. It is well to point out here that this fillet should be made fairly large, as otherwise, if the steel should be slightly below the proper temperature while it is being worked in the dies, it is easily injured at the junction between flange and web. The thickness of the flange of the equivalent section is $1.1a$. The moment of inertia of such a section is

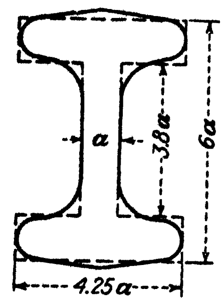


Fig. 5.

$$\frac{4.25a \times (6a)^3 - 3.25a \times (3.8a)^3}{12} = 60a^4.$$

The distance c of the outermost fiber from the neutral axis is $3a$, hence the section modulus of the equivalent section is

$$\frac{60a^4}{3a} = 20a^3.$$

The moment of inertia of this same section around its vertical axis is

$$\frac{2 \times 1.1a \times (4.25a)^3 + 3.8a \times a^3}{12} = 14.3a^4,$$

and since the distance c in this case is $2.125a$, the section modulus around the vertical axis is

$$\frac{14.3a^4}{2.125a} = 6.73a^3.$$

The section modulus is a measure of the strength of the section, and the section shown in Fig. 5 therefore is three times as strong under vertical as under horizontal bending moments.

The end portion of the axle center, from the center of the spring seat outward, may be regarded as a cantilever, and the equation of its external and internal moments is

$$Wl = SZ,$$

where W is the reaction of the ground on the wheel (equal to the load on the wheel); l, the distance from the vertical plane through the center of the wheel track to the center of the adjacent spring seat; S, the maximum stress under static load in the material of the section between spring seats, and Z the section modulus of the axle section around the horizontal axis. The following table gives, in round figures, the vertical bending moments which axles of different sizes, with sections of the form shown in Fig. 5, will sustain when stressed to 15,000 psi.

Height of section (in.).......	1½	2	2½	3	4
Allowable moment (lb-in.)...	4700	11,000	22,000	37,500	89,000

With other limiting stresses than 15,000 psi the allowable moments are in direct proportion. The allowable moment divided by the distance l between the center plane of the wheel and the adjacent spring center gives the maximum load which the wheel may carry.

Section Near Steering Head—As already mentioned, the principal stress on the axle center near the steering head is due to torsion, and reaches its maximum value when the front wheel is locked (or nearly locked) by the brake. Assuming the maximum road-adhesion coefficient to be 0.6, the tangential force on the wheel rim is $0.6W$, and the torsional moment on the section referred to then is $0.6WR$, where R is the wheel radius. Making use of the equation for the torsional strength of round shafts we have,

$$0.6WR = 0.196d^3S,$$

where d is the diameter of the section and S the allowable shear stress. If the axle is forged of either No. 1035 or 1040 steel and suitably heat-treated, a stress of 18,000 psi may safely be allowed, for even though the elastic limit in shear is materially less than that in tension, the torsional moment created when the wheel slips on a hard, dry pavement is practically the limit, whereas the bending moment may be greatly increased over its static value by road shock. The minimum diameter for a circular section then becomes

$$d = \sqrt[3]{\frac{3WR}{S}} = \sqrt[3]{\frac{WR}{6000}} \text{ in.}$$

In the case of a section of different form the torsional modulus must be the same as that of a circular section of the diameter given by the above equation. The load on the front wheel, of course, is not definitely known when the design is being prepared, but usually it can be sufficiently closely estimated from that on each front wheel of a similar existing vehicle.

Spring Pads—Drop-forged axle centers always are forged with integral spring pads. Each pad has four holes drilled through it for the spring clips. The distance apart of the centers of the holes in the direction of axle length is made equal to the width of the spring plus the diameter of the hole, and the distance apart in the other direction is made about the same. After the centers for these holes have been located, arcs are struck from these centers to radii 2.5 to 3 times the

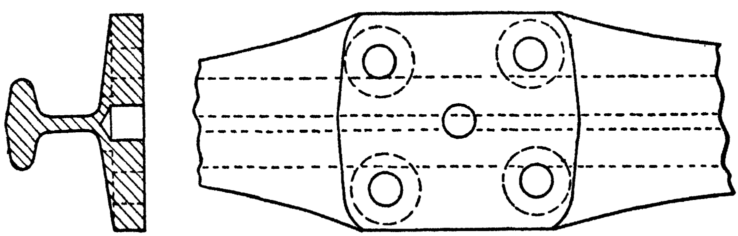


Fig. 6.—Spring Pad of Front Axle.

radius of the hole, which form the "corners" of the pad. However, in the case of axles manufactured by specialists for the market, the pads are always made considerably longer than necessary to accommodate the springs, to make the axles adaptable to different spring-center distances. Front and rear edges of the pads are made parallel, while the sides are given draft. The top of the pad is milled off to a level $\frac{1}{16}$ to $\frac{1}{8}$ in. above the highest point adjacent to the pad, and a suitable finish allowance must be made. Seats for washers on the spring clips on the bottom of the pad are finished by end-milling. The finished thickness of the pad can be made equal to the diameter of the spring-clip shank, that is, to $\sqrt{W/c}$, where W is the weight carried by the spring, and the value of c ranges from 4000 for light, fast, to 6000 for heavy vehicles. In addition to the four holes for the spring clips, a central hole is drilled in the pad to accommodate the head of the center bolt. Fig. 6 shows a typical spring pad.

Steering Heads—Previous to the advent of front-wheel brakes, the most popular type of steering head was the Elliott (Fig. 7), named after Sterling Elliott, a Massachusetts pioneer in automotive development, who patented a motor tricycle and showed this type of steering head in his patent drawings. In this type the ends of the axle center are forked. Ordinarily, the steering head is integral with the axle center, but in Fig. 7 a drop-forged steering head is shown assembled to a tubular axle center by means of rivets. The top of the wheel spindle is practically on a level with the top of the vertical part of the knuckle, which seems to be necessary in this case to provide room for the brake cylinder and mechanism above the steering head. The knuckle is forged with a flange to which the brake backing plate is secured. To clear the steering head, the backing plate in this case had to be made with an impression in its lower part. Cotter pins secure the knuckle

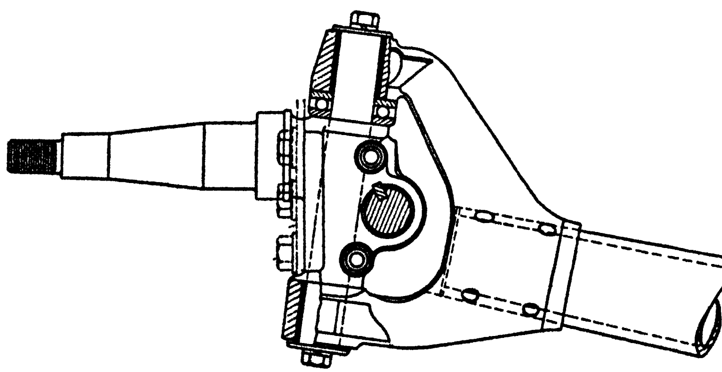


Fig. 7.—Elliott-Type Steering Head on Tubular Front Axle.

pin (sometimes called the kingpin) against both angular and axial motion in the knuckle. The pin has bushed bearings in the arms of the steering head, while end thrust (due to the weight on the front axle) is taken on a ball thrust bearing.

Reverse Elliott Head—At present the reverse Elliott type of steering head, illustrated in Fig. 8, is used almost exclusively, apparently because it facilitates the mounting of the brake backing plate on the knuckle. Where the brake drum is entirely outside the steering head, the backing plate can be passed over the knuckle spindle and fastened with screws to the shoulder at the inner end thereof. This construction sometimes is used even where the brake drum partly surrounds the steering head, the central part of the backing plate then being "cupped." More frequently, however, the backing plate is made flat, or with only a shallow depression at the center, and is fastened to lugs forged on the steering knuckle, which latter then sometimes partly surrounds or enshrouds the axle end. In the design shown in Fig. 8 a cotter pin holds the knuckle pin from turning. Wheel load is taken on a ball thrust bearing provided with a dust cap. Both of the bearings for the knuckle pin are provided with end plates to keep out dust, and with lubrication fittings to which high-pressure grease guns can be applied. The knuckle arm is bolted to the same lugs (shown dotted) on the knuckle as the brake backing plate.

Various other types of steering head have been used. Instead of being straddle-mounted, the knuckle may be substantially L-shaped and have its vertical arm carried in bearings in a housing extending either up or down from the end of the

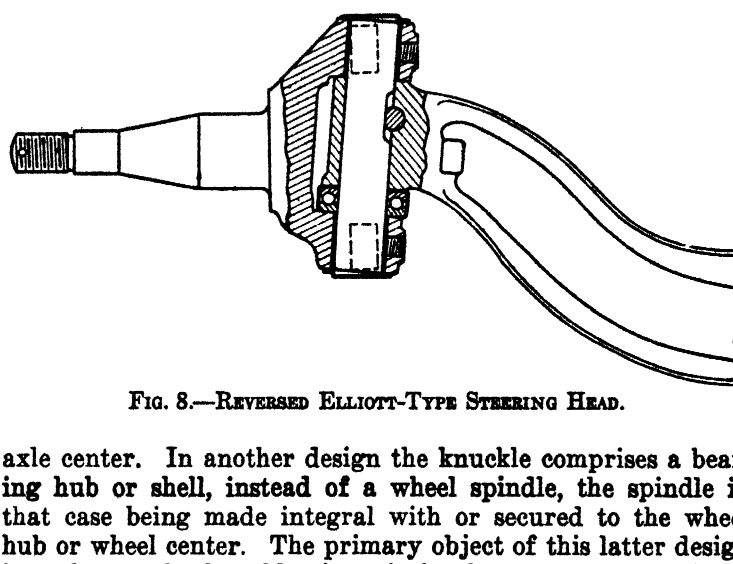


Fig. 8.—Reversed Elliott-Type Steering Head.

axle center. In another design the knuckle comprises a bearing hub or shell, instead of a wheel spindle, the spindle in that case being made integral with or secured to the wheel hub or wheel center. The primary object of this latter design is to locate the knuckle-pin axis in the center plane of the wheel, to obtain what is known as "center-point steering," which tends to reduce the transmission of road shocks to the steering wheel, and stresses in the steering mechanism. The same object is now accomplished more satisfactorily by inclining the knuckle pins.

Front-Axle Bearing Loads—The vertical load on the knuckle is substantially equal to that carried by the wheel mounted on it. This load is usually taken on a thrust bearing of the anti-friction type, to reduce the effort required to steer the vehicle. The load is a steady one, and in selecting the size of thrust bearing, one with a load rating at least 50 per cent greater than the static thrust load should be chosen.

The thrust load and the knuckle-pin-bearing loads may be calculated in terms of the wheel load as follows (Fig. 9): Let P_w represent the maximum static reaction of the wheel on the wheel spindle, which acts vertically through the center of tire-ground contact; P_t, the load on the thrust bearing, and P_u and P_l, the loads on the upper and lower knuckle-pin bearings, respectively. These are all of the forces acting on the knuckle while the vehicle is at rest, and moments taken around any given point therefore must vanish. Taking moments around point m, the center of the upper knuckle-pin bearing, we get

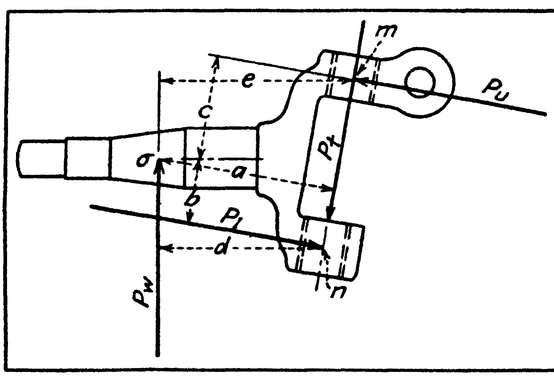


FIG. 9.—DIAGRAM OF FORCES AND REACTIONS ON STEERING KNUCKLE.

$$P_we - P_l(b + c) = 0$$

from which it follows that

$$P_l = \frac{e}{b + c} P_w.$$

When moments are taken around point n, the center of the lower knuckle-pin bearing, we get

$$P_wd - P_u(b + c) = 0$$

so that

$$P_u = \frac{d}{b + c} P_w.$$

To obtain an expression for the load on the thrust bearing in terms of the wheel load we take moments around point o, a point on the spindle axis in the center plane of the wheel:

$$P_t a - P_l b - P_u c = 0.$$

Hence,

$$P_t a = P_l b + P_u c,$$

and substituting the values of P_l and P_u found in the foregoing and simplifying we get

$$P_t = \left[\frac{be + cd}{a(b + c)}\right] P_w.$$

In Fig. 9, the lengths of the lines representing the various bearing loads are proportional to the magnitudes of these loads.

There are other loads on the knuckle-pin bearings, such as those due to the rolling resistance and to road shocks, but they are all proportional to the static load, and therefore can be provided for in setting the unit bearing load. This is usually about 600 psi in axles for light vehicles, and 1000 psi and even somewhat more in truck axles. These knuckle-pin bearings are made from 1.25 to 1.50 diameters long, relatively greater lengths being used in light axles, and from these proportions and the unit bearing loads the necessary bearing diameter can be calculated. As the load on the lower bearing is greater than that on the upper one in the proportion of e to d, Fig. 9, the lower bearing sometimes is made of proportionately larger diameter than the upper one, and the section of the knuckle pin between bearings is then tapered. Knuckle-pin bearings usually have bronze bushings, but steel-back, lead-bronze bearings and needle or quill bearings also have been used.

Inclined Steering Pivots—In cars having front-wheel brakes, the point of intersection of the pivot axis with the ground should be close to the center of the tire contact surface on the ground. In most modern cars and trucks the pivot axis is inclined, as shown in Figs. 7, 8 and 9. The distance between the point of intersection of the pivot axis with the road and the center point of tire-contact forms the lever arm through which road shocks act. The shorter this lever arm, the smaller will be the stresses imposed by road shocks on the steering gear and its linkage, and the less the fatigue induced in the driver's arms. Although the term "irreversible steering gear" has been much used, no present-day steering gears are fully irreversible, and some of the shock sustained by the road wheels is always transmitted to the steering wheel. An objection to strongly-inclined steering pivots is that in any but the straight-ahead position the front wheels are considerably inclined, in which position they are less resistant to vertical loads. Moreover, when the wheels are being turned away from the straight-ahead position, the whole front part of the car must be raised, and this increases the resistance to the steering motion.

With low-pressure tires it has been found advantageous to have the intersection of the pivot axis with the ground come a short distance inside the center line of the track, rather than to intersect it, as that makes it easier to maneuver the front wheels when the car is at rest. With so-called center-point steering, where the pivot axis passes through the center point of tire contact, the motion between tire and road when turning the steering wheel while the car is at rest is pure sliding motion, and as with low-pressure tires the area of ground contact is comparatively large, the resistance to such motion is relatively great. If the pivot axis strikes the road some distance inside the center of the track, the motion of the tire on the road surface will be partly rolling and partly sliding. Knuckle-pin inclinations in American passenger cars range between 3.5 and 8.5 deg, and average a little over 5 deg. In truck axles the inclination is preferably made such that the knuckle-pin axis strikes the road at between 2 and 2.5 in. from the center of tire contact.

Camber—Axle ends and steering knuckles are so designed that when the steering gear is set for a straight-ahead course, the knuckle-spindle axis or wheel axis is not parallel with the plane on which the vehicle stands, but inclines slightly downward from the knuckle-pin axis outward. This downward inclination is known as the camber of the axle. When the knuckle spindle deviates from the horizontal, the wheel deviates equally from the vertical, and camber therefore can be measured on the wheel, as shown in Fig. 10. It is usually expressed in degrees, the sine of the camber angle being equal to the quotient of the difference between B and A (Fig. 10) by the distance between the points of measurement on the wheel. Camber was taken over from horse-vehicle practice, in which there was a very logical reason for it. The wood wheels of horse vehicles always were strongly dished, to increase their ability to resist lateral impacts. Such a wheel forms a cone, bounded by the steel tire at its base, rather than a disc. The most favorable conditions of operation for the spokes are ensured if the spoke momentarily at the bottom (which carries most of the load) is vertical, and that it may be so, the axle must be cambered. In motor vehicles the wheels usually are not dished, and this reason for camber therefore does not exist. The only logical reason for camber in motor vehicles is that if the spindles were set absolutely horizontal, flexing of the axle under load and any wear in the knuckle bearings would cause the wheels to spread at the bottom, which is very unsightly, as it conveys the impression of a state of advanced wear and weakness.

The problem of camber is related also to the factor of road crowning. For pure rolling motion, the plane of the wheel must be perpendicular to the road surface on which the wheel rolls. Therefore, if the road is strongly crowned, the axle must be suitably cambered. However, the concrete slabs of modern roads are practically flat, and do not call for camber. While camber is still provided, and manufacturing and adjustment limits are set on it, it is usually less than 1 deg, which probably is no more than that required to compensate for the additional flexing of the axle when the vehicle is fully loaded,

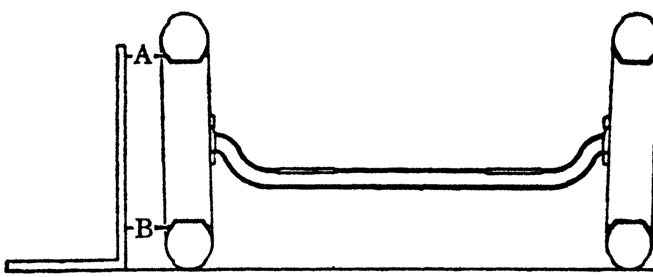


FIG. 10.—DIAGRAM ILLUSTRATING CAMBER.

and for any looseness in the knuckle-pin bearings after the vehicle has been in service for some years. In some cases the lower limit on camber specified is negative, such as minus ½ deg. In truck axles a positive camber of 1 deg is quite common.

Toe-In or Gather—When a car is at rest, and its steering gear is set in the straight-ahead position, the front wheels are not absolutely parallel with each other in the fore-and-aft direction, but closer together in front. This phase of wheel setting is known as "toe-in" or "gather." For a long time it was contended that the principal object of toe-in was to compensate for, or to counteract, certain ill effects of camber. When a pneumatic-tired wheel is set rolling by hand on a smooth surface, with its axis slightly inclined to that surface, instead of rolling straight ahead, it will roll in a circle, as indicated by Fig. 11. This, of course, is due to the fact that when the wheel is inclined, the rolling radius is shorter at one

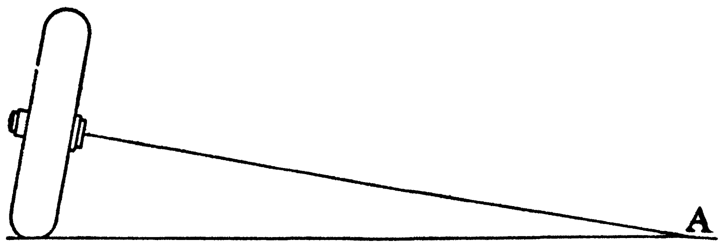
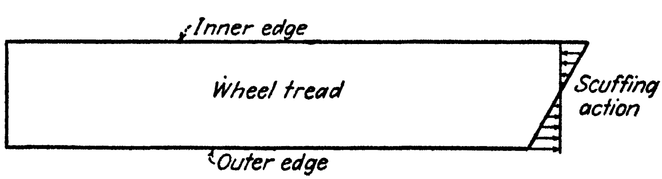


FIG. 11.—DIAGRAM INDICATING THAT CAMBERED (INCLINED) WHEEL TENDS TO ROLL IN A CIRCLE.

side of the tread than at the other. During a certain angular motion of the wheel, a point at one side of the tread moves through a smaller distance than a point at the other side, and this causes the wheel to deviate from the straight course.

There is a similar tendency for a strongly cambered vehicle wheel to deviate from the straight course, but its connection to the other wheel and the steering gear prevents it from doing so. If the vehicle is held to a straight course by the driver, the wheel will advance along a straight line, and as a point in the outer edge of the tread describes a smaller peripheral motion than a point in the inner edge, but all parts of the wheel must advance the same distance along the highway in a given time, it is clear that there must be slippage between certain parts of the tread and the road surface. Most likely the distance traveled along the road corresponds to the effective rolling radius at the center point of contact between tire and road, so that tread portions near the outer edge will slip



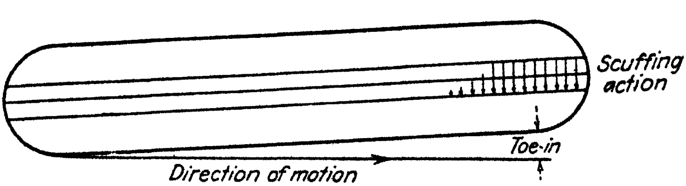


FIG. 12 (*above*).—SHOWING DIRECTION OF SCUFFING DUE TO EXCESSIVE CAMBER.

FIG. 13 (*below*).—SHOWING DIRECTION OF SCUFFING DUE TO EXCESSIVE TOE-IN.

forward and portions near the inner edge backward. This is illustrated in Fig. 12.

In the foregoing we have considered the wheel to have camber but no toe-in. Now let us consider a wheel with considerable toe-in, but no camber. The wheel can progress by pure rolling motion only in the direction of its central plane, and as the actual direction of motion deviates therefrom by the amount of the toe-in, it can continue in this direction only by a constant lateral sliding motion (scuffing), as indicated in Fig. 13. It is obvious that a lateral sliding motion (scuffing) of the tire due to toe-in cannot neutralize the bad effects of a circumferential sliding motion (scuffing) due to excessive camber. On the contrary, the ill effects of the two are likely to be cumulative.

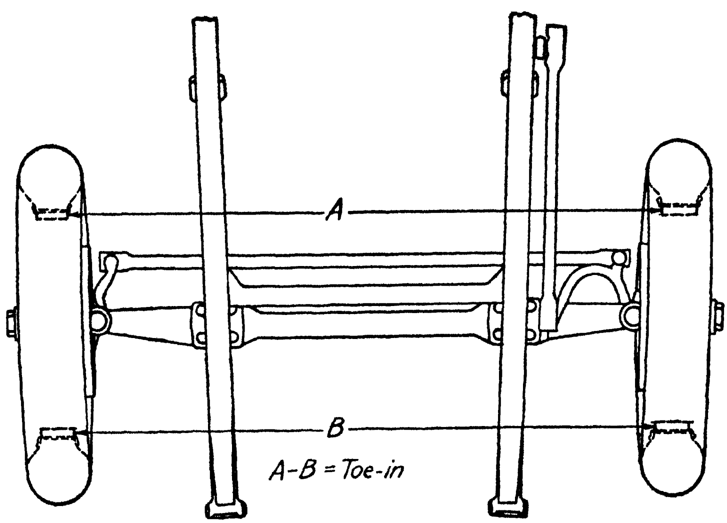


FIG. 14.—DIAGRAM ILLUSTRATING "TOE-IN."

The only sound reason for toe-in, as illustrated in Fig. 14, is that as the resistance to forward motion acts on the wheels, while the force necessary to overcome this resistance is applied to the axle through the springs or suspension links, when the car is in motion there is a tendency for the wheels to move apart in front and approach each other at the rear. If the front wheels have a slight toe-in when the car is at rest, they will be substantially parallel when the car is in motion, which is the condition ensuring minimum rolling resistance and minimum tire wear. Therefore, the toe-in should be very small, only sufficient to compensate for the spreading of the wheels in front due to the resistance to motion, and it is usually specified as either 1/16 or 1/8 in. Toe-in, of course, can be adjusted by means of the threaded joint of the connector on the tie-rod.

Wheel Caster—The knuckle pins, in addition to being inclined transversely, as already explained, are generally inclined also in the vertical fore-and-aft plane, as illustrated in Fig. 15. This inclination of the knuckle pins is known as caster. There are two distinct reasons for this practice. The first is that the combined load due to the weight on the axle and the road resistance encountered by the wheel is in a slightly inclined direction, and in the case of an I-section axle there is an advantage in making the plane of maximum axle strength coincide with the direction of the load on it. The other reason—and undoubtedly the most important one—is that this construction produces a caster effect, and tends to cause the steering wheels to "right" themselves automatically when deflected from the straight-ahead position. This effect is similar to that obtained with the front wheel of a bicycle, whereby a cyclist is enabled to ride with his hands off the handle bar. The point of wheel contact with the ground is located to the rear of the point at which the steering spindle axis produced meets the ground, hence the steering wheels trail and are automatically kept in the straight-ahead position by the road resistance. Ordinarily this trailing effect is produced by placing wedges under the front springs (W, Fig. 16).

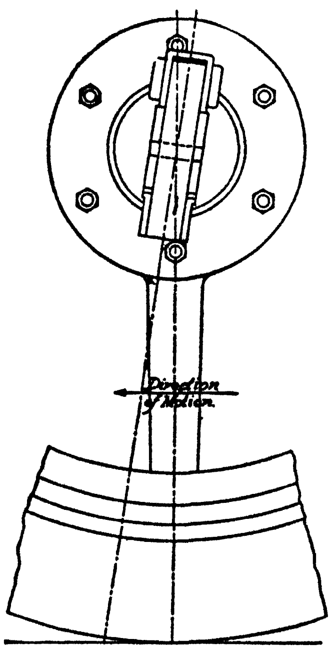


FIG. 15.—DIAGRAM ILLUSTRATING WHEEL CASTER.

The same effect can be obtained also by placing the axis of the knuckle spindle slightly to the rear of the pivot axis, and this practice has been followed abroad in the past, in connection with built-up knuckles. Owing to the more or less flexible connection of the axle to the chassis frame through the springs, the caster is not a positive figure, but varies in service with the rolling resistance encountered and with brake application. In vehicles with rigid front axles the caster is usually specified as 3° min. and 7° max.

Steering-Knuckle Materials—Steering knuckles and knuckle arms for high-speed or heavy-duty vehicles are always made of alloy steel. There are good reasons for choosing a material of high grade for these parts. In the first place, failure of a steering knuckle or knuckle arm while the car is traveling at high speed is always a very serious matter; and, secondly, owing to the more complicated shapes of these parts as compared with the axle center, there is greater danger of stress localization. A low nickel-chromium steel, No. 3141, and a molybdenum steel, No. 4042, are used for these parts. A 3½ per cent nickel steel, No. 2330, is also suitable for the purpose.

Spindle Design—Front wheels always are carried on two antifriction bearings, and the design usually is so arranged that the center plane of the wheel is much closer to the center

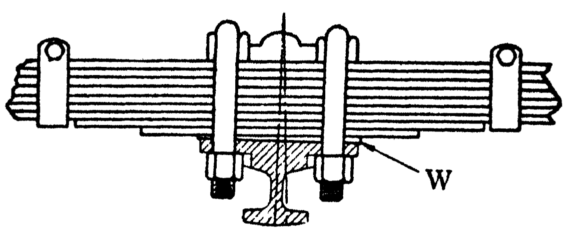


Fig. 16.—Caster Produced by Inserting Wedge Between Spring and Seat.

of the inner than to that of the outer bearing. The reason for this is, of course, that it is desired to bring the wheel as close as possible to the knuckle-pin axis, to reduce the stress in the steering linkage and the steering effort required. The static loads on the bearings are easily determined. The two bearings together carry the entire load supported by the wheel, and the load is divided between them in the inverse proportion to the distances of their respective centers from the center plane of the wheel. The static bearing loads having been determined, bearings of suitable capacity can be selected, and the bores of these bearings determine the spindle diameters at the bearing seats. The way knuckle spindles are usually proportioned, their factors of safety under static load are very high. However, the maximum stress on the spindle usually results from the wheel running up against a curbstone or other large, rigid obstruction. Owing to road shocks, the stress on the spindles varies constantly, and fatigue must be guarded against, especially at the inner shoulder, which should always be provided with a large fillet. As the standard rounding radius of antifriction bearings is quite small, it has become customary to place a spacer between the shoulder and the inner race of the adjacent bearing, the bore of which can be rounded to suit the fillet. Between the two bearing seats the spindle diameter decreases at a uniform rate, hence there are no other sudden changes in section in the stressed part of the spindle. The outer end of the spindle, which is reduced in diameter below the outer bearing seat, is threaded, and it also has a keyway cut in it. A washer with an integral key is passed over this part of the spindle, the nut is screwed up, and flaps on the washer are turned over the flats of the hexagon nut to lock the latter in position. Alternately, the nut may be locked by a cotter pin. As the end of the spindle always is protected by a hub cap or wheel cap, all that is necessary is that the locking means be secure.

Front-Wheel Bearings—In normal operation on level roads the loads on the front-wheel bearings are entirely radial, but when a corner is being turned at some speed, the centrifugal force produces strong thrust loads, for which reason combined radial and thrust bearings are widely employed. In both the United States and Great Britain, front axles of heavy commercial vehicles are usually equipped with taper roller bearings. In the United States the field of passenger-car front-wheel bearings is about equally divided between the taper-roller and the angular-contact ball types, while the front wheels of European light cars for the most part run on annular ball bearings. In the latter case the larger inner bearing has both of its races fixed endwise and takes thrust in both directions. The so-called "deep-groove" radial bearing is used for this purpose, which has considerable thrust-load capacity. The thrust-load due to centrifugal force when turning a corner is given by the equation

$$F = \frac{WV^2}{15R} \text{ lb}$$

where W is the weight on the front axle in lb; V, the speed of the car in mph, and R, the turning radius in ft. This radius should be measured to the middle of the front axle, and is therefore about 2.5 ft less than the outside turning radius in the case of passenger cars, and about 3 ft less in the case of heavy commercial vehicles. If the coefficient of adhesion between tire and road is 0.6, the centrifugal force becomes equal to the friction between tire and road (and skidding begins) when $V = 3\sqrt{R}$ mph. Except at very high speeds, the rear wheels turn in smaller circles than the front wheels, which helps to explain why they usually skid first.

With front axles equipped with either taper roller or angular-contact ball bearings, the thrust load is taken chiefly by the larger or inner bearing on the outside of the curve described. This is due to the fact that the centrifugal force virtually transfers weight from the inner to the outer wheel, so that when turning curves, the inner wheel is only lightly loaded, it skids easily, and cannot take any great amount of thrust. With taper roller or angular-contact ball bearings

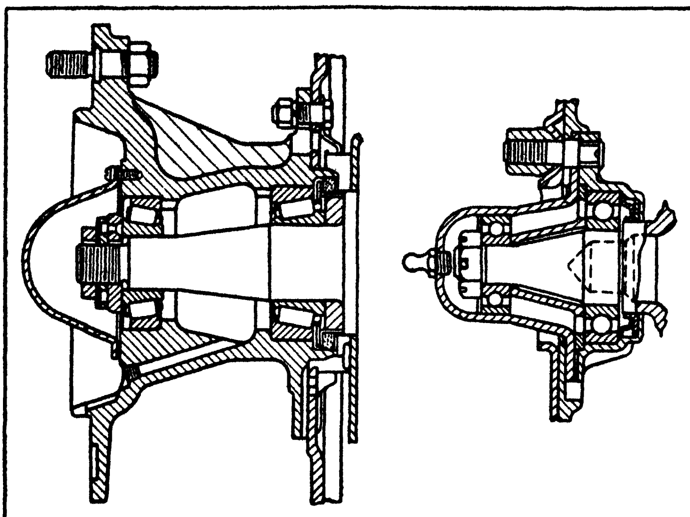


FIG. 17 (*left*).—FRONT-WHEEL HUB WITH TAPER ROLLER BEARINGS.
FIG. 18 (*right*).—FRONT-WHEEL HUB WITH ANNULAR BALL BEARINGS.

the outer races of the two bearings in each wheel are supported axially by shoulders or flanges in the wheel hub, while the inner races are supported between the shoulder at the inner and the nut at the outer end of the wheel spindle. With annular ball bearings the inner races of both and the outer race of the large one are fixed endwise, while the outer race of the small bearing is left free endwise.

In selecting bearing sizes for the front axle it is always advisable to consult with bearing manufacturers, whose experience may be of great help.

A typical bearing mounting for a commercial-vehicle front axle (Timken-Detroit) is shown in Fig. 17. It will be seen that there is a spacer between the shoulder on the knuckle spindle and the inner bearing, and an oil seal is inserted in the annular space between this spacer and the wheel hub. The inner end of the wheel hub always must be well sealed, to prevent oil or grease from getting into the brake drum. In this particular design, any oil working past the oil seal is caught by a sheet-metal ring or guard secured to the brake backing plate, and delivered into an annular space between the web of the brake drum and a pressed-steel ring secured thereto, from which space it can drain off through holes in the web of the brake drum. The wheel hub is provided with two flanges, for the brake drum and the disc wheel respectively. At the outer end it is closed by a pressed-steel hub cap secured by cap screws. Fig. 18 is a section of the front-

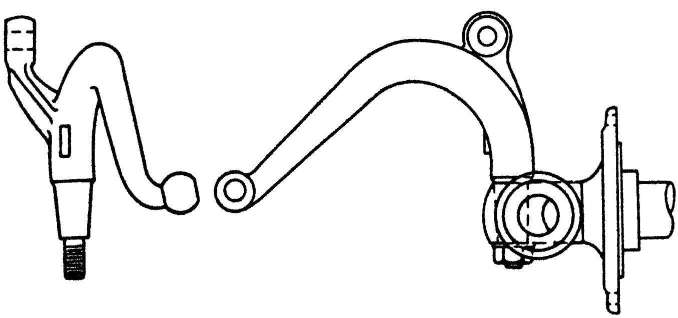


FIG. 19.—DOUBLE KNUCKLE ARM SECURED BY TAPERED JOINT WITH KEY.

wheel hub of a British light car (Austin). This hub consists of two parts, the larger one of which is a deep drawing closed at the end, so that no hub cap is needed. The two parts are held together at their flanges by the wheel-retaining studs, which also hold the brake drum in place.

Knuckle Arms—Steering knuckles are provided with arms by means of which those at opposite ends of the axle can be interconnected through a tie rod, and one of the two also must have an arm for connection to the steering arm through the drag link. Sometimes the knuckle which connects directly to the steering arm is provided with two separate arms, this practice being followed particularly in the case of reverse Elliott steering heads, one arm being secured to each of the two bearing bosses of the knuckle. More generally, however, the two arms are a single forging. As shown in Fig. 19, it is usually necessary to bend the arms out of the horizontal plane of their connection to the knuckle, because the tie rod must pass underneath the chassis springs, and the drag link must pass either over or under the axle center. Moreover, the arm to which the drag link connects must be given a considerable curvature in the horizontal plane, because with the steering gear in the straight-ahead position it must extend substantially parallel with the axle, and therefore must curve around the vertical part of the steering head or the inclined part of the axle center, which it must clear for any position of the steering gear. Occasionally the knuckle arms are forged integral with the steering knuckle, especially in the case of designs intended for large-scale production, but in the majority of cases they are separate forgings which are secured to the knuckle. Two methods of fitting the knuckle arms to the knuckle are in wide use. One consists in bolting and keying the arm to the knuckle, using a tapered seat. The taper is made about 1:10 and the nut is locked with a cotter pin. A key is necessary because of the crank effect, the outer end of the arm being out of the plane through the axis of the tapered joint and perpendicular to the knuckle-pin axis. The second method, illustrated in Fig. 20, consists in fastening the arm to the knuckle by means of two bolts, sometimes to lugs forged on the knuckle for attachment of the brake backing plate. With modern automotive vehicles the failure of a knuckle arm (or of a steering arm) is of quite rare occurrence, but when it does occur it is almost invariably a fatigue failure. Therefore, in the design of these parts the aim should be to provide high fatigue resistance, by avoiding sharp corners, rough surface finish, etc.

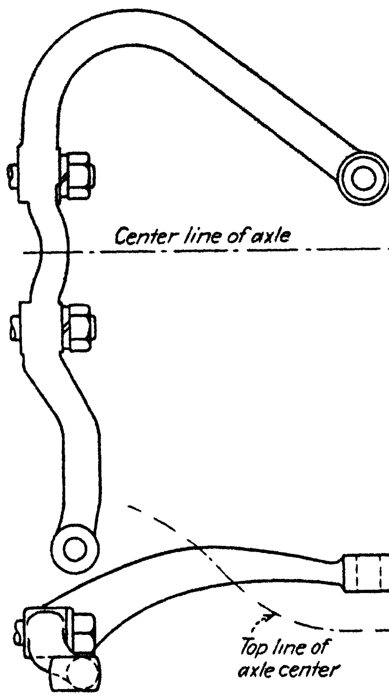


FIG. 20.—DOUBLE KNUCKLE ARM SECURED BY TWO STUDS AND NUTS.

Strength of Steering-Linkage Parts—The bending moment at the root of the steering arm and the compressive load on the tie rod and drag link depend on the moment around the knuckle-pin axis set up by lateral forces or shocks on the wheels, which reach their maximum values when the wheel strikes the curb at an angle or when the driver pulls hard on the steering wheel to get the front wheels out of deep ruts. This moment is substantially proportional to the load carried by the wheel, and to the wheel diameter. As regards the knuckle arm, the critical section is that where it projects from its socket in the knuckle. This being a circular section, its resisting moment (section modulus) is proportional to the cube of its diameter. The bending moment imposed on the knuckle is proportional to the maximum wheel load W and the wheel diameter D. Of course, at the critical section the moment is somewhat less than the maximum, in the proportion of the distance between the center of the ball at the end of the knuckle arm and the center of the section, to the distance between the center of the ball and the knuckle-pin axis. This ratio, however, will not vary much in practice, and we may therefore say that the cube of the diameter d of the critical section should vary as the product WD. We then have

$$d = \sqrt[3]{\frac{WD}{c}} \text{ in.,}$$

where c has a value of about 27,500 in the case of axles for high-speed vehicles, and up to 40,000 for the heaviest trucks.

The arm proper is generally made of oval section, of such dimensions as to have approximately the same section modulus as the big end of the tapered root near that section, and tapering down somewhat toward the free end. Drawings of typical knuckle arms are shown in Figs. 19 and 20. The knuckle arms for the tie-rod always extend back from the knuckle. One reason for this is that the tie-rod is better protected against injury by road obstructions back of the axle, and another that if it were in front of the axle the free end of the knuckle arm would have to approach the wheel to give correct steering, and there would generally be interference with the brake backing plate.

Tie-Rod—The tie-rod, which connects the knuckles at opposite ends of the axle, if located behind the axle, acts as a strut or column, for which reason it is made of tubular stock. The maximum compression load on the rod may be represented by the expression

$$P = \frac{cWD}{b} \text{ lb},$$

where c is a constant; W, the maximum load carried by one front wheel; D, the diameter of the wheel, and b, the length of the knuckle arm to which the tie-rod connects. According to Rankine, the maximum safe load for a column is

$$F = \frac{SA}{1 + q\frac{l^2}{r^2}} \text{ lb},$$

where S is the maximum safe stress for the material in compression; A, the area of the section; q, a constant depending on the condition of the ends of the column (whether fixed or free) and on the material; l, the length of the column, and r, the least radius of gyration of its section. In the case of tie-rods the value of $q(l^2/r^2)$ is always materially greater than unity, and it is therefore permissible to omit the "1" from the denominator in the above equation for the safe load of the column. By equating the expression for the load on the tie-rod and that for the load which the rod, acting as a strut, will sustain, assuming a tubular tie rod of outside diameter D and inside diameter d, and inserting the value of the least radius of gyration in terms of D and d, $(D^2 + d^2)/16$, we arrive at the equation

$$D^4 - d^4 = \frac{WDl^2}{CSb}.$$

Since regular steel tubing is always used, the permissible stress is constant, and in practice the value of the product CS ranges between 20,000,000 and 30,000,000, the lower value being used in the case of high-speed vehicles, and *vice versa.* In making use of this equation, the right-hand member can be evaluated from the known values of the design, a value for D assumed, and the required value of d calculated. If the result should not be satisfactory, the assumed value of D can be changed and the calculation repeated. The wall, of course, must not be too thin, as the tube must be threaded for the connector. Sometimes the ends of the tube are swaged down to increase the wall thickness where it must be threaded. The above equation for the necessary section diameters applies to straight tie-rods only. If the rod is bent to avoid obstructions, or if the connectors are offset, a heavier section must be used. In such cases a tube of elliptic section, with the long diameter in the plane of the "offset," has advantages.

Steering Connectors—If the pivot axes at opposite ends of the axle are vertical (and therefore parallel), the ends of the knuckle arms swing in the same plane and can be connected by plain forked connectors. If, on the other hand, the pivot axes are not parallel, the ends of the arms swing in planes making an angle with each other, and in that case the connectors must be of a universal type, similar to those used with drag links. Connectors may be either formed integral with the drag link or tie-rod, or they may be made in the form of separate units which screw over the ends of the link or rod. Integral connectors have been much used with drag links, while separate connectors are more common with

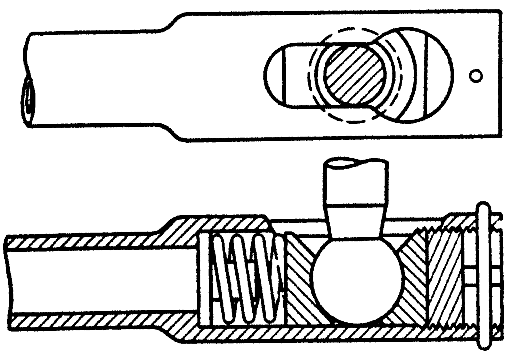


Fig. 21.—Integral Connector for Steering Link.

tie-rods, the former usually being considerably shorter and therefore easier to handle during machining operations.

A design of integral connector for a drag link is shown in Fig. 21. The end of the drag-link tube is enlarged by swaging, and has the following parts introduced into it in succession: A spring rest and guide, a coil spring, a ball socket, the ball at the end of the steering arm, another ball socket, and a screw plug secured by a cotter pin. In this particular design the shell of the connector has a hole drilled through it, of a diameter sufficiently large to permit of entering the ball, which latter, after being introduced, is moved endwise into a slot only sufficiently wide to just accommodate the neck of the ball stud. In some other designs the slot extends all the way to the end, so that the ball can be introduced into the connector from the end, which latter is then closed by a screw cap instead of a plug. It was formerly customary to provide fairly long, flexible springs back of both socket blocks in the drag-link connector, with the object of providing a cushion in the linkage, thus preventing the transmission of road shocks to the steering wheel; but since it was found that these favor low-speed wheel wobble, they are omitted, and only a single, stiff spring is used, to prevent backlash in the connector.

Separate connectors that screw over the end of the tube sometimes have the thread extend only about halfway through the shank, the outer end being bored out to the diameter at the bottom of the thread, and the tube within this part left unthreaded. This part of the shank is split longitudinally and provided with lugs which are drilled for a clamping bolt. There naturally will be some stress concentration where the tube leaves the connector, but this will be less if the tube at

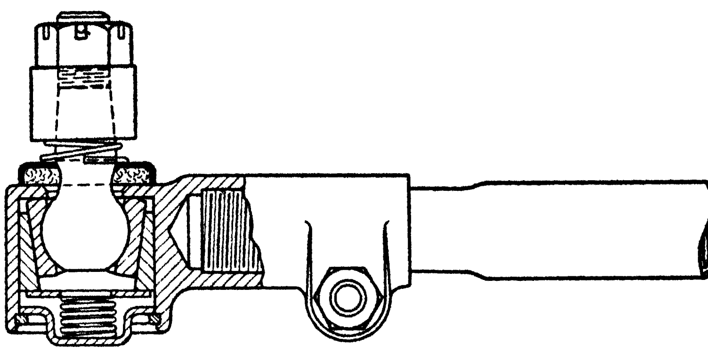


FIG. 22.—SEPARATE CONNECTOR FOR TIE-ROD.

that point is not threaded. A separate connector for steering linkages, the product of a parts manufacturer, is shown in section in Fig. 22. It will be seen that the ball stud is inserted from below, through a hole only large enough to just pass the shank, and the connector therefore cannot possibly come apart accidentally. The spherical sockets are pressed against the ball by a coil spring through the intermediary of wedges, and there is, therefore, no cushioning effect in the connector. The latter is closed at the bottom by a cupped disc acting as spring rest, which is held in place by a snap ring, and at the top by a rubber washer held in place by a spring. Such connectors are always provided with lubricator fittings.

Still another connector for tie-rods is shown in Fig. 23. Here the spring which takes up slack due to wear on the ball or socket is located within the ball, and the joint is closed by a pressed-steel disc which is set into a counter bore in the

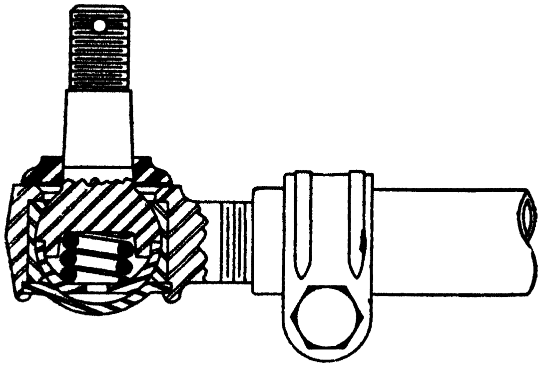


FIG. 23.—ANOTHER DESIGN OF SEPARATE CONNECTOR.

connector body, which is then flanged over. The shank of this connector is screwed into the tubular tie-rod, which latter is slotted lengthwise and clamped to the shank.

Steering Stops—To prevent contact between the tire and the car frame or the drag link of the steering mechanism when the front wheels are swung over to one side all the way, the front axle is provided with steering stops. Such a stop is easily provided in the case of a reverse Elliott steering head, as shown at *L* in view *A* of Fig. 24. In view *B* of the same figure is shown an arrangement used with Elliott-type axles. In this case a lug *L* on the knuckle arm contacting with an adjustable stop on the axle center limits the steering motion.

Carrying Thrust Load on Hardened Buttons—As an example of Continental practice, an assembly view of the axle

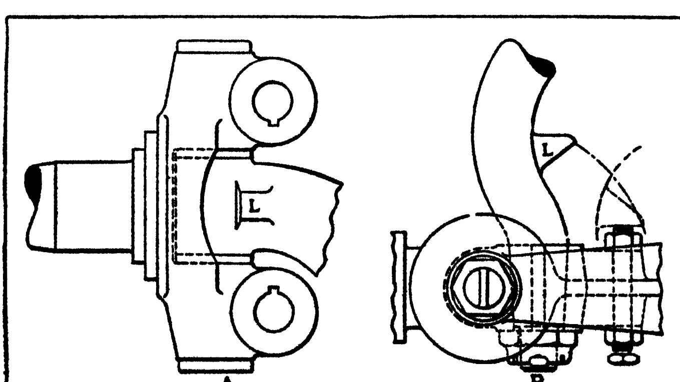


FIG. 24.—STEERING STOPS.

end of a heavy truck front axle (Fross-Buessing of Vienna) is shown in Fig. 25. The most interesting feature of this design is that the load is transferred from the axle center to the steering knuckle by a pair of hardened thrust buttons, instead of the usual antifriction thrust bearing. The lower bearing boss of the knuckle, in which one of the buttons is mounted, is virtually closed at the bottom, and to be able to assemble the knuckle pin it is necessary to make the other bearing boss of the knuckle a separate piece. A somewhat

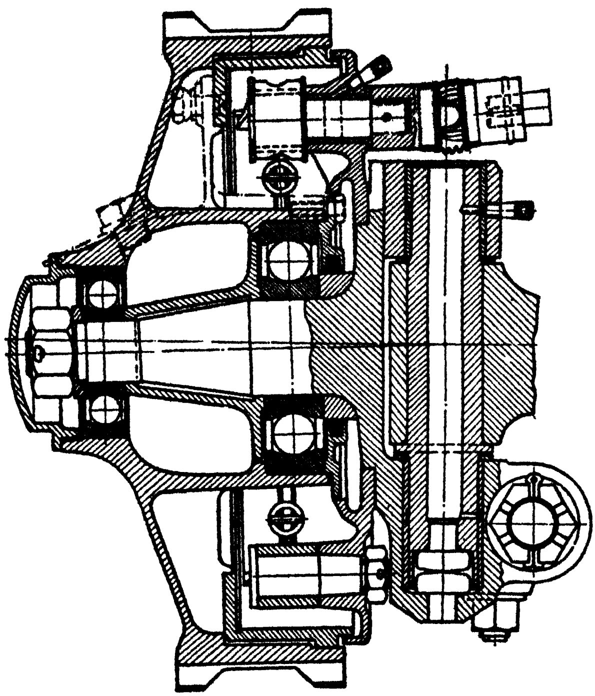


FIG. 25.—A EUROPEAN DESIGN OF HEAVY FRONT-AXLE END.

similar plan is followed by Mack Manufacturing Corp. in this country and by Associated Equipment Company in England. In the designs of these concerns the lower bearing boss is bored out to permit the introduction of the knuckle pin from below, after which the bore is closed by a screw plug, by means of which the clearance between the knuckle and the boss on the axle center can be adjusted. Attention may be called to the large radial ball bearings on the knuckle spindle in Fig. 25.

CHAPTER V

Steering Gears

Instead of the fifth-wheel steering arrangement used on horse vehicles and traction engines, the divided-axle system of steering is used on motor vehicles. This steering mechanism was invented by Lankensperger, a Munich carriage builder, in 1817. The English patent on it was taken out in the name of Rudolph Ackermann, who acted as Lankensperger's agent, and in English-speaking countries it has come to be known as the Ackermann steering gear.

One of the drawings of Ackermann's patent (reproduced in "Motor Cars" by Rhys Jenkins, London, 1902) shows the running gear of a horse-drawn vehicle. The tongue is pivoted at the middle of what we now call the axle center, and has a rearward extension with an articulated connection to the middle of the tie rod. It was pointed out in the patent that by making the length of the tie rod less than the distance between the knuckle pins, "it will occasion that fore wheel which is on the side toward which the carriage is intended to turn, to have a greater degree of obliquity than the opposite wheel. This is conducive to quick turning, because the axles of all the four wheels of the carriage become directed to one point."

A perfect steering gear would so control the deflections of the steering wheels that their axes produced would intersect each other on the axis of the rear wheels produced under all conditions, because then all of the wheels of the vehicle would always have a pure rolling motion. The Ackermann "trapeze" form of steering linkage, now commonly used, gives perfect steering for one degree of deflection (or for one turning radius) only. Under all other conditions there is a small error in the deflections, and in the design of a steering linkage the lengths of the knuckle arms and their inclination toward the vehicle axis should be so chosen that the mean of the errors throughout the steering range is a minimum.

Theory of Steering Mechanism—In the following investigation we will denote the length of the wheelbase by W; the distance between steering pivots by L; the inclination of the inner wheel axis by α; the inclination of the outer wheel axis by β; the inclination of the knuckle arms by θ; the length of the arms by l.

Referring to Fig. 1, it will be seen that

$$\frac{ac}{cd} = \cot \alpha$$

and

$$\frac{bc}{cd} = \cot \beta.$$

Hence

$$\frac{bc - ac}{cd}\left(= \frac{L}{W}\right) = \cot \beta - \cot \alpha.$$

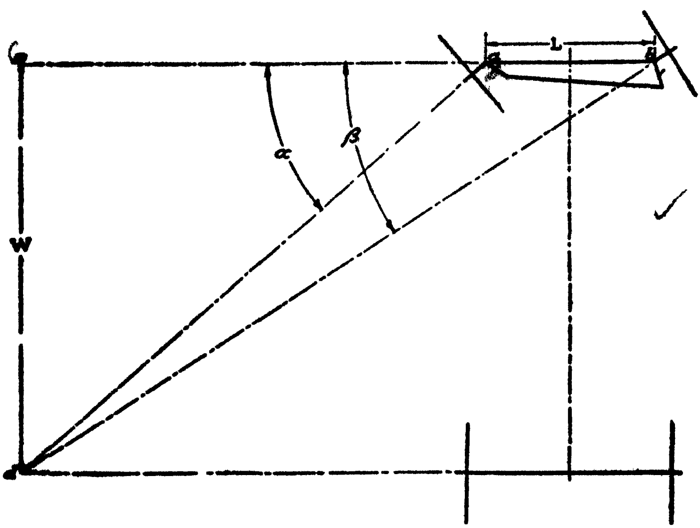


Fig. 1.

This equation enables us to plot the required values of β corresponding to different values of α for any ratio L/W. It expresses the condition which should be satisfied by the gear, but furnishes no guide as to how this may be accomplished.

Graphical Solution of Steering Problem—There is no direct analytical method for determining the most advantageous angle of the knuckle arms, and some graphical method is usually employed. By laying the steering diagram off on the drawing board to, say, half size, a sufficient degree of accu-

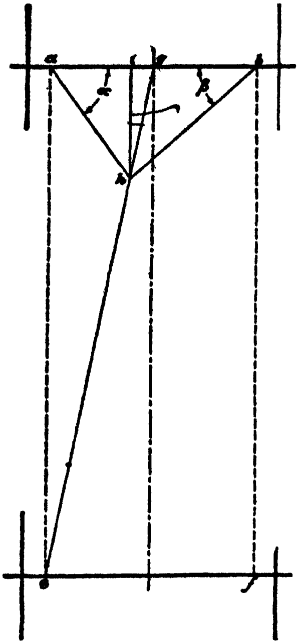


Fig. 2.

racy is attained. Unfortunately, for small deflections of the front wheels, the distance of the point of intersection of the wheel axes is so far from the axis of the car that it falls far outside the limits of an ordinary drawing board, and accuracy of the linkage with small deflections is of special importance, for the reason that the wheels are turned through a small angle very much oftener than through a big angle, and the car generally runs at a much higher speed when describing curves of large than of small radius. This difficulty may be overcome as follows (Fig. 2): From points a and b, denoting the steering pivots, draw lines perpendicular to the axles which will intersect the rear axle at e and f. Next, draw a line from the middle point g of the front axle to point e. Then lines drawn from the pivot points a and b to any point on line ge will make with the front axle corresponding steering angles. This may be proved as follows:

$$\cot \alpha = \frac{ai}{ih} = \frac{ag - ig}{ih},$$

and

$$\cot \beta = \frac{bi}{ih} = \frac{ag + ig}{ih}$$

$$\therefore \cot \beta - \cot \alpha = \frac{2ig}{ih}.$$

But

$$\frac{ig}{ih} = \frac{ag}{ae},$$

and substituting,

$$\cot \beta - \cot \alpha = \frac{2ag}{ae} = \frac{L}{W}.$$

Now assume steering arms of a definite length and making a certain angle with the longitudinal axis of the vehicle.

Next determine graphically the deflection of the outer wheel for various assumed deflections of the inner wheel, say, 8°, 16°, 24°, 32°, 40° and 48°, for these steering arms. This has been done in Fig. 3 for two particular cases. The following dimensions were assumed in making this drawing: $L = 50$ in.; $l = 7$ in. and angle $\Theta = 15°$ and $20°$, respectively. The values of angle β were determined graphically for values of angle α of 8°, 16°, 24°, 32°, 40° and 48°, respectively. After these values of angle β had been found, corresponding angles α and β were laid off on opposite ends of the line L. Through the points of intersection of the lines describing corresponding angles were drawn curves, one for 15° and the other for the 20° knuckle arms. These may be called steering-error curves, because their deviation from the diagonal line ge (Fig. 2) indicates the error in the steering angles. In Fig. 3 the diagonal ge corresponding to the value $L/W = 0.45$ is drawn in. It will be seen that for small deflections the angle of the outer wheel is too large with both 15° and 20° knuckle arms. The 20° knuckle arm gives the correct deflection of the outer wheel at about 26° of the inner wheel, and the 15° knuckle arm gives the correct deflection of the outer wheel at 46° deflection of the inner wheel. Beyond these points the angle of the outer wheel is too small. It may readily be seen from this that the most advantageous angle of the knuckle arms depends upon the turning range of the inner wheel. Thus, if the motion of the inner wheel were limited to 32°, the 20° knuckle arm would be the best, whereas if the range of motion of the inner wheel were as large as 45°, the angle of the knuckle arm should be about 18°—for a value of $L/W = 0.45$.

In order to use this method for the practical determination of the proper knuckle-arm angle, steering-error curves for different knuckle-arm angles and lengths should be laid out very carefully for permanent use, and in any particular case the diagonal ge corresponding to the particular value of L/W should be placed on the chart in pencil, when the most advantageous knuckle-arm angle will at once be apparent.

An Analytical Method—The problem of the proper angles and lengths of steering arms also lends itself to analytical treatment. Four different cases have to be considered, viz., with the knuckle arms forward and back of the axle respectively, and with the knuckle-arm angle θ greater and less than the deflection β of the outer wheel, respectively. In Fig. 4 the knuckle arms extend to the rear of the axle and angle θ is greater than β. In this diagram the knuckle arms are pur-

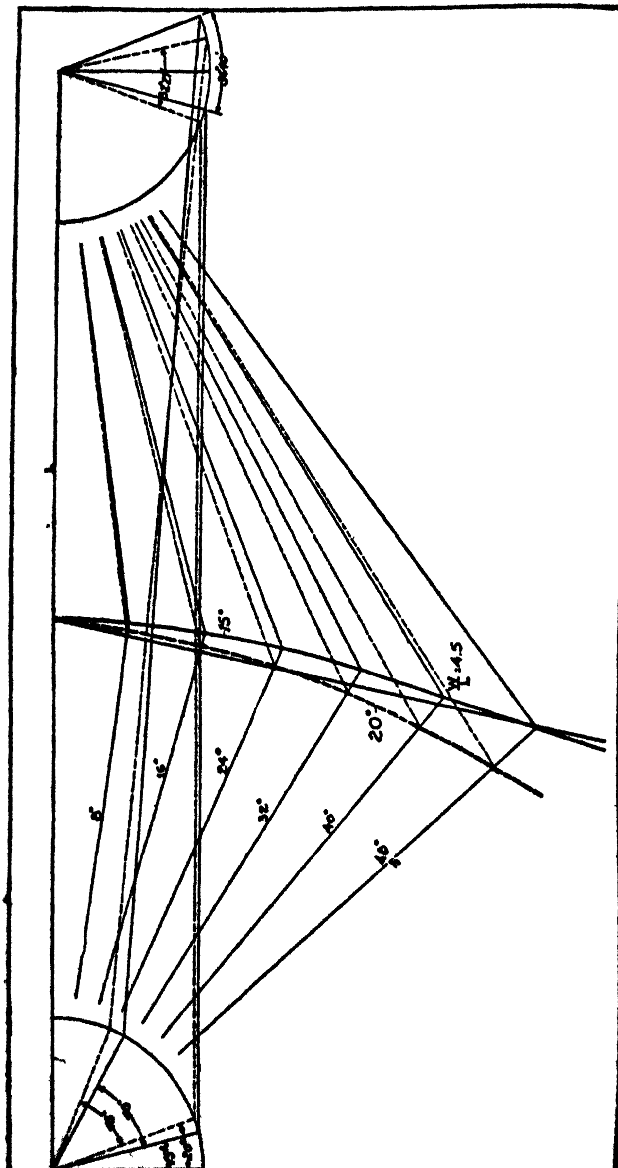


FIG. 3.

posely shown abnormally long, for the sake of greater clearness. Referring to the drawing,

$$M = L - 2l \sin \theta \qquad ge = l \sin (\alpha + \theta)$$

$$ga = l \cos (\alpha + \theta) \qquad hi = jb = l \sin (\theta - \beta)$$

$$jf = l \cos (\theta - \beta) \qquad ge + M - N + hi = L$$

Substituting values of ge and hi,

$$l \sin (\alpha + \theta) + M - N + l \sin (\theta - \beta) = L.$$

Substituting the value of M,

$$l \sin (\alpha + \theta) + L - 2l \sin \theta - N + l \sin (\theta - \beta) = L.$$

Transposing and dividing by l,

$$\sin (\theta - \beta) = 2 \sin \theta - \sin (\alpha + \theta) + \frac{N}{l} \quad \ldots\ldots\ldots \quad (1)$$

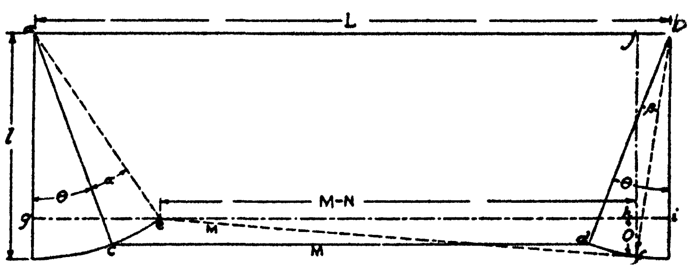


FIG. 4.

By a similar process of reasoning it is found that with β greater than θ and the arms extending to the rear

$$\sin (\beta - \theta) = \sin (\alpha + \theta) - 2 \sin \theta - \frac{N}{l} \quad \ldots\ldots\ldots \quad (2)$$

with β smaller than θ and the arms in front of the axle—

$$\sin (\theta - \beta) = 2 \sin \theta - \sin (\theta + \alpha) - \frac{N}{l} \quad \ldots\ldots\ldots \quad (3)$$

and with β greater than θ and the arms in front of the axle

$$\sin (\beta - \theta) = \sin (\theta + \alpha) - 2 \sin \theta + \frac{N}{l} \quad \ldots\ldots\ldots \quad (4)$$

These various equations cannot as yet be solved, because N

is not known. The value of N in terms of known factors may be found as follows:

$$O = jf - jh.$$

When β is smaller than θ,

$$jf = l \cos (\theta - \beta)$$

hence

$$O = l \cos (\Theta - \beta) - l \cos (\alpha + \Theta)$$
$$= l [\cos (\Theta - \beta) - \cos (\alpha + \Theta)] \quad . \; . \; . \; . \; . \quad (5)$$

Similarly, when β is greater than θ,

$$jf = l \cos (\beta - \Theta)$$

and

$$O = l [\cos (\beta - \Theta) - \cos (\alpha + \Theta)] \quad . \; . \; . \; . \; . \quad (6)$$

In every case

$$M - N = \sqrt{M^2 - O^2}$$

and

$$N = M - \sqrt{M^2 - O^2} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (7)$$

The equations thus derived permit of accurately determining the angle of the outer wheel corresponding to any angle of the inner wheel, the proportions of the linkage being given. The following example shows its method of application: Suppose the distance L between pivots to be 50 in.; the length l of the knuckle arms, 6.5 in.; θ, 20°, and α, 30°, the knuckle arms extending to the rear of the axle. Then according to equation (2)

$$\sin (\beta - 20°) = \sin (20 + 30)° - 2 \sin 20° - \frac{N}{l}.$$

Disregarding the term N/l for the moment we get for a first trial value

$$\sin (\beta - 20°) = 0.082.$$
$$\beta = 24° \, 42'.$$

Inserting this value of β in equation (6) we get

$$O = 6.5(0.9966 - 0.6428) = 2.2997,$$
$$M = 50 - (2 \times 6.5 \times 0.342) = 45.56$$

and inserting these values in equation (7) we get

$$N = 0.053.$$

Now using this value N in equation (2) we get

$$\beta = 25° \, 10'.$$

The calculations could be continued further, but it has been shown that the second trial value is correct within one minute in the most extreme case, which is as high a degree of accuracy as is required in practical work.

By this method the errors in the steering-gear angle were calculated for maximum angles of deflection of 25° and 30° of the inner wheel, with knuckle arms set at from 15° to 28°

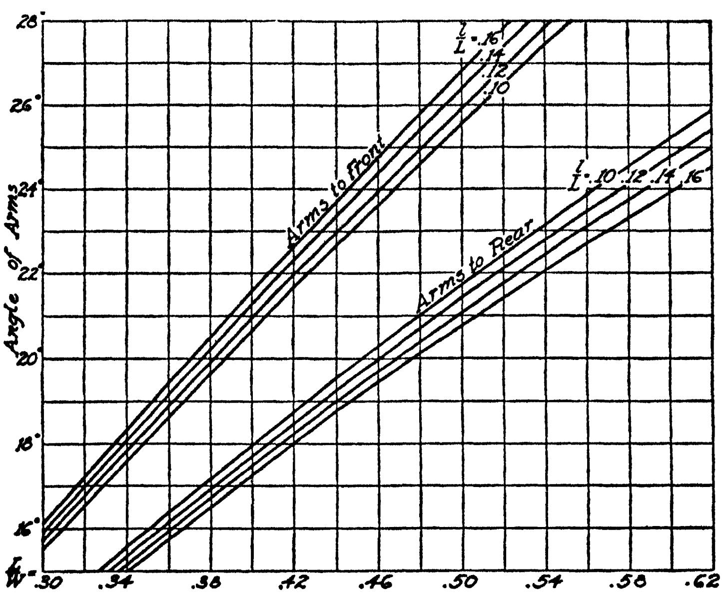


FIG. 5.—PROPER KNUCKLE-ARM ANGLES FOR A LIMITING DEFLECTION OF 33°.

and extending both forward and back from the axle, and for four different values of the ratio l/L. The results of these calculations are plotted in Figs. 5 and 6. These diagrams give the best values for knuckle-arm angles for limiting turning angles of 33° and 40° of the inner wheel when the values of L/W and l/L are known.

General Arrangement of Steering Gear—Motor vehicles are steered by means of hand wheels, the wheel being secured to the upper end of a shaft which at its lower end carries the steering gear proper. As a rule, the shaft is made tubular, so the cable from the horn button on the steering wheel can be passed through it, but solid shafts also are being used. The steering wheel usually is 17 in. in diameter in light passenger cars, 18 in. in heavier passenger cars and light trucks, and 20 and even 22 in. in the heaviest trucks and buses. The steering shaft usually is surrounded by a light tubular jacket, which sometimes is clamped both in a boss on the steering-gear housing and in a bracket secured to the cowl, while in other cases it is supported by the cowl only. The assembly of steering shaft and jacket is referred to as the steering column.

The steering gear, which reduces the angular motion im-

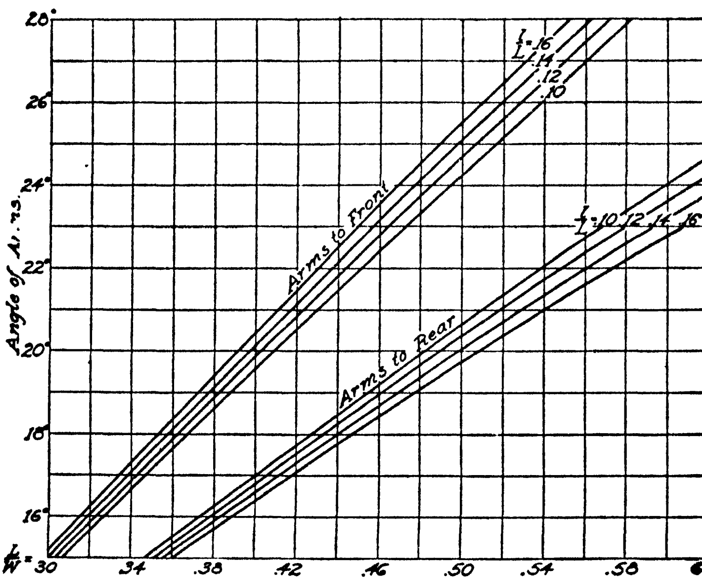


Fig. 6.—Proper Knuckle-Arm Angles for a Limiting Deflection of 40°.

parted to the steering wheel, is located at the bottom of the column in a cast housing mounted on the frame of the vehicle. A shaft protruding from this housing carries a steering arm, which connects by a linkage to the two knuckle arms. As the steering arm is spring-suspended while the knuckle arms are not, this linkage must be so arranged that it is not influenced by spring action. That is to say, with the steering wheel in a definite position, any compression or rebound of the front springs must not cause the front wheels to turn around their knuckle pins. Since the knuckle arms move in a horizontal and the steering arm in a vertical plane, the joints with the link between the two must be of a universal type, and ball joints are generally used. A general view of a complete steering mechanism for a vehicle with a divided steering axle is shown in Fig. 7. The arrangement of the

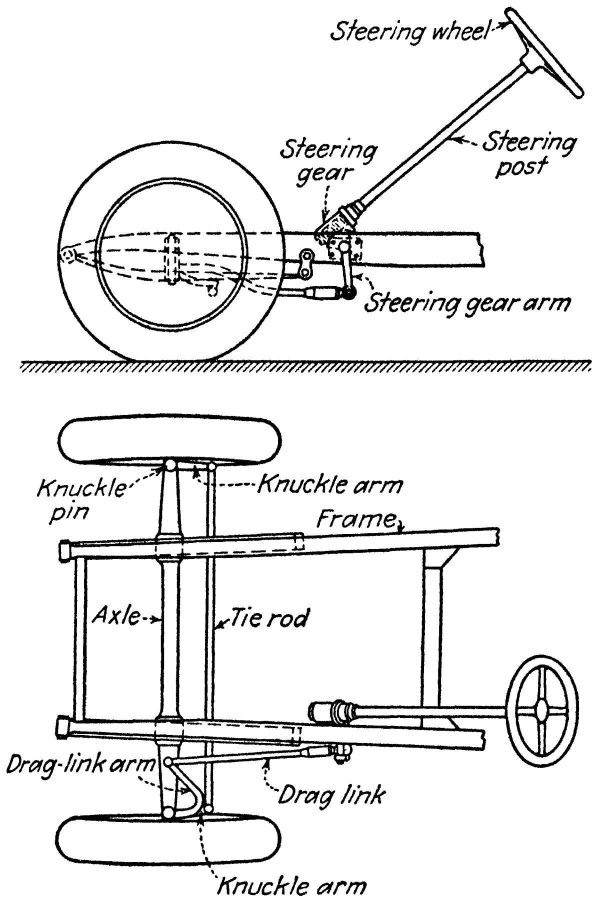


FIG. 7.—ARRANGEMENT OF CONVENTIONAL STEERING GEAR.

steering linkage differs with the type of front suspension, and also with the location of the steering gear relative to the front wheels. For instance, with the conventional front axle there is always a tie-rod between the knuckle arms on opposite sides, which keeps these arms a constant distance apart. This arrangement evidently is impractical where the steering knuckles are supported by parallel links or double wishbones, because with this arrangement the fulcra of the knuckle arms move relative to each other under spring action, and the outer ends of these arms must move similarly if spring action is not to affect the steering. Also, whereas in the conventional car the steering gear usually is located a considerable distance back of the steering knuckle on its side, so the two can be connected by a relatively long drag link, in other types of vehicle (COE trucks, for instance) the steering gear may be close to the knuckle in the fore-and-aft direction, or even a considerable distance ahead of it, in which case it is preferable to connect the steering arm to the knuckle arm on the opposite side of the vehicle, which leads to what is generally known as "cross-steering."

The steering column is always mounted on the car at an angle to the vertical, for comfort in driving. In commercial vehicles the inclination of the column is usually limited to about 20°, for the sake of space economy; in racing cars and some of the sport cars or speedsters turned out by foreign manufacturers, because the driver is seated very low, the inclination may be as much as 70°. On stock passenger cars an inclination of 50° generally makes for very comfortable steering. The proper inclination of the column, of course, is affected not only by considerations of driver comfort but also by the location of the steering gear on the frame, which must be such that the steering linkage can be properly worked out, preferably without resort to the use of curved or bent links.

Left vs Right Steering—In road vehicles built for use in the United States the steering gear is now always located on the left side. During the pioneer days the opposite practice was followed, because in horse vehicles the driver always sat on the right side, so that when he wanted to get out at the curb he did not have to disturb the passenger on the seat beside him. With the American rule of the road, requiring all vehicles to keep to the right, the driver can better gauge his distance from vehicles coming in the opposite direction when he is seated at the left. He could better judge his distance from the curb or from a vehicle being overtaken if he were seated on the right, but then, a collision with a vehicle traveling in the opposite direction would be likely to have much more serious consequences than one with a vehicle traveling in the same direction, or running into the curb, because of the greater relative speed. Hence it is important that the driver be afforded every facility to avoid collision with vehicles traveling in the opposite direction, and this makes left-hand drive preferable. In England, where the rule of the road is to keep to the left, right-hand steering has the same advantages as left-hand steering in this country, and motor vehicles built in Great Britain for use within the country have right-hand steering.

Steering-Gear Requirements—The requirements made of a steering gear are somewhat conflicting, and, like most other engineering products, the average steering gear is a compromise. The primary object of the gear is to multiply the turning effort impressed on the wheel by the driver, so that he can steer the vehicle without exerting too much physical effort. Then, the steering gear should be to a certain degree irreversible or back-locking, so that shocks sustained by the road wheels will not be transmitted in toto to the driver's hands. These two requirements already conflict in a degree, for the more friction there is in the steering gear, the better the shocks are absorbed, while, on the other hand, with a great deal of friction in the gear it is difficult to operate it. Fortunately, there are mechanisms that are much more efficient when transmitting motion in one direction than when transmitting it in the opposite direction, and this difficulty, therefore, can be circumvented. However, it is generally desired to have a trailing or self-righting effect, so that if the road wheels are deflected from the straight-ahead course, and the driver then releases his hold, they automatically return to the straight-ahead position. Such an effect can be obtained only if the resistance to the transmission of motion from the wheels to the steering wheel is comparatively low. Requirements with respect to the ratio of the gear also are conflicting. For high-speed driving a moderate or low gear ratio is required, with which the driver can perform any given steering maneuver quickly. On the other hand, in congested traffic, where speeds are naturally low, and where the car often may have to be parked in confined spaces, a higher ratio is advantageous, as it reduces the effort required on the steering wheel. In 1950 American passenger cars the steering-gear ratio varied from about 14:1 in the lightest to about 26:1 in the heaviest models.

Types of Steering Gear—In very light vehicles (cycle cars and midget cars) steering gears have been used occasionally which, while they reduced the motion of the steering wheel, did not give a back-locking effect. They comprised either a pinion and rack or a bevel pinion and sector. For standard cars capable of high speeds the self-locking feature is essential to safety. There are two fundamental types of mechanism that give this self-locking effect, viz., the worm and worm wheel (or worm-wheel sector), and the screw and nut, and most of the earlier steering gears comprised either one or the other of these mechanisms. Modern steering gears also are based on these same types of mechanism, but have been refined with a view to reducing friction and the effort required to operate them. A sufficient degree of irreversibility is obtained by using a high reduction ratio. The manufacture of steering gears has been highly specialized, and only three makes of steering gear are used on American passenger cars. The same three manufacturers supply steering gears also for most makes of commercial vehicles.

The plain screw-and-nut type of gear has been abandoned, because with the large contact surface in such a device there

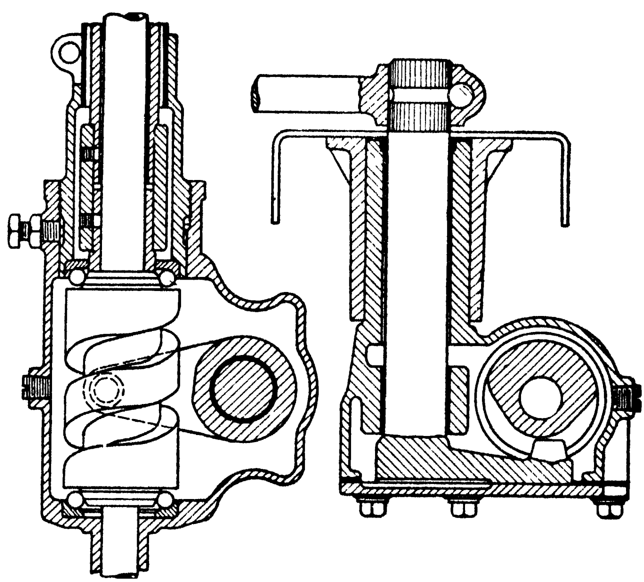


FIG. 8.—ROSS CAM-AND-LEVER STEERING GEAR.

is too much friction when the gear is cold and the lubricant is stiff. In fact, in the newer steering mechanisms sliding contact is replaced by rolling contact wherever practical.

Ross Cam-and-Lever Steering—A number of different types of steering gear are being manufactured by Ross Gear & Tool Co., Lafayette, Ind., all based on the cam-and-lever principle. Referring to Fig. 8, the cam is essentially a worm of variable pitch, carried on the lower end of the steering shaft. A conical stud, carried at the end of a short lever arm, engages into the groove or tooth space of the worm. The arm is formed integral with a shaft perpendicular to the steering shaft, and to the opposite end of this shaft, outside the steering-gear housing, is secured the steering arm. End thrust on the worm in both directions is taken up on ball bearings at top and bottom. In the latest model an adjusting screw with lock nut is provided in the cover plate, concentric with the steering shaft, which limits end play of the shaft. Owing to the small area of contact between stud and cam, the friction between them is comparatively small. The pitch of the cam groove is smallest at the center, which gives the driver the maximum mechanical advantage when the gear is in and near the straight-ahead position, where it is most of the time. Of course, when the gear approaches either extreme position, the mechanical advantage is smaller, and parking maneuvers therefore are somewhat more difficult.

A modification of the gear just described has the stud at the end of the lever carried in two taper roller bearings, which results in rolling instead of sliding motion between stud and cam, and is claimed to reduce the steering effort required by 50 per cent.

Ross steering gears are also being made in a "twin-lever" form (Fig. 9), with either fixed or roller-mounted studs. There is, in reality, only a single lever in the gear, but this lever is forked and carries two studs which engage the cam groove three pitches apart. When the gear is in the central position, both studs contact the cam, while near the extreme positions, where the pitch is a maximum, only one stud contacts the cam. With this gear the mechanical advantage from the driver's end is a minimum in the central position, and increases toward both extreme positions, which makes parking maneuvers easier. Besides, near the central position, where most of the motion occurs, the load is divided between two contact surfaces, which reduces wear. Fig. 10 shows the variation of the steering-gear ratio with the angular distance of the steering arm from the central position for both the single lever and the twin-lever types. It will be seen that in the extreme positions the mechanical advantage or leverage from the driver's end is about 45 per cent greater with the twin than with the single-lever type. In the single-lever type the steering-arm motion is limited to 80°; in the twin-lever type, to 100°. Of course, mechanical advantage and steering motion required are always inversely proportional. That is to say, if the mechanical advantage is great and the steering wheel therefore turns easily, the driver will have to turn the wheel farther for a certain road-wheel deflection.

Worm-and-Roller Steering Gear—In another type of steering gear, in production both in this country and abroad, a

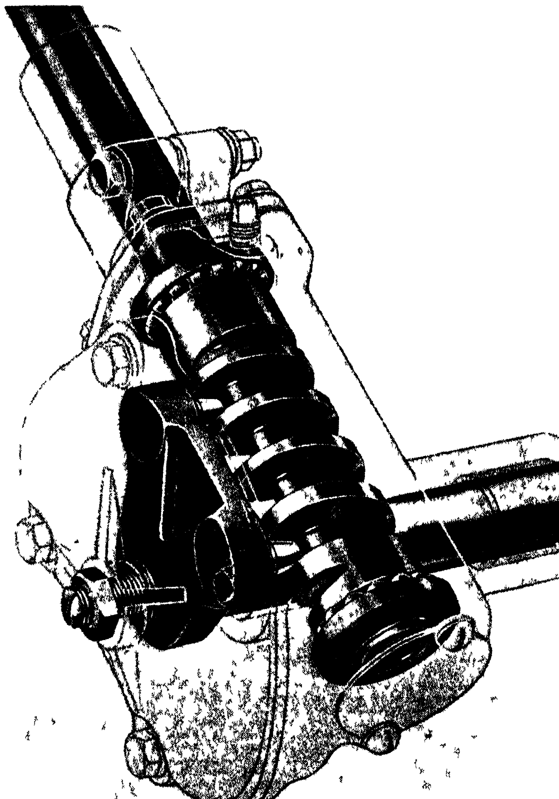


FIG. 9.—ROSS TWIN-LEVER STEERING GEAR

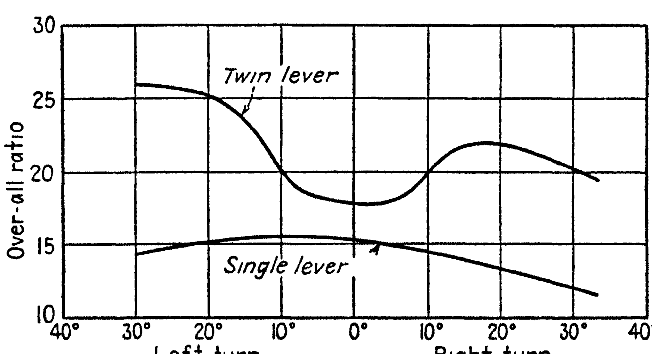


FIG. 10.—VARIATION OF STEERING-GEAR RATIO WITH DEFLECTION.

roller having one, two, or three teeth extending around its circumference is in mesh with an hour-glass type of worm on the steering tube. The roller is carried on anti-friction bearings in the end of a short arm on the steering shaft. In Fig. 11 are shown two sectioned views of a gear of this type in production by Gemmer Manufacturing Company. End thrust on the worm in both directions is taken up on taper roller bearings, which can be adjusted by means of shims under the cover plate. The plane through the axes of worm and roller (which contains the center points of tooth contact) is not quite at right angles to the steering-shaft axis, and any axial motion of the steering shaft brings the teeth into closer mesh or moves them farther apart. Adjustment for mesh is made by means of shims under a thrust washer at the short end of the straddle-mounted steering shaft. By removing a shim and then tightening the nut on the thrust washer, the steering shaft is moved axially a distance equal to the thickness of the shim, and the roller is brought into closer mesh with the worm. The adjusting nut is locked by means of a lock plate, which latter is of such design that successive locked positions of the nut are closely spaced. There are two quill

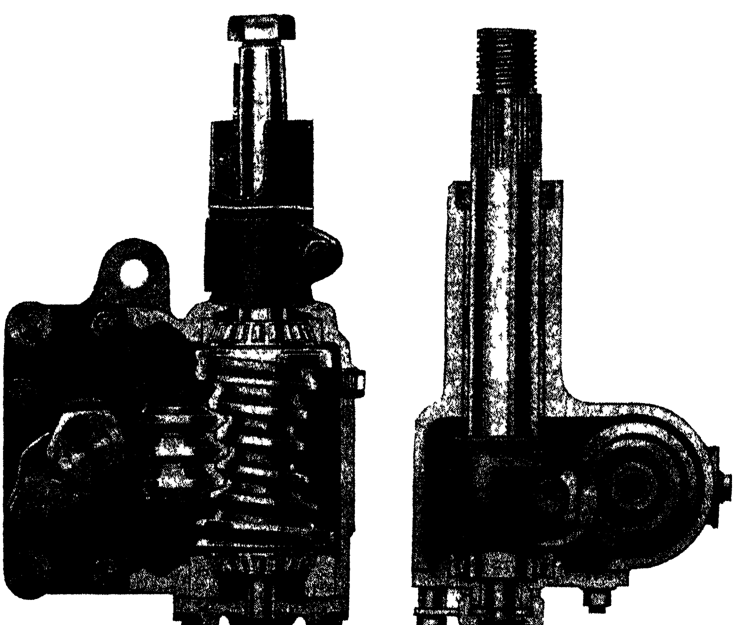


Fig. 11.—Gemmer Worm and Roller-Lever Steering Gear.

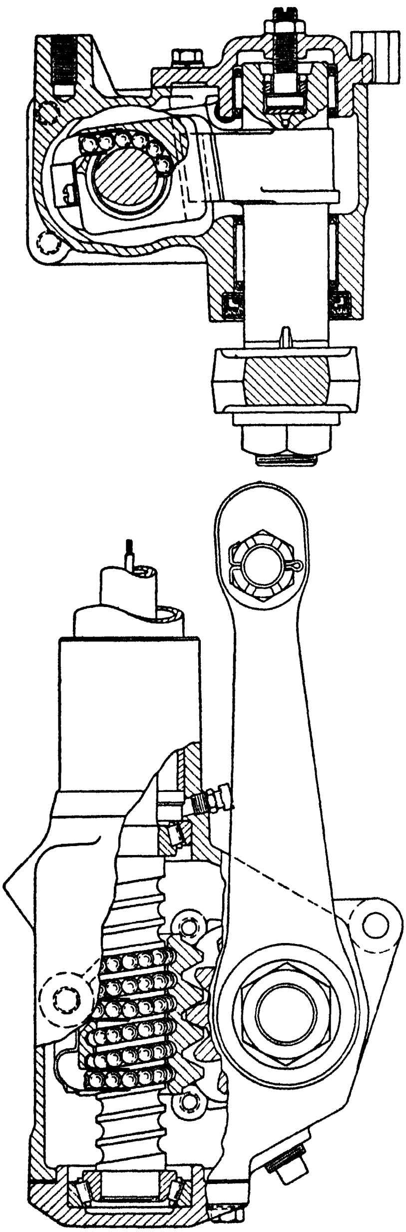


FIG. 12.—SAGINAW BALL-NUT STEERING GEAR.

bearings in the triple-toothed roller.

Circulating - Ball Type Steering Gear—A variation of the old screw-and-nut type steering gear, in which rolling motion is substituted for the sliding motion between screw and nut, is being manufactured for heavy vehicles by the Saginaw Steering Gear Division of General Motors Corporation. As shown in Fig. 12, the "screw," which is welded to the steering shaft, is provided with a helical groove forming a ball race, and there is a corresponding internal groove in the nut, both grooves being ground. Within the length of the nut the thread groove is filled with steel balls. Tubular ball guides secured to the nut deflect the balls from their helical path near one end of the nut, and lead them diagonally across the outer face back to the helical path at a point near the center of the nut. There are thus two distinct circuits for the balls, one in each end of the nut. The screw is mounted between two roller bearings; one of these is adjustable and permits taking up slack and preloading the bearings.

This steering gear is made in both constant- and variable-ratio types. The one here illustrated, which is being used on large buses, has a ratio of 25.6 and requires 6.04 turns of the steering gear for a steering-arm travel of 85°. The gear sector is straddle-mounted on quill bearings. It will be seen that the sector tapers from one side toward the other, so that the nut when meshing with the sector assumes an inclined position in the gear. This makes it possible to control back-lash between the teeth by a simple endwise adjustment of the steering-arm shaft. In the variable-ratio type the ratio increases from 21.75 at the center to 27.1 after two turns of the steering wheel in either direction, and remains constant during the remainder of the motion. An obvious advantage of this type of steering gear is that while low friction is ensured by the steel balls, the load between screw and nut is distributed over a comparatively large area.

A rather unusual mechanism is employed in the steering gear of Mack buses. As shown in Fig. 13, a worm meshes with a gear sector on the steering shaft, the sector having conjugate teeth on one side of its rim. Presumably the reason for the use of this type of gear is that tooth contact occurs well below the level of the bearings, so that a lubricant of low viscosity can be used without fear of loss by leakage through the bearings after these have become worn. The low-viscosity lubricant reduces the friction in the mechanism and thus makes it easier to steer the vehicle. In the bus the steering gear is mounted at the extreme forward end of the chassis, and the drag link extends back from it.

Steering-Gear Capacities—Steering gears are made in stock sizes by their manufacturers, and are rarely specially designed for a given vehicle, except possibly as regards the mounting lugs or brackets on the housing. The maximum torque loads on the input and output shafts (steering shaft and steering-arm shaft) are proportional to the maximum weight on the front wheels and the diameter of these wheels. Since the torque load which a shaft can sustain varies as the cube of its diameter, the steering-arm shaft diameter should be equal to the cube root of the product of maximum front-axle load and wheel diameter, in lb-in., divided by a constant. Usually one member of the steering gear (a worm sector, for instance) is integral with the steering-arm shaft, which is therefore made of alloy steel of high tensile strength. Steering gears are subjected to maximum loads when a car is being maneuvered out of a deep rut, or when a front wheel hits a curb at a fairly sharp angle, and these loads are entirely indeterminate. In passenger-car practice the constant c in the equation for the steering-arm shaft diameter,

$$d = \sqrt[3]{\frac{WD}{c}} \text{ in.},$$

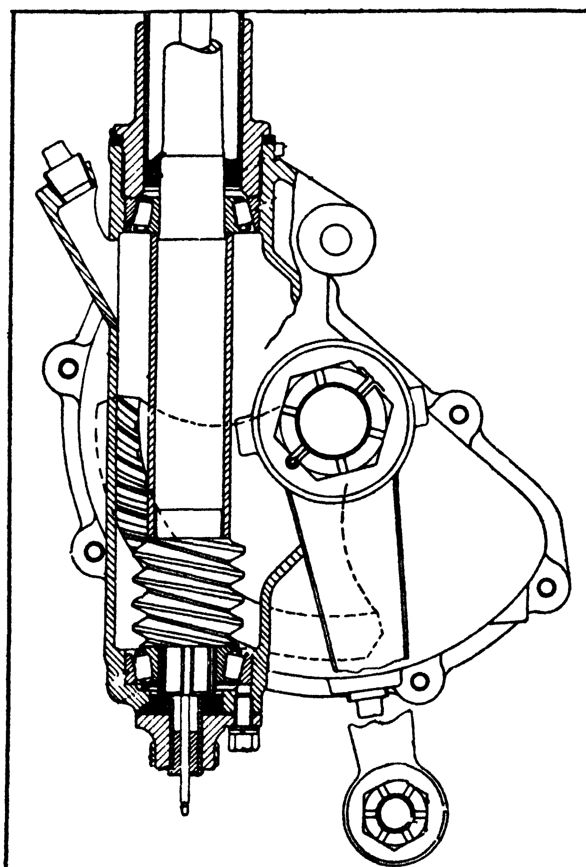


FIG. 13—MACK WORM AND SECTOR STEERING GEAR.

varies between 40,000 and 60,000, while in the case of steering gears for heavy commercial vehicles it ranges between 60,000 and 75,000. In the above, W is the weight carried by the front wheels under full load, and D the nominal diameter of these wheels. The section modulus of the steering shaft evidently can be made less than that of the steering-arm shaft in the proportion of the product of the steering-gear ratio and the minimum steering-gear efficiency to unity. Usually, however, the steering shafts are made somewhat larger, and diameters range from ¾ in. for light passenger cars to 1 in. for trucks and buses.

The steering arm or drop arm is secured to its shaft by a tapered, serrated joint, which does not reduce the torsional strength of the shaft materially and gives the best assurance against loosening. Serrated taper shaft ends have been standardized by the S.A.E. in sizes from ½ in. to 3 in. diameter. The taper is ¾ in. per ft on the nominal outside diameter, and the cutting angle is 1° 37′. Thirty-six serrations are used on shafts of ½ and ⅝ in. diameter, and 48 on all larger shafts. The hub of the steering arm is firmly held in place on the shaft by a large nut, which is locked by a lock washer, a split pin, or both.

Steering Linkage—In discussing details of the linkage connecting the steering gear with the steering knuckles, a distinction must be made between the conventional steering layout, in which the steering arm swings in a vertical fore-and-aft plane, and cross steering, in which it swings in a plane parallel with the steering-column axis and making the same angle with the vertical as the latter. With the conventional layout, the steering-arm shaft extends across the frame side rail above or below, or through its web, and the arm is vertical when the gear is in the "straight-ahead" position. At its free end the steering arm carries a ball for a ball joint, which may be made integral with it, but is now generally made separate in the form of a ball stud. The ball at the end of the steering arm must be located at a definite height, in order that spring action may have a minimum effect on the steering mechanism. There is likely to be always a slight effect of this kind. That is to say, with the steering gear set for straight-ahead motion, for instance, compression or rebound of the chassis spring adjacent to the drag link will deflect the front wheels slightly from the straight-ahead position, which, of course, is undesirable. This effect can be minimized in various ways, as, for instance, by making the drag link relatively long. With a fore-and-aft drag link back of the axle, and semi-elliptic front springs, the best arrangement (Fig. 14) is to articulate the rear end of the spring and shackle the front. The front axle then will describe an arc around the axis of the rear spring bolt, while the forward end of the drag link will describe an arc around the center of the steering-arm ball, and there will be little deviation between these arcs,

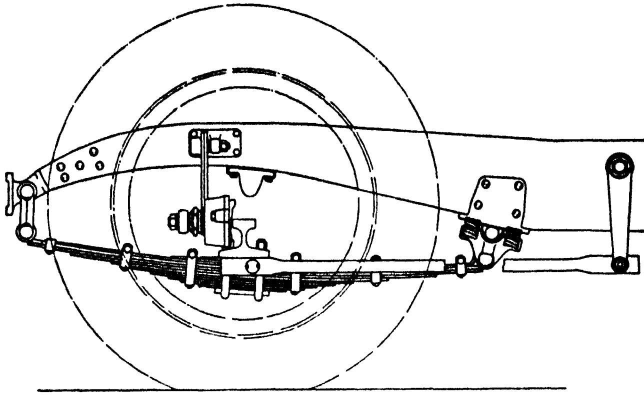


Fig. 14.—Connection of Spring to Frame for Minimum Effect of Spring-Action on Steering.

which may be made to coincide when the spring is under normal static load. The drag link should be parallel with the rear half of the front spring when both are in their normal positions.

Not quite such good results are obtainable if the front end of the front spring is articulated to the frame and the rear end shackled. In that case, as illustrated in Fig. 15, as the front spring compresses and rebounds, arcs are described by the front axle and the front end of the drag link which are tangent to each other, but whose centers, instead of being on the same side, are on opposite sides of the point of tangency, for which reason there is greater divergence between them. The divergence will be a minimum when the drag link is in line with the forward section of the spring when the spring is under normal static load, as shown in Fig. 15.

With this type of steering gear, which is used with vehicles having a front axle, the steering knuckles on opposite sides of

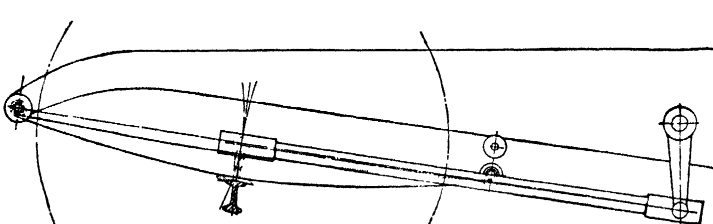


Fig. 15.—Best Lay-out of Front Spring and Drag Link When Rear End of Spring Is Shackled.

the vehicle rotate on pins which are fixed relative to each other, so that the distance between them is constant. The free ends of the knuckle arms therefore also can remain a constant distance apart, and can be connected by a tie rod. When cross steering is used on a vehicle having a front axle, the knuckle arms on opposite sides also are connected by a tie rod. In that case, in order to minimize the effect of spring action on steering, it is advisable to provide a radius rod between the side of the frame where the steering gear is located and the end of the axle on the opposite side. In vehicles in which the steering knuckles are carried on parallel links, and therefore are capable of motion relative to each other, the steering-linkage problem is somewhat different.

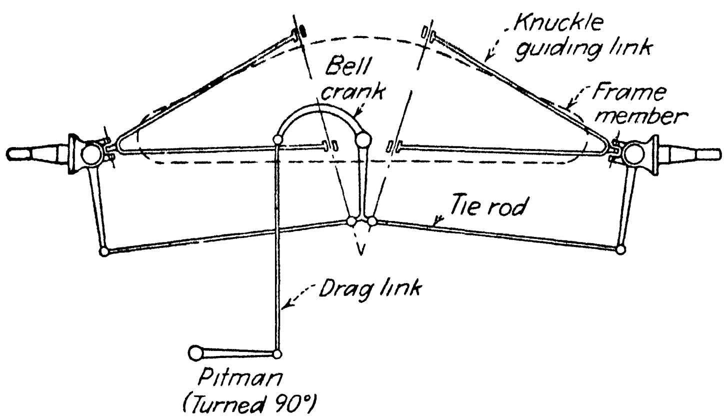


FIG. 16.—STEERING LINKAGE FOR CARS WITH FRONT INDEPENDENT SUSPENSION.

Steering Linkage for Independently-Sprung Cars—With the predominant system of independent front suspension, the steering heads, on which the steering knuckles are swiveled, are supported by two wishbone-type links each, of which the lower is considerably longer than the upper one. As indicated in the diagram Fig. 16, the lower wishbone, or knuckle-guiding link, is pivoted to the frame front cross member on an axis making a small angle with the longitudinal center line of the chassis. At the middle of the cross member there is a bell crank which is capable of turning around a vertical axis. The rearwardly-extending arm of this bell crank, which is at substantially the same level as the pivots for the wishbone, has two ball studs secured to it, from which tie rods connect to the free ends of the knuckle arms. As the connection between bell crank and tie rod is in the axis around which the wishbone turns, it is obvious that steering will not be affected by spring action. With this layout the steering arm swings in a vertical fore-and-aft plane.

Another steering linkage for cars with independent front suspension is shown in Fig. 17. In this case the steering gear is mounted on the inner side of one frame side rail and its arm swings in a transverse plane. An idler arm of the same length is pivoted on a base secured to the other side rail, and a tie rod connects the two arms. From points along this tie rod two links with universal fittings connect to the knuckle arms. The points of connection between the links and the tie rod are not on the axes of the lower wishbones when the front wheels are in the straight-ahead position, but farther out from the center of the chassis. The reason for this is that the bosses at the ends of the knuckle arms are above the level of the

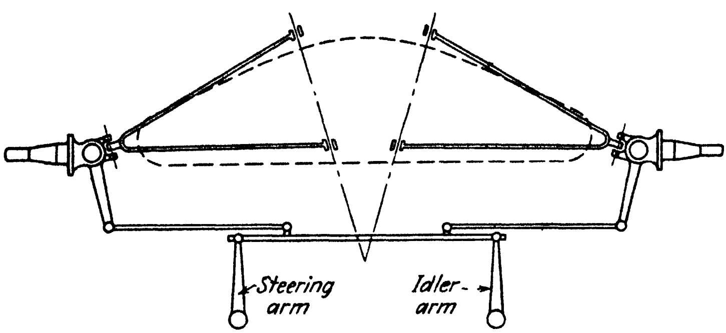


Fig. 17.—Another Steering Linkage for Cars with Independent Suspension.

connections between the lower wishbones and the steering heads, and the connections must approach the axes of the upper wishbones (in the plan view) in proportion as the bosses at the ends of the knuckle arms approach the level of the upper wishbones.

Steering Arm—Steering arms for passenger cars are made from 5 to 8 in. in length, while those for heavy commercial vehicles usually are between 8 and 10 in. long. The proper length for the steering arm depends on the ranges of angular motion of the steering knuckle and steering arm respectively, and on the length of the knuckle arm to which the drag link is connected. Steering arms are proportioned approximately as indicated in Fig. 18. The arm is secured to the steering-gear shaft by means of a tapered, serrated joint, being drawn up tight on the tapered part of the shaft by means of a nut which is securely locked. A steering arm can be, and sometimes is, made with an integral ball for the ball-and-socket joint at its lower end, but the preferred practice is to fit a ball stud into it. Ball studs have been standardized by the S.A.E.

"Insulated" Steering Arms—In order to minimize the transmission of road shocks to the driver's arms, the steering arms of passenger cars sometimes are "rubber-insulated." The steering-arm shaft then is provided with a comparatively short, double-armed lever, with bosses at its ends that are machined to receive two annular rubber insulators each, these insulators resting against an internal flange at the middle of the boss and projecting slightly above its surfaces. The steering arm extends parallel with the lever and is secured to it by studs or bolts passing through the rubber insulators, so that there is no direct metallic contact between the two parts.

Drag Link—The member connecting the steering arm with the knuckle arm is usually referred to as the drag link.

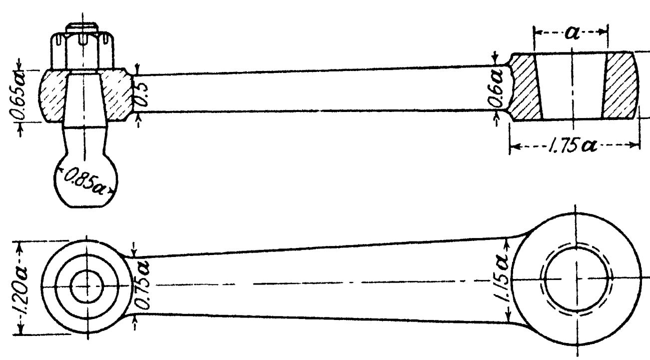


FIG. 18.—PROPORTIONS OF STEERING ARM.

It is tubular, and usually of the same section as the tie-rod. If it is much shorter than the tie-rod it could be made of somewhat smaller section and still have the same factor of safety as the latter, but usually very little weight would be saved, and a little higher factor of safety than absolutely required is not undesirable in this important part. Moreover, there is a slight manufacturing advantage in being able to make both units of the same tubular stock, especially since both usually are fitted with connectors of the same design. If it is desired to rationalize the choice of section for the drag link, the load on it in lb can be taken as that corresponding to the maximum torque which the steering-gear shaft will withstand safely. The value of this load can be inserted in the Rankine formula for columns, and the necessary section modulus determined, using the same factor of safety as in the case of the steering arm. The ball joints used for drag links are of the same general type as those used for tie-rods.

Steering Stop—Stops must be provided to limit the angular deflections of the front wheels in both directions, as otherwise the tires might rub against the frame, and possibly also

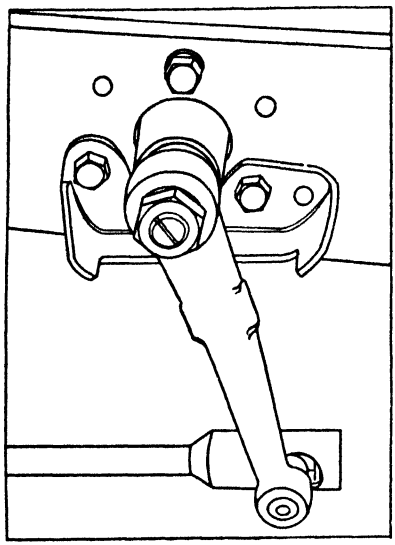


FIG. 19—STEERING STOP ON GEAR.

against the fenders, which would cause undue wear on them. These stops can be placed in the path of motion of either the steering arm or the steering knuckle (or a part rotating with the later). A steering stop coacting with the steering arm, as used on Chevrolet cars, is shown in Fig. 19.

Steering Wheels—In the case of a passenger car the steering wheel serves an ornamental as well as a utilitarian purpose. Steering wheels for cars are now generally made with die-cast hubs, flexible spokes of stainless steel, and a rim of cellulose-acetate plastic with a steel core. The steel core is made of $\frac{3}{8}$-in. round stock, the blanks being cut to length, flattened where the spokes come, punched for the spokes, and then formed into a ring and butt-welded. Die castings are used not only for the hub, but also for a spoke-spacing ring and possibly also for a horn-operating ring. Stainless steel is used for the spokes because it retains its natural lustre. These spokes usually consist of four or five wires or strips, hence they are quite flexible in the axial direction, which reduces the transmission of vibrations to the driver's arms. At their inner ends the spokes are threaded by rolling, and they screw into the hub, which has threads die-cast in it. After the core of the rim has been assembled to the spoked hub, the spokes are silver-soldered to it, electric heat being used in the process. Finally, the plastic material is molded to the core in an hydraulic press.

Steering wheels sometimes are made with only two spokes, or with three spokes unequally spaced angularly, to enable the driver to look through the upper part of the wheel at his instrument board without having his vision obstructed by a spoke. Steering wheels for commercial vehicles usually are made with malleable iron spiders and rims of hard rubber or similar plastic material, on metallic cores.

Safety in Steering Gears—Every part of the steering mechanism must be designed with an eye to a high degree of safety. The major parts, such as the steering knuckles and arms, are made of high-grade alloy steels, carefully heat-treated. All bolted connections must be positively locked, preferably by both lock washers and cotter pins. The tie-rod and drag-link tubes must be of such wall thickness that if threads are cut on them for the connectors, these will not reduce their strength inordinately. Welded and pinned (or riveted) joints between these tubes and their connectors are preferred by some engineers. If a threaded joint is to be used, it is advisable to counter-bore the threaded hole in the connector for some distance to a snug fit over the unthreaded portion of the tube, slotting the hub of the connector lengthwise and clamping it down over the tube with a substantial clamp bolt. The end of the connector hub should be tapered down to an edge, in order to prevent concentration of stress where the tube leaves the connector. If a portion of the tube outside the connector hub were threaded, that would be a particularly weak section.

Turning Radius—The turning radius of a motor vehicle may be measured in different ways, viz., to the center of the track of the outside front wheel, to the outside of the wheel or rim of the outside front wheel, and, finally, to the most projecting part of the vehicle, which is usually the tip of the outside front fender. In order to avoid confusion due to inexactedness of nomenclature, the Society of Automotive Engineers adopted the following definition: "The turning radius of an automotive vehicle is the radius of the arc described by the center of the track made by the outside front wheel of the vehicle when making its shortest turn."

This turning radius depends upon the wheelbase w (Fig. 20), the distance b between knuckle pivot axes, the maximum angle θ through which the inside front wheel can be deflected from the straight-ahead position, and the distance d from the intersection of the front-wheel axis with the knuckle-pivot axis to the center plane of the front wheel. Referring to Fig. 20, by one of the rules for the solution of the oblique triangle,

$$c = \sqrt{a_2 + b_2 - 2ab \cos (180° - \theta)}$$

but

$$\cos (180° - \theta) = -\cos \theta,$$

hence

$$c = \sqrt{a^2 + b^2 + 2ab \cos \theta}.$$

If now we substitute the value of a in terms of the wheelbase and angle of lock ($a = w/\sin \theta$), we have an expression for c entirely in known terms:

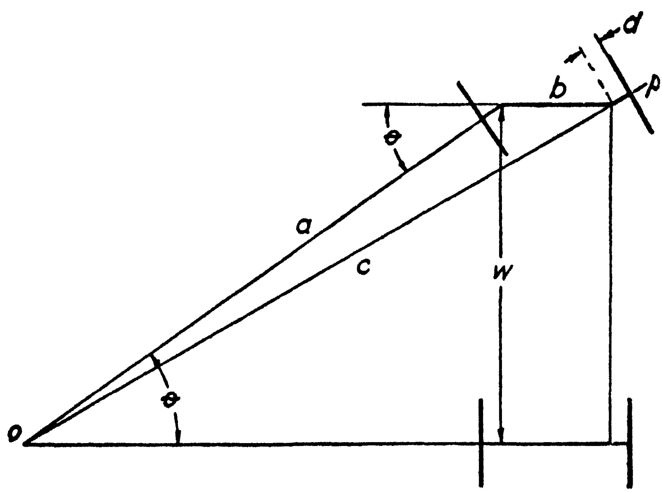


FIG. 20.—TURNING-RADIUS DIAGRAM.

$$c = \sqrt{\left(\frac{w}{\sin \theta}\right)^2 + b^2 + \frac{2wb}{\tan \theta}}.$$

The turning radius is eoual to $c + d$, and hence is

$$R = \sqrt{\left(\frac{w}{\sin \theta}\right)^2 + b^2 + \frac{2wb}{\tan \theta}} + d \text{ in.}$$

However, the turning radius is most conveniently found graphically, by laying off the plan of the chassis to scale, with the front wheels deflected to the limit, and measuring the distance from the point where the axes of the front-wheel spindles produced intersect the axis of the rear wheels, to the center of the outside front-wheel track.

Turning radii of American passenger cars of other than the midget type range between 17 5 and 25.5 ft, and average 20 ft.

Steering Radii of Trailers—Each wheel of a four-wheeled vehicle, as a rule, describes a separate track, but it is usually

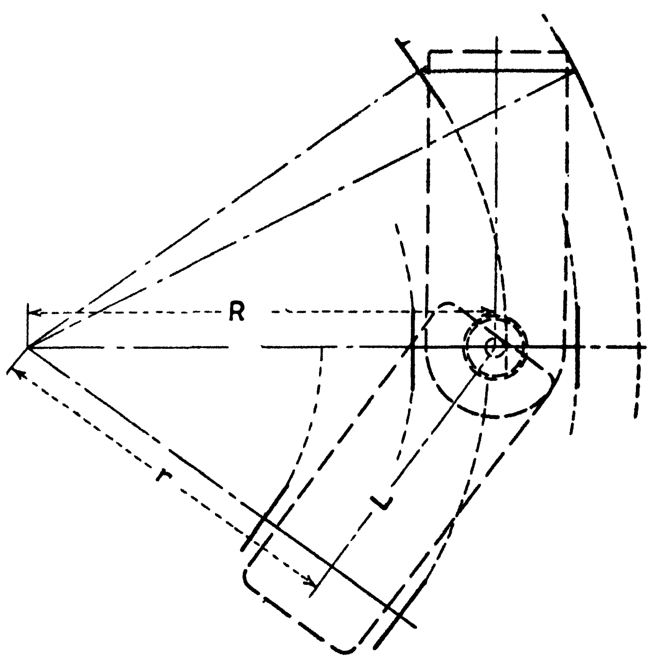


FIG. 21.—TURNING-RADIUS DIAGRAM FOR TRACTOR AND TRAILER.

most important to know the radius of the track described by the outside front wheel, as it is the largest. However, under certain conditions it is desirable to know also the radius of the track described by the inside rear wheel, as the distance apart of these two tracks, which is equal to the difference between their radii, determines whether a vehicle or tractor-trailer combination can negotiate the turn at intersecting streets of given widths. The radius of the track described by the inside rear wheel also is best determined graphically. For the case of a four-wheeled vehicle it can be measured off directly from a diagram similar to that of Fig. 20. For a tractor and trailer combination the diagram takes the form shown in Fig. 21. That portion pertaining to the tractor is identical with the diagram Fig. 20. Let L be the distance from the fifth wheel axis of the trailer to the center of its rear axle. The distance R, from the turning axis to the center of the rear axle of the tractor, can be determined by measurement on a diagram similar to Fig. 20. Now, the axis of the rear axle of the trailer evidently must intersect the turning axis, and the angle ϕ between the axis of the two rear axles therefore must be such that $L/R = \sin \phi$. After this angle has been determined, the plan of the trailer can be laid off, and the distances from the turning axis to the center of the trailer inner wheel can then be measured off. The distance R from the turning axis to the center of the trailer rear axle is equal to

$$\sqrt{R^2 - L^2}$$

and the distance to the center of the rear inner track evidently is less by half the tread.

Where a vehicle has to be designed so as to be capable of turning in very close quarters, it is made to steer by both the front and the rear wheels. If both wheels on the same side of the vehicle are deflected equally from the straight-ahead position, the front and rear wheels will "track."

Steering of Six-Wheelers—Since perfect steering requires that the axes of all of the wheels should always pass through a single vertical line, it is obvious that the six-wheeler cannot have perfect steering; for the axes of the two pairs of rear wheels are parallel. Fig. 22 shows a steering diagram for such a vehicle. There must, of necessity, be a small amount of slippage of the rear wheels on the ground when describing curves, but it has been shown that if the two driving axles are fairly close together, this slippage is less than that with large dual rear tires which, although turning at the same angular speed, must describe circles of different diameters while describing curves.

Shimmy and Wheel Wobble—With high-speed vehicles fitted with low-pressure pneumatic tires, a good deal of trouble has been experienced from wheel wobble and shimmy. Wheel wobble consists of vibration of the front axle assembly around a horizontal longitudinal axis through its center. The front axle is mounted between two "springs," the chassis spring and the tire, and when there is wheel wobble, the chassis spring on one side and the tire on the other compress simultaneously, while the other spring and tire rebound. Such vibration can become troublesome only if there is synchronism between a periodic exciting force and the natural period of the vibrating body. The amplitude which may be attained by the vibration depends on the exciting force on the one hand, and on the damping forces on the other. The natural period of vibration of the vibrating body, the front-axle assembly, varies in the same sense as the moment of inertia of the assembly around the axis of vibration, and inversely as the restoring forces, that is, the spring forces of the tires and chassis springs, which may be expressed as so many pounds per inch of deflection or compression.

The other phenomenon referred to in the foregoing, shimmy, consists of a vibration (a flapping motion) of the front wheels around the knuckle pins. Shimmy is closely related to wheel wobble. When a car is in motion, the front wheels act as gyroscopes, and when the axis of a gyroscope is moved angularly in a given plane, so that the plane of rotation changes, a force is set up which tends to move the axis angularly in a plane at right angles to that of the first angular motion. The moment with which this gyroscopic force tends to deflect the spinning wheel from its plane of rotation is proportional to the polar moment of inertia of the wheel, to its angular velocity, and to the velocity with which its axis is being moved by the applied force. When there is wheel wobble, since the vertical movement of the spinning wheel

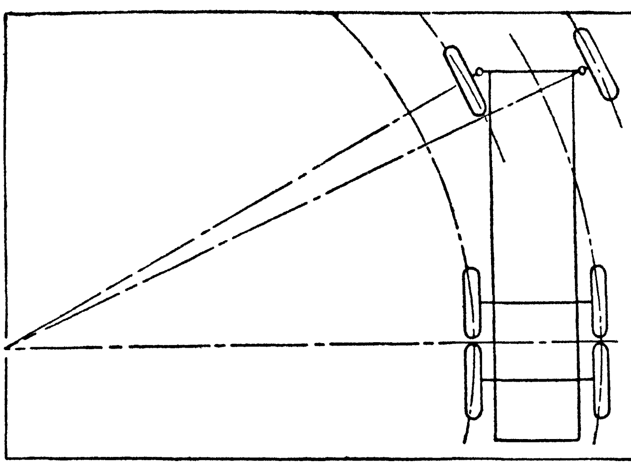


FIG. 22.—TURNING-RADIUS DIAGRAM OF SIX-WHEELER.

alternates in direction, the force causing movement in the horizontal plane also alternates in direction, hence a flapping motion around the knuckle pins is produced, which is known as shimmy.

There is also another coupling element between the two vibratory systems, which may be explained as follows: Wheel wobble involves alternate compression and rebound of the chassis springs. Whenever the chassis spring on the side where the drag link of the steering gear is located is compressed, if the steering gear is very rigid and only slightly reversible, the front wheels are compelled to move slightly around the knuckle-pin axes, owing to geometrical imperfections in the steering linkage. For instance, if the forward end of the front spring is pin-jointed to the chassis frame, when the spring is compressed the axle will describe an arc around the front-spring front-eye axis as a center, while the forward end of the drag link will describe an arc around the center of the steering-arm ball. These two arcs deviate from each other, and this deviation must be compensated for by angular motion of the front wheels around the knuckle pins.

Nature of the Exciting Forces—The foregoing makes it clear that in the conventional front end we have two coupled vibratory systems, capable of vibration, respectively, around a central longitudinal axis and the knuckle pins, or around one horizontal and two vertical (or nearly vertical) axes. Because of the coupled condition, vibration in either the vertical or horizontal plane always is accompanied by vibration in the other plane. The question now arises as to the nature of the exciting forces. These may have their origin either in the wheels or in the road surface. It is quite evident that if one of the front wheels should be slightly eccentric (or have "run out"), the adjacent axle end would rise and fall once during every wheel revolution, and wheel run-out has been definitely shown to be a potential cause of wheel wobble. The effect, of course, is amplified if both front wheels have run-out and their eccentricities happen to be diametrically opposed. Wobble may be caused also by lack of balance of the front wheels. Therefore, to guard against wheel wobble, front wheels should be carefully checked for run-out and also accurately balanced.

Wheel wobble and shimmy can be induced also by road irregularities, and the latter, apparently, are chiefly responsible for the so-called low-speed shimmy, since humps in the road surface are likely to be separated by distances considerably shorter than the wheel circumference, which latter, together with the car speed, determines the frequency of impulses due to imperfections in the wheels. In general, if the wheels tend to wobble on perfectly smooth roads, it is due to imperfections in the wheels, whereas wheel wobble on rough roads is likely to be the result of road conditions.

Preventatives and Palliatives—No absolute preventative against, or cure for, shimmy and wheel wobble is known. Troublesome vibration in other parts of the car often can be eliminated by raising the natural frequency of the elastic body, but application of this principle to wobble and shimmy problems holds out little promise, as it would require stiffening of the front springs and raising the tire-inflation pressures, both of which are contrary to the requirements of a comfortable ride. One thing that can be done to reduce the tendency to shimmy is to minimize the play in all connections of the steering linkage, as with rigid parts and little play in the connections the possible amplitude of shimmy is limited. The fact that comparatively high ratios are now used in passenger-car steering gears undoubtedly tends to restrict the shimmying proclivities of these cars. It was formerly the practice, especially in connection with high-grade cars, to place cushion springs in the drag links, which gave the latter a certain freedom relative to the steering arm, and was intended to prevent the transmission of road shocks to the steering wheel. This evidently favors shimmy, and the practice therefore has been discontinued, although a single short, stiff spring is still used in the connector to prevent rattle.

Since wheel wobble and shimmy are vibration phenomena, one way to control them is by the provision of strong damping forces. Unfortunately, since the wheels have to go through the same motion in the steering operation as in shimmy, damping the shimmying motion makes steering harder, at least if the damping force acts directly on the pivoted knuckles. Damping of the chassis springs would have a similar effect, and would be less objectionable from the standpoint of steering effort required. It has been suggested in this connection that the shock absorbers (which are actually vibration dampers) are most effective in preventing wheel wobble if they are connected to the axle center at its extreme ends. Shock absorbers always are mounted on the frame side rails, and originally their arms extended parallel with the side rails. To increase their effect in controlling wheel wobble, they later were so mounted that their arms extend parallel with the axle, as shown in Fig. 23. This same arrangement is now extensively used in connection with the parallel-link system of front independent suspension, where the shock-absorber arms form one of the pair of links.

The wheel-wobble problem became quite acute shortly before front independent suspension was adopted generally on passenger cars, and the opinion was frequently voiced that adoption of this suspension system would be the solution of the problem. In this connection it must be remembered that there are different types of independent suspension. In one of these the steering knuckles are so guided that under spring action they move parallel to the plane in which the wheels rotate, and in that case there can be no gyroscopic effect. On the other hand, with the more popular type of independent suspension, with parallel links, the plane of rotation of the wheels does change with spring deflection, and a gyroscopic effect is produced. I once owned a car with this type of front suspension which developed a bad case of shimmy after about 50,000 miles. The only thing the service stations could do was to tighten the steering gear, which evidently increased the damping forces, but it gave relief for a few days only.

Another preventative that has been suggested consists in connecting each steering knuckle separately to the steering gear, the idea being that if there is no direct link between the

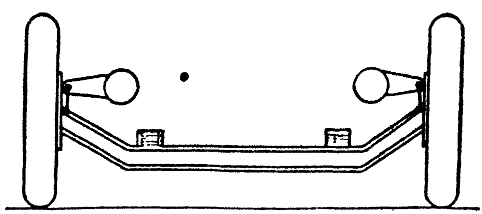


FIG. 23.—SHOCK ABSORBERS MOUNTED TO DAMP SHIMMY MOST EFFECTIVELY.

two steering knuckles, they cannot so readily vibrate in unison, as they must when there is shimmy. However, whatever the linkage used, it must be such that any motion of one wheel around its knuckle pin is accompanied by a similar though slightly different motion of the other wheel, and it is therefore difficult to see what difference it can make whether the knuckles are connected by a single link or by two in series. Packard Motor Car Co. claimed to have materially reduced the shimmying tendency of its cars by adopting the front suspension illustrated in Fig. 14, where the front spring is shackled at its forward end, and is pin-jointed to a spring-cushioned rocking bracket at the rear. This type of spring mounting enables the axle to move slightly in the fore-and-aft direction relative to the frame, thereby compensating for inaccuracies in the steering linkage, and preventing "wheel kick."

In recent years stabilizers have been widely adopted for passenger cars, and while this does not seem to be their primary object, there is little doubt that they tend to minimize trouble from wheel wobble. Such a stabilizer consists essentially of a bar extending transversely across the frame and supported in bearings thereon. At its ends the bar is provided with radial arms which are linked to the steering heads or support arms. If one of the wheels passes over an obstruction, and the adjacent chassis spring is compressed thereby, the stabilizer helps to resist the compression and transfers part of the load to the spring on the opposite side, which therefore compresses more than it would under the same conditions without the stabilizer. In wheel wobble the spring on one side compresses while that on the other side rebounds, and the action of the stabilizer therefore hampers wobble.

A hydraulic shimmy damper was invented by F. W. Lanchester, a British engineer who has specialized on vibration problems. It comprised a segmental chamber in which there was a vane capable of angular motion therein. The entire chamber was filled with oil, which could pass from the compartment on one side of the vane to that on the other side through a valve within the vane. The oil could pass through this valve only at a slow rate, and the idea was that while it would strongly damp the rapid motion associated with shimmy, it would not have very much effect at the slower motion of the vane in regular steering operation. So far as the author knows, the device did not come into commercial use.

Power Steering—With increase in the size of commercial vehicles the effort required to steer them also increased. It increased even faster than the load carried by the steering road wheels, because the greater this load, the greater the contact area between tire and ground, and the longer the average lever arm through which friction between tire and ground resists steering motions. Power units for applying the brakes have long been standard equipment on heavy commercial vehicles, and it was therefore only natural that attempts should be made to develop such units also for steering gears. Activity in this field began during the late twenties. But either because there is less actual need for power steering than for power brake application, or because the problem of a practical power-operated steering gear involves greater difficulties, power steering has had only very limited application so far. Certain auxiliary military vehicles used in World War II carried as much as 23,000 lb on their steering wheels, and this necessitated the use of power steering for their cross-country operation. This application seems to have revived interest in power steering.

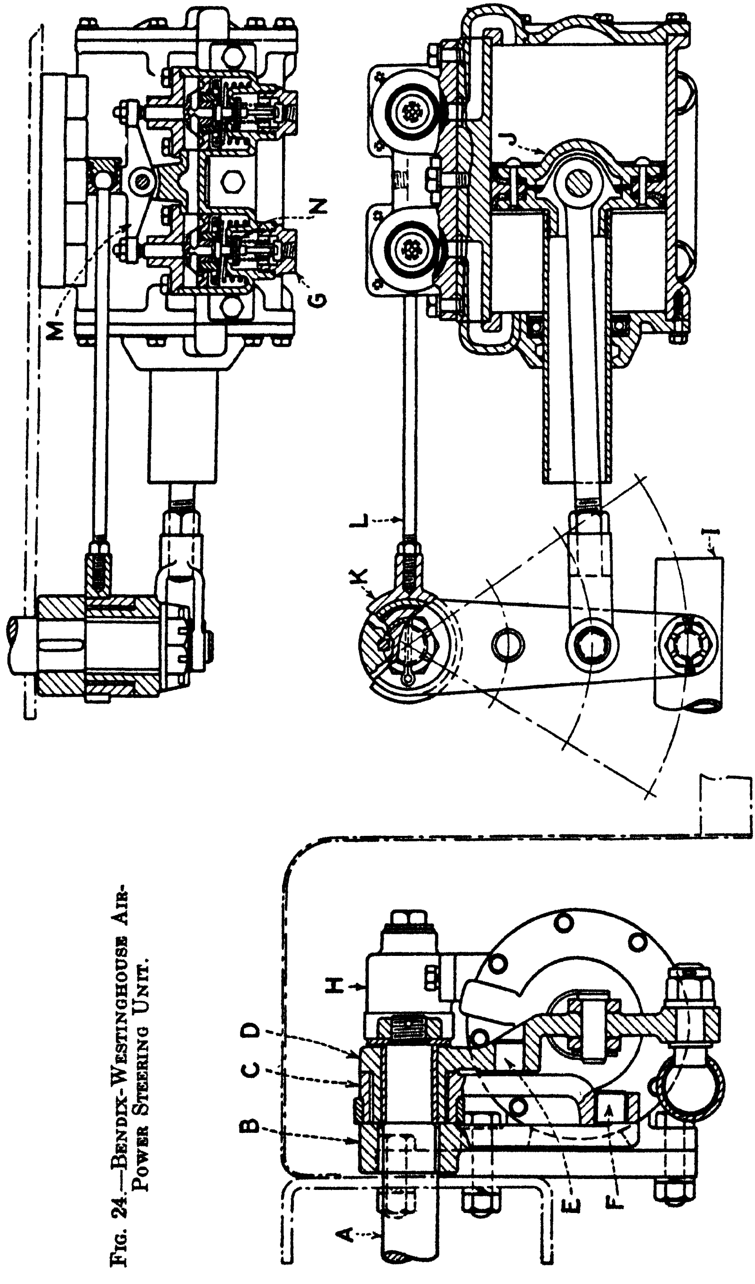


FIG. 24.—BENDIX-WESTINGHOUSE AIR-POWER STEERING UNIT.

A power steering unit consists essentially of two parts, one for generating or transmitting power, the other for controlling it. The control unit must be actuated by the steering wheel. When there is no torque (or pull) on the steering wheel, the power must be shut off. If the steering wheel is pulled around in one direction, the power must be applied to force the road wheels around in the same direction, and if the pull on the steering wheel is reversed, the direction of the force applied to the road wheels also must be reversed. For the sake of safety, these units must be so designed that if the source of power, or the means by which it is transmitted, fails, the vehicle can be steered by hand in the usual way. Some objection has been raised to power steering systems which fail instantly when the engine stalls, as with power steering the driver normally has only a light hold on the steering wheel and is unprepared to instantly exert the very much greater steering effort required. Another problem encountered in the development of power steering units is that of preventing "wandering" of the vehicle. As the steering wheel when set for straight-ahead motion must have a slight degree of freedom for the operation of the control unit, it cannot be depended upon to hold the vehicle to a steady course, which function must be performed by the power unit.

Bendix-Westinghouse Air Steering Unit—A power steering unit operated by compressed air is being offered by the Bendix-Westinghouse Air Brake Co. and three sectional views of same are shown in Fig. 24. Mounted on the steering shaft *A* are three levers—the control lever *B*, the intermediate lever *C*, and the steering arm *D*. Control lever *B* is secured to the steering shaft. Intermediate lever *C* is fulcrumed on the steering arm by pivot pin *E*, and its lower end is connected to control lever *B* by pin *F*. The hub of lever *C* is bored out slightly larger than the diameter of the steering shaft, so that this end has freedom of motion through a small range for the operation of control valves *G* and *H*. Drag link *I* is connected to the end of the steering arm, and piston *J* of the power cylinder is connected to the steering arm between the drag-link connection and the pivot pin *F* of the intermediate lever *C*. Yoke *K* is mounted on the upper end of intermediate lever *C* and holds control rod *L* in line with the steering shaft.

When the steering wheel is turned for a right-hand turn, control lever *B* moves to the right. As there is always a certain resistance to deflection of the road wheels, control lever *B* moves the upper end of intermediate lever *C* to the left, since the later is pivoted on the steering arm at *E*. Movement of the upper end of intermediate lever *C* causes valve rod *L* to move, and the latter, through rocker arms *M*, then exerts pressure on the plunger of control valve *G*, which admits air to the left end of the double-acting cylinder. Air enters the cylinder until its pressure on piston *J* is sufficient to overcome the resistance of the wheels to deflection, and it then moves steering arm *D* to the right. As long as the steering wheel is being pressed toward the right, the valve remains open. When movement of the steering wheel stops, piston *J* continues to move until the upper end of intermediate lever *C* reaches a position where intake valve *N* can close and shut off the flow of air. To maintain a certain pressure in the air cylinder, it is necessary to maintain a certain force on the plunger of the control-valve piston. For a left-hand turn the right-hand control valve *H* is actuated, and admits air to the right-hand end of the power cylinder.

In addition to greatly reducing the physical effort required to steer a heavy vehicle, this system of power steering is claimed to minimize shimmy. This is said to be due to the fact that shimmy motion in either direction will be transmitted through the drag link to the air-control levers, and will so change the admission of air to the power cylinder that the shimmy motion is opposed. Another advantage claimed is that, since the force of the power cylinder is not transmitted through the steering gears, wear on these gears is reduced.

Air-power steering units can be used only on vehicles equipped with an air compressor and an air storage tank. Most heavy vehicles for which power steering would be considered already carry such plant as part of their air-brake system. Comparatively little air is consumed in power steering, and this can readily be furnished by the compressor installation used with the brakes. This is a point in favor of pneumatic steering units.

Hydraulic steering units comprise a pump driven from the engine which puts the hydraulic fluid (oil) under pressure, and a tank from which the pump draws its oil and to which oil is returned after being exhausted by the power cylinder.

Bendix Hydraulic Steering Unit—Fig. 25 illustrates an hydraulic steering unit developed by Bendix Products Division of Bendix Aviation Corporation. It is adaptable to any stock steering gear and is here shown fitted to a Ross gear. It will be recalled that the stock Ross steering gear comprises a simple lever on the steering-gear shaft. In the power-steering assembly this simple lever is replaced by a double-armed lever, one arm of the lever engaging the worm-type cam, the other a sliding block adapted to be reciprocated by the piston in the power cylinder. The lever contacts the sliding block through a roller bearing. Thus the power cylinder is hitched directly to the steering-gear shaft and the steering arm. The control valve, which is located at the lower end of the steering gear housing, is shown in section in Fig. 26. It comprises a housing bored out centrally to receive a piston-type control valve and having a number of smaller bores surrounding the central bore in which are lodged plungers and centering

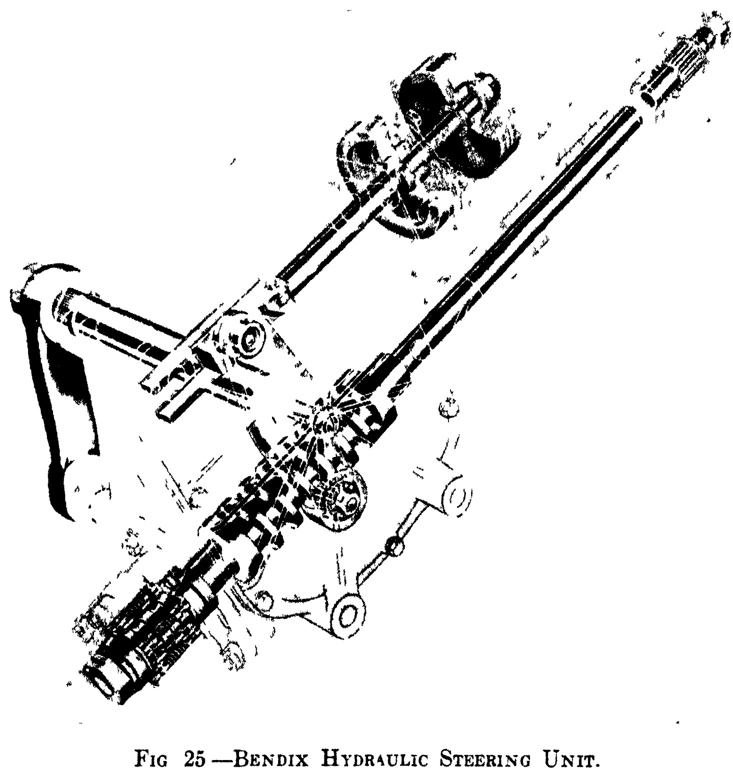


FIG 25.—BENDIX HYDRAULIC STEERING UNIT.

springs. The control valve consists of a hollow cylinder fitted over the lower end of the cam shaft, with two circumferential grooves in its outside. It is held on the cam shaft between roller-type thrust bearings and is capable of a slight axial motion with that shaft. Normally the valve is held in its central position (as shown by the lower view in the illustration) by the four centering springs and their plungers. Oil from the pump then enters the valve housing through the inlet

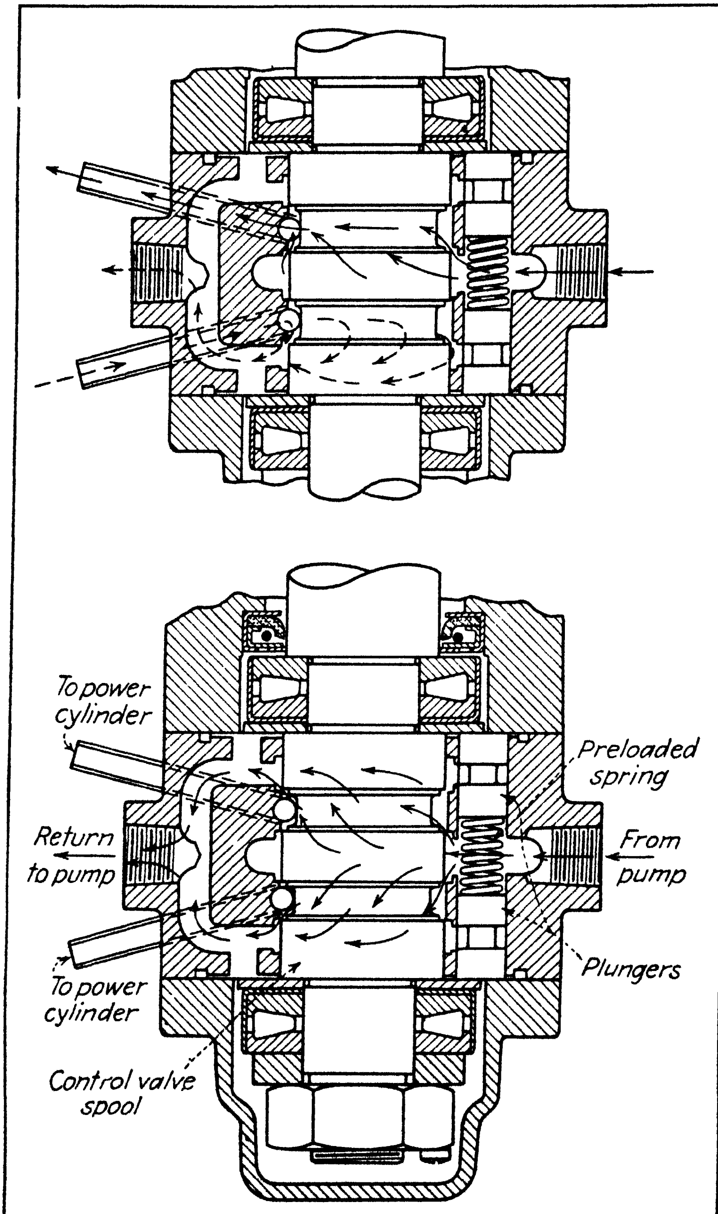


FIG. 26.—CONTROL VALVE OF HYDRAULIC STEERING UNIT.

at the right, passes through both grooves in the valve, through ports cored in the valve housing, and out through the outlet connection to the pump.

If now the steering wheel is turned for a right turn, as shown in the view at the top, the reaction between the cam and lever of the steering gear causes the cam and the piston valve to move downward. Oil from the inlet to the valve housing then can get only into the upper groove on the valve. There it enters a passage from which there is pipe connection to one end of the power cylinder. Oil returning from the other end of the power cylinder passes through the lower groove of the valve to the outlet connection, and thence to the pump. For a left turn the piston valve is moved upward, and the roles of the two grooves in the valve are then reversed, the lower groove passing oil to the power cylinder, and the upper groove, oil returning from the power cylinder to the pump. As soon as the pull on the steering wheel is released, the valve is returned to the central position by the centering springs, and the pressures in opposite ends of the power cylinder then become equalized. The pump used with this system is fitted with a relief valve which holds the hydraulic pressure to a maximum of 800 psi. The hydraulic system functions similarly when the steering gear is subjected to road shocks, and prevents kick-back at the steering wheel.

The Gemmer "Hydraguide"—Gemmer Manufacturing Company in 1951 brought out a power steering gear for passenger cars which was first offered on some large Chrysler models that same year. It consists of two basic units, a standard worm-and-roller mechanical gear combined with a hydraulic power unit in the same housing, and an Eaton engine-driven, rotor-type pump combined with an oil reservoir and filter. In operation, any slight motion of the steering wheel in either direction admits oil under pressure to one of two hydraulic cylinders, and a definite, large proportion of the power required to turn the road wheels is then supplied by the hydraulic unit.

Fig. 27 is a section of the steering gear through the worm axis and includes a side view of the power cylinders. As may be seen from this drawing, the shaft of the steering wheel is not connected directly to the worm of the steering gear, but is in driving relation with it through a shaft coupling *A* comprising a rubber cushioning member, an intermediate or valve-control shaft *B*, and a pair of spur gears *C,D*. Shaft *B* is supported in a spherical bearing *E* at its upper (right) end, but has no bearing at its lower (left) end, and is said to float. The steering mechanism, owing to its connection to the road wheels, may be considered to be always under load, and when the steering wheel is moved from the neutral position in either direction, a force is set up between the teeth of gears *C* and *D*. This force tends to move the lower end of shaft *B* in a direction parallel with it. The shaft cannot move in the plane through the axes of the two gears, because it is restrained by an eccentrically mounted back-up roller *F*; but it can move transversely, that is, at right angles to the plane of the paper. Back-up roller *F* is in contact with the valve-operating block *G*, which latter has a bearing on shaft *B*. The shaft and the block therefore have a transverse motion imparted to them by rotation of the steering wheel, and it is that motion which

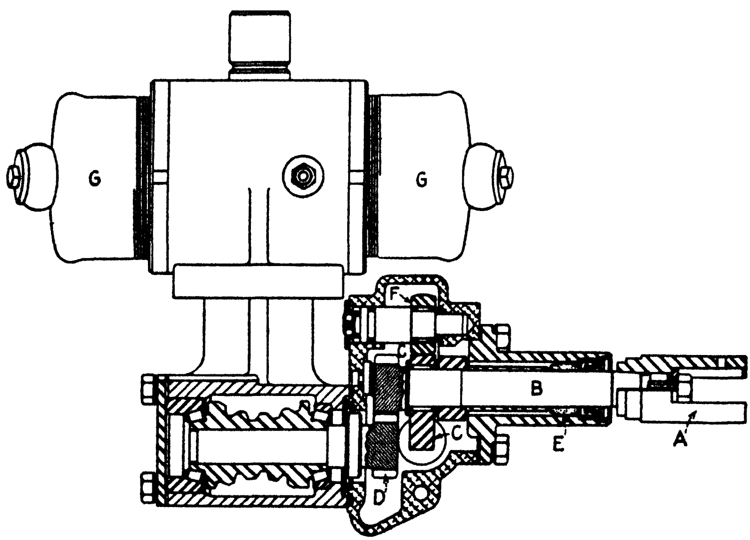


FIG. 27.—GEMMER HYDRAGUIDE, SECTION THROUGH WORM AXIS.

controls the hydraulic valves, whose pistons are in contact with valve block *G*. The mechanical work done in swinging the road wheels around their pivots is always contributed to in a definite proportion by manual effort and hydraulic power.

As shown in Fig. 28, the drawn-steel power cylinders are screwed into the gear housing, and are locked therein by spanner nuts. The two pistons are connected by a yoke, which holds them in permanent alignment, and are provided with composite packing rings comprising a T-section ring of synthetic rubber, backed up on each side by two split laminated phenolic rings. The object of the phenolic rings is to prevent destruction of the rubber rings by extrusion. The

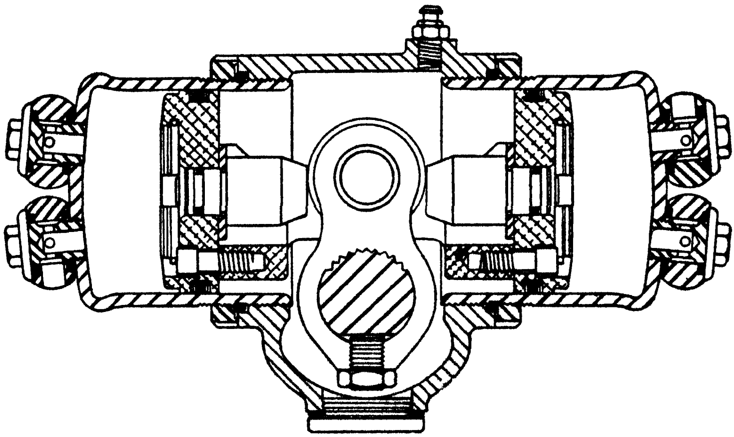


FIG. 28.—GEMMER HYDRAGUIDE, SECTION THROUGH CYLINDERS.

hydraulic pressure on the piston is transmitted to the power arm (shown in the center in Fig. 28) by hardened steel pins in contact with a needle-bearing roller on the arm. Attention may be called to the method of securing the power arm to the steering-gear shaft, by means of serrations extending one-fourth around the shaft, and by a set screw with jamb nut. This connection makes misassembly impossible, and the form

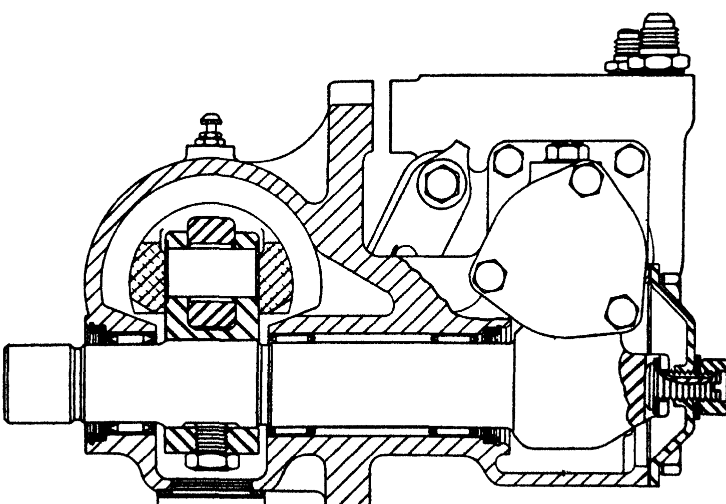


FIG. 29.—GEMMER HYDRAGUIDE, SECTION THROUGH AXIS OF PITMAN SHAFT.

of the serrations minimizes weakening of the shaft. The high-pressure oil lines connect to the cylinder heads through banjo connectors with seal rings. From Fig. 29 it can be seen that all heavily-loaded bearings of the gear are of the needle or quill type.

Referring to Fig. 30, the valve housing (which may be either of aluminum or cast iron) has valve sleeves with ports set into it. Within each valve sleeve there is a piston that is bored out from one end, and the interior of the piston communicates with grooves in the outer surface through radial holes. There are two valves for each power cylinder. The distributor valve directs the oil to the proper cylinder, while the reaction valve controls the flow of oil and the pressure

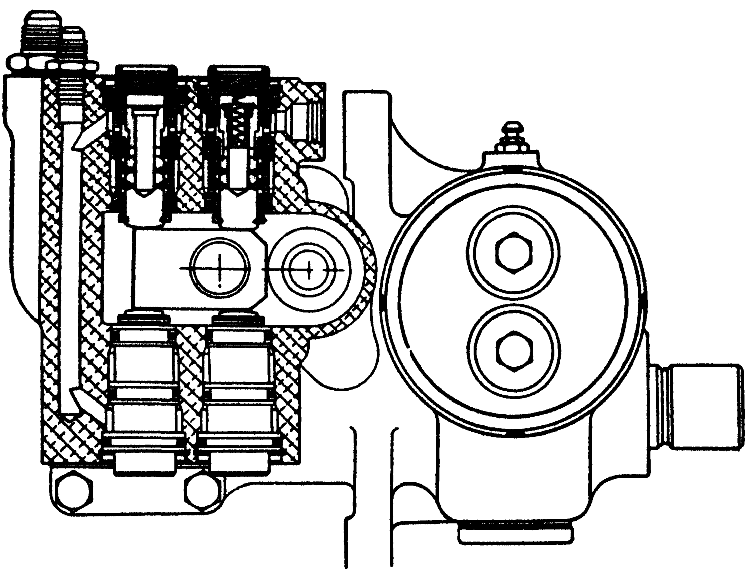


FIG. 30.—GEMMER HYDRAGUIDE, SECTION THROUGH VALVES.

within the cylinder, and thus determines the proportion between the manual and the hydraulic power applied to the gear.

From the foregoing description it is obvious that the distributor valve must be fully opened before the reaction valve begins to move. How the proper timing is effected may be explained with reference to the diagrams of Fig. 31. The shaft center is closer to the center line of the reaction valve, and the valve block reacts against the adjustable eccentric roller, which prevents backlash between the gear teeth. Each valve has a collar which bottoms on the valve sleeve when the valve is fully closed. The two distributor valves communicate with each other through a passage in the housing.

With the steering gear in the neutral position, all of the valves are open, and the fluid passes through them freely. The first slight movement of the valve-gear shaft closes one distributor valve and allows the other to open farther under the unbalanced hydraulic pressure on it. Only 0.003 in. movement of the valve is required to start application of the

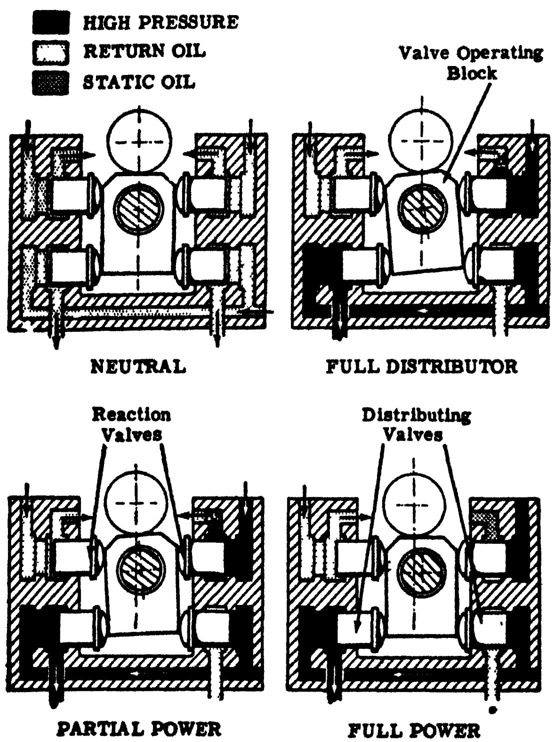


FIG. 31.—GEMMER HYDRAGUIDE, DIAGRAMS OF VALVE ACTION.

hydraulic power. Since the closed distributor valve has bottomed, any farther movement of the valve block moves the reaction valve in the same direction. This first throttles the flow of oil through that valve and then practically shuts it off, thus allowing the pressure in the power cylinder to build up fully.

The pump and oil reservoir on the engine are connected to the steering gear by flexible lines, and two valves are incorporated in the line between the two assemblies, a pressure-relief valve and a flow-control valve. The latter is a spring-loaded piston valve with an orifice in it, and its object is to render the rate of oil flow to the cylinders independent of engine speed, thereby reducing the consumption of mechanical power. The oil reservoir on top of the pump is provided with a standard replaceable oil filter. Owing to the fine orifices and ports in the valves, it is necessary to keep the circulating oil quite clean. No. 10W oil is used as the working fluid, and it takes 3 pints to fill the system.

With power steering a smaller ratio is used in the gear. In the Chrysler application the ratio is 16:1, and the proportion between manual effort and hydraulic pressure in the operation of the gear is 12:88. Thus, as compared with a conventional steering gear with a ratio of 26:1, the steering-wheel motion required to produce a certain steering effect is reduced nearly 40 per cent, and the effort on the steering wheel, 75 to 80 per cent. When the engine is not running, or if the hydraulic system should fail for any reason, the car can be steered by manual effort, as usual. After a turn has been made the car straightens out again automatically, the same as with the conventional gear.

A hydraulic steering gear for motor vehicles has been developed also by Vickers, Inc., of Detroit.

Mechanical Steering Unit—The engine is the natural source of power for operating the steering gear, but engine power can be either applied directly through a power take-off and friction clutches, or it can be converted into hydraulic, electric, or pneumatic power and applied as such. One of the first power units for steering gears was known as the Bethlehem torque amplifier, and was sponsored by the Bethlehem Steel Company. It was exhibited on a Stutz car at the S.A.E. summer meeting in 1927 by its inventor, H. W. Nieman. The device comprised two adjacent band clutches in drums concentric with the steering post. The clutches had a strong self-energizing or wrapping effect. With the steering wheel in the neutral position, both clutches were disengaged. Turning the wheel in one direction engaged one of the clutches, while turning it in the opposite direction engaged the other. Both clutch drums had gear teeth on their periphery and were driven mechanically from the engine. The brake bands had their fixed end secured to the steering shaft. A British patent on a power unit making use of the same general principle was issued in 1943.

Steering Tracklayer Tractors—Tracklayer tractors can be steered either by applying a driving force to one and a retarding force to the other track, or by driving the two tracks from the same power source through gear trains of different ratios, so that the two will move forward at different speeds. The first method is in common use on farm and low-speed industrial tractors, while the second was specially developed for high-speed military tractors. In either case the steering units are incorporated in the drive from the torque converter or transmission to the track-driving wheels. The system usually comprises a pair of bevel gears through which power is transmitted from the main drive shaft of the transmission to a transverse shaft. In one system, illustrated diagrammatically in Fig. 32, the transverse shaft is divided, its center section connecting to the two outer sections through friction

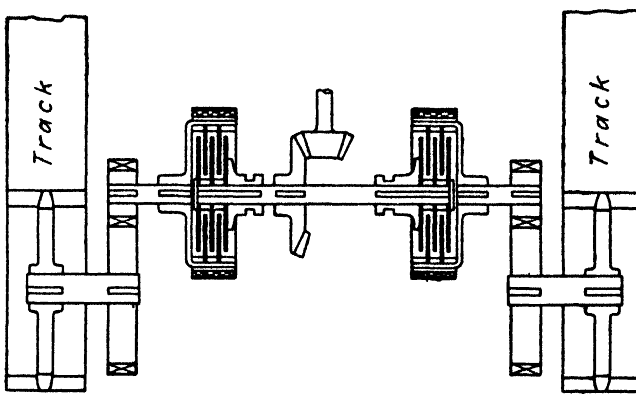


FIG. 32.—DIAGRAM OF A "CLUTCH-BRAKE" STEERING GEAR FOR TRACK-LAYERS.

clutches. A brake is required on each outer section of the shaft, and can be conveniently combined with the driven member of the friction clutch. Normally both friction clutches are engaged, and both track-driving wheels are rotated at the same speed, so that the tractor follows a straight course. If now the driver wants to make a turn, say to the right, he disengages the right friction clutch. Driving force is then applied to the left track only, and this alone will cause the tractor to turn to the right; but if a relatively sharp turn is to be made, the brake on the right side is applied in addition. If the brake is applied with sufficient force to lock it, the tractor will turn around the center point of the right track as an axis, and this corresponds to the minimum turning radius. More gradual turns can be made by allowing the brake to slip.

A steering gear of this type, as used on Caterpillar tractors, is shown in Fig. 33. Each of the steering clutches must transmit torque equal to several times the engine torque, and multiple-disc type clutches are used, which have great torque capacity. Each clutch is operated by a separate lever, and each brake by a separate pedal.

Differential-Brake Steering—Instead of transmitting engine power to the two tracks through separate friction clutches, of which one must be disengaged whenever it is desired to deviate from the straight course, it can be transmitted through a differential gear. In that case steering is

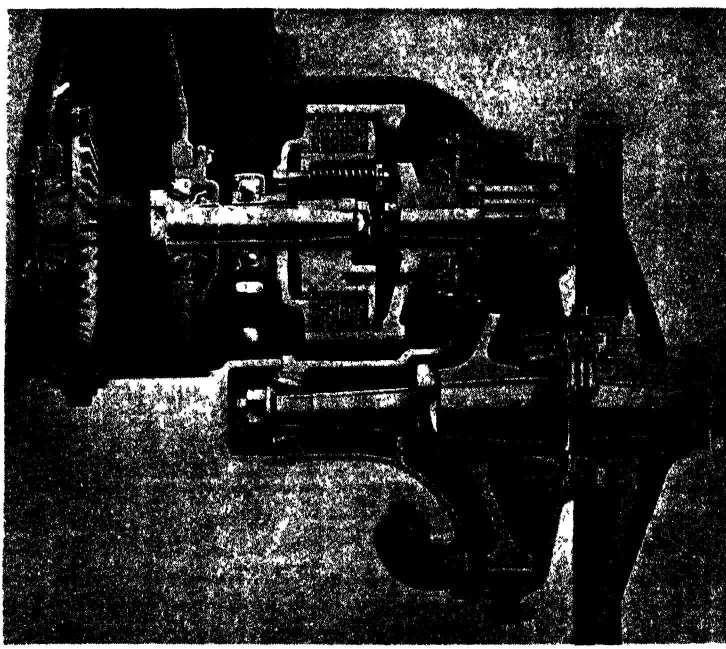


FIG. 33.—CATERPILLAR "CLUTCH-BRAKE" STEERING GEAR.

effected by applying a brake—directly or indirectly—to one of the differential shafts. If the brake locks, all power is applied to the track driven through the other differential shaft, and the tractor will then execute the shortest possible turn.

A difference between this differential-brake method and the clutch-brake method of steering is that whereas with the latter the reduction ratio between the transmission main drive shaft and the track-driving shaft is the same while making a turn as when following a straight course, with the differential-brake method the reduction ratio is halved, so that for a given drive-shaft speed the tractor will turn twice as fast, but can develop only one-half as much maximum propelling effort. This, however, does not seem to involve any difficulty, because in most kinds of heavy farm work, such as plowing, turns are made without any appreciable drawbar load.

A tractor steering gear of the differential-brake type, as used on Cletrac tractors, is illustrated in Fig. 34. The differential gear in this case is of the spur type, and the brake drums, instead of being mounted on the differential shafts,

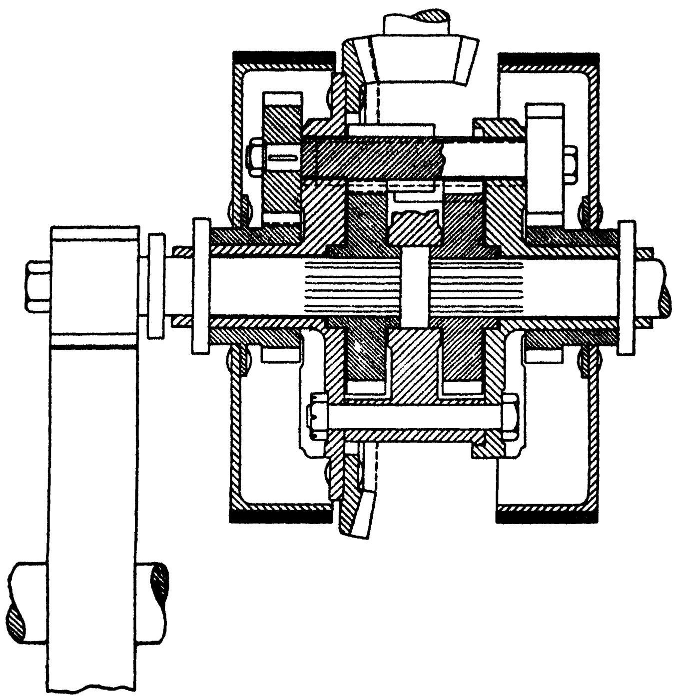


FIG. 34.—CLETRAC STEERING MECHANISM.

are mounted on hollow stub shafts concentric therewith, which connect by gearing to the differential side gears. This method of arranging the brake drums results in a reduction in the brake force required in the proportion of the numbers of teeth in a differential side gear and a pinion respectively. When the brake is applied to one of the drums, the corresponding side gear and the track driven through it slow up, and the tractor begins to turn. Both brakes are applied by means of a steering wheel.

A tracklayer-type tractor consumes considerable power while making a turn, because of the large contact area between the track and the ground, which involves a good deal of sliding or skidding motion. There is a further loss of power in the brakes and clutches as long as these are allowed to slip. These losses are of no great importance in farm work, because of the relative infrequency of turns, but in the case of high-speed military tractors this low efficiency on turns may have serious consequences. Moreover, steering such a machine by disconnecting one of its tracks from the engine involves a certain risk. Imagine a high-geared road tractor to be coasting down a hill with its throttle closed. If there should be a turn in the road and the driver disengage one of the steering clutches to change his course, the tractor is apt to turn in the opposite direction to that required. This effect is sometimes referred to as "reverse steering." It is due, of course, to the fact that the engine, being throttled, instead of speeding up

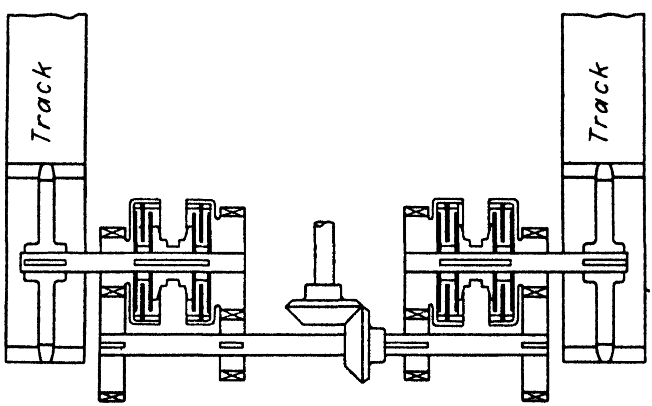


FIG. 35.—STEERING GEAR FOR HIGH-SPEED TRACKLAYERS.

the track to which it is connected, acts as a brake and actually slows it down relative to the other track.

A method of steering applicable to this type of tractor is shown in diagram in Fig. 35. The primary transverse shaft connects to each of the two secondary shafts, or bull-pinion shafts, through two pairs of gears of different ratios, each pair being engaged by means of a friction clutch. For straight-ahead motion the two shafts are driven through gear trains having the same ratio. When it is desired to make a turn, the track on the outside of the turn is driven through the pair of gears with the higher and that on the inside through the pair with the lower ratio.

CHAPTER VI

Propeller Shafts and Universal Joints

The transmission of power from the gearbox to the rear axle or the rear wheels involves certain difficulties, because of the change in relative position of the two units under spring action. There are essentially two methods of transmitting power from a spring-supported gearbox to the rear axle or rear wheels—by chain or chains, and by a propeller shaft with universal joints. The chain came into use first, but it has now been almost entirely superseded by the propeller shaft. The problem of enclosing a motor-vehicle drive chain in a dust- and oil-proof housing was never fully solved, and an exposed chain wears rather rapidly. With shaft drive all bearing surfaces can be protected by dustproof housings, so they can be properly lubricated.

Arrangements of Shaft Drive—There are two principal arrangements of shaft drive. In one of these the propeller shaft is surrounded by a torque tube which is rigidly secured to the axle housing and supported at its forward end on the transmission housing or a frame cross member by means of a spherical joint. The torque tube then transmits the driving thrust from the rear axle to the frame, and it also takes the reaction to the rear-axle torque. With this arrangement only a single universal joint is required in the propeller shaft, at its forward end, and must be enclosed in the sperical joint, concentric therewith. Instead of a torque tube, a torque arm may be used, alongside of the propeller shaft, rigid with the rear-axle housing and pivotally connected to the frame at its forward end on a transverse axis passing through the center of the universal joint, but this construction is now rarely used.

In the other arrangement the driving thrust of the rear axle is transmitted to the frame through the forward ends of semi-elliptic springs or through radius rods or torque arms, all of which are generally considerably shorter than the propeller shaft. In that case the propeller shaft must be able to move angularly with relation to the bevel-pinion shaft to which it connects, and a universal joint is required at the rear end also. Driving thrust and torque can be taken on separate members, the thrust on radius rods with pivotal connections to both the axle housing and the frame, and the torque on a torque arm rigid with the axle housing and linked to the frame at its forward end, but this arrangement also is now rarely met with.

Shaft Whirl—In ordinary power-transmission practice, the stresses in shafts are proportional to the torque transmitted, and the maximum torque which a shaft will safely carry varies as the cube of its diameter, and as the elastic limit of its material. However, propeller shafts of modern road vehicles often are quite long, and they also occasionally operate at very high rotative speeds. Under these conditions the shafts tend to "whirl," and the material may then be subjected to bending (tensile and compressive) stresses which far exceed the shearing stress due to the torque transmitted. Whirling occurs at certain critical speeds, which are the higher the smaller the mass of the shaft and the greater the moment of inertia of its section. For this reason, propeller shafts that must be quite long are always made tubular. Whirling ordinarily is induced by a slight eccentricity or lack of balance. To reduce the tendency to whirl, the shaft should be perfectly balanced. The tubes of which propeller shafts are made now are often rolled up from flat stock and seam-welded, as this ensures a more nearly uniform wall thickness, and therefore better balance than is obtainable with drawn tubing. It is also necessary that the ends of the shaft be so supported that the axis of rotation coincides as nearly as possible with the longitudinal mass axis.

Critical Speeds—Trouble from whirling of propeller shafts was experienced first when unit powerplants were adopted for passenger cars, with which the propeller shafts had to be longer than with separately-mounted transmissions. Whirling of shafts was not an entirely new phenomenon in engineering, however, for it had been observed previously in steam turbines, which rotate at much higher speeds. The theory of whirling motion, which was developed when whirling first occurred in high-speed mechanisms, leads to the following equation for the critical speed of a round tubular shaft:

$$n_c = 4{,}800{,}000 \frac{\sqrt{d^2 + d_1^2}}{L^2} \text{ rpm},$$

where d is the outside diameter; d_1, the inside diameter, and L, the length of the shaft, all in inches.

Propeller shafts for passenger cars usually are so designed that their theoretical critical speed is about 60 per cent higher than the peaking speed of the engine. This makes sufficient allowance for the fact that the actual critical speed usually is somewhat lower than the calculated one, and the further fact that engines often are operated at higher than their peaking speeds, which are not the maximum speeds but the speeds at which the engines develop their maximum power. A certain car for which the author has the data has a propeller shaft 51 in. long and consisting of a tube of 2.25 in. outside and 2.06 in. inside diameter. The theoretical critical speed of this shaft is

$$4{,}800{,}000 \frac{\sqrt{2.25^2 + 2.06^2}}{51^2} = 5630 \text{ rpm.}$$

The engine of this car peaks at 3600 rpm.

A similar margin should be allowed in the case of trucks and buses, for although the engines of such vehicles usually are governed, and cannot exceed their governed speed while under load, they can exceed that speed, and even the peaking speed, while coasting. In trucks of large capacity the distance from the rear end of the transmission to the rear axle is always so great that a propeller shaft connecting the transmission main drive shaft directly to the rear-axle pinion shaft would be entirely too long to be safe from the whirling standpoint, and an intermediate bearing is therefore provided, the propeller shaft being divided, as will be explained in more detail further on. In passenger cars of long wheelbase an extension is sometimes built onto the rear of the transmission housing, through which the transmission main drive shaft is passed, with the object of reducing the required length of propeller shaft.

Propeller-Shaft Assemblies—The propeller shaft with joints at both ends—either two universal joints or one universal and one plain splined joint—forms an assembly which is usually made by a parts manufacturer. As propeller shafts have to be made of large diameter to prevent whirling, the necessary torsional strength can be readily obtained with ordinary material. The predominant practice is to use seam-welded tubing, the tubes being rolled up from flat stock of low-carbon steel, and gas- or electric-welded. The steel contains from 0.10 to 0.25 per cent carbon, which welds better than steel with higher carbon content. This material has an elastic limit in shear of about 34,000 psi. Where the risk of

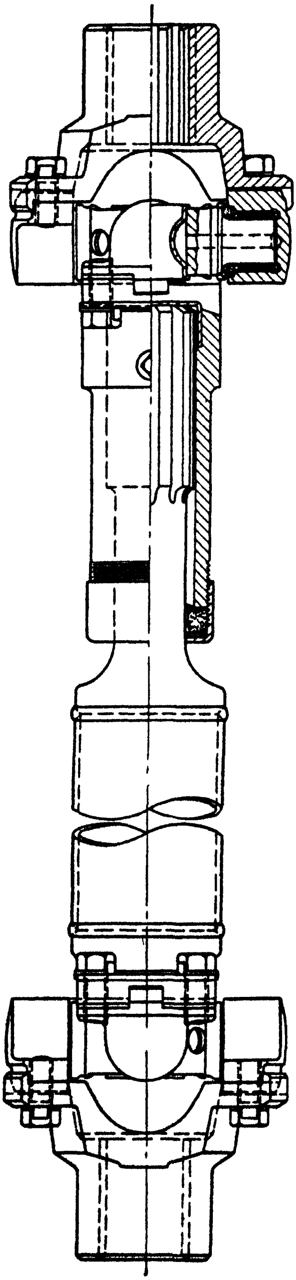


FIG. 1.—PROPELLER-SHAFT AND UNIVERSAL-JOINT ASSEMBLY (MECHANICS).

injury by whirling is low (i.e., where the shaft is short and operating speeds are low), and where at the same time the torque to be transmitted is high, drawn tubing of 1040 steel can be used to advantage, which has an elastic limit in shear of about 50,000 psi. In calculating the necessary section, it is advisable to allow a factor of safety of 3, and 2 is the absolute minimum. Propeller shafts furnished by specialists are given a torque rating which represents the torque or moment required to stress them to the elastic limit. This torque or moment is given by the equation $M = ZS$, where Z is the section modulus of the shaft section, and S the elastic limit in shear. The section modulus of a hollow circular section is

$$Z = 0.196 \frac{d^4 - d_1^4}{d} \text{ in.}^3,$$

where d is the outside and d_1 the inside diameter of the tube, both in in. Hence,

$$M = 0.196S \frac{d^4 - d_1^4}{d} \text{ lb-in.}$$

This value divided by the factor of safety should be equal to the maximum torque theoretically applied to the shaft; that is, the product of the maximum engine torque by the low-gear ratio of the transmission.

Slip Joint—At the ends the propeller-shaft tube usually is welded to solid stub shafts of alloy steel. These shafts are turned down at one end to form a pilot that enters the tube under pressure, an interference of 0.010 to 0.020 in. being allowed. A scarf is formed between the end of the tube and a shoulder on the stub shaft, and the two are joined by seam-welding all around. The stub shaft at the forward end of the propeller shaft has splines cut on it, and forms one member of a slip joint. Ten-spline joints are now commonly used for this purpose, with splines of maximum radial depth (0.095 times the outside diameter). To provide sufficient bearing surface, the length of the splined portion is made 1.5 to 2.0 times the outside diameter. After propeller shafts have been assembled with their joints, they are tested for run-out, and carefully balanced dynamically. A typical propeller-shaft assembly is shown in Fig. 1.

Improved Propeller-Shaft Design—For vibrationless operation at high speeds, a special design of propeller shaft has been developed by Mechanics Universal Joint Division of Borg-Warner Corporation, and a drawing of one end of this

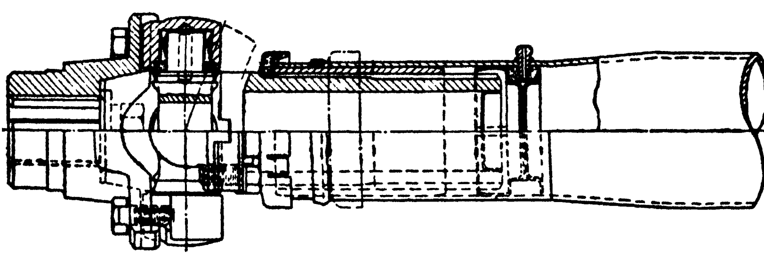


FIG. 2.—PROPELLER-SHAFT ASSEMBLY FOR HIGH-SPEED OPERATION.

assembly is shown in Fig. 2. In the conventional design, as represented by Fig. 1, the greatest concentration of mass is at the ends, and particularly at the slip joint. Vibrating forces are proportional to the product of the unbalanced mass by the distance of the center of mass from the center of support at the ends. Balance, therefore, is dependent on accurate alignment, reduction of the mass at the ends, and a reduction of the distance between the center of mass and the center of support.

In the new design the conventional stub shaft is eliminated, and replaced by a thin-walled tube with internal splines mating with splines on the outside of the slip yoke. In this way the total weight is reduced about 2 lb, and the offset of the center of mass from the center of support materially reduced.

Accurate alignment, which also contributes to good balance, is readily obtained, because in this design the heaviest element is the slip yoke, which can be accurately machined. The accurately machined slip sleeve is a press fit in the tube over its entire length; the weld at the end serves merely to lock it in place, and has no effect on the alignment (which it has in the conventional design). As in this assembly the splines are on the outside of the slip yoke, instead of on the inside, their mean diameter is materially greater, and 32 or 36 splines are used, instead of the usual ten. This, in combination with the increased diameter, cuts the unit pressure almost in half, and the reduced bearing pressure makes it possible to reduce the hardness, and machine all parts after they have been heat-treated.

Universal Joints—A universal joint is a device which makes it possible to transmit motion or power from one shaft to another whose axis lies in the same plane but makes with that of the first shaft an angle which may vary during operation. Universal joints sometimes are used also between shafts whose angular relation is fixed, but there they are not needed, as a pair of bevel gears could be used. The most important application of universal joints in the automotive field is in the propeller shaft between the transmission and the rear axle. Every shaft-driven car with conventional springing has at least one universal in its propeller-shaft line, and many have two. In vehicles with front drive (and steering) a universal joint is required in the front axle at each steering knuckle, in addition to the joint or joints in the propeller shaft, and in four-wheel driven vehicles they are needed in both propeller shafts and in the steering axle.

Invention of the universal joint is generally ascribed to Jerome Cardan, an Italian mathematician who lived during the sixteenth century, and in Continental countries the device is known as Cardan's joint. The British, however, credit its invention to Robert Hooke, and call it Hooke's joint. As Cardan lived more than a century earlier than Hooke, priority undoubtedly must be conceded to him, though Hooke may well have invented the joint independently.

As will be shown further on, the ordinary Cardan or Hooke's joint has the peculiarity that it transmits motion irregularly, so that if the driving member turns at uniform angular velocity, the driven member will turn non-uniformly —faster than the driving member during two periods of a revolution, and slower during the intervening periods. This is a serious disadvantage when the joint is used to connect rotating parts of great moment of inertia and turning at relatively high speed, especially when the angle between the connected shafts may attain large values. For such applications a special type of constant-velocity universal joint has been developed.

The common universal joint comprises two yokes at right angles to each other, and an intermediate member which may be either a cross, a block, or a ring. In operation the yokes swing around trunnions formed on them or on the intermediate part, hence one diameter of the intermediate part (one bar of the cross) rotates in a plane perpendicular to the driving yoke, while the other rotates in a plane perpendicular to the driven yoke. Fig. 3 shows a simple universal joint of the ring type, the intermediate part consisting of two rings bolted together, which between them form bearings for four trunnions on the yokes. Automotive universal joints have been highly refined and differ materially from that shown. The

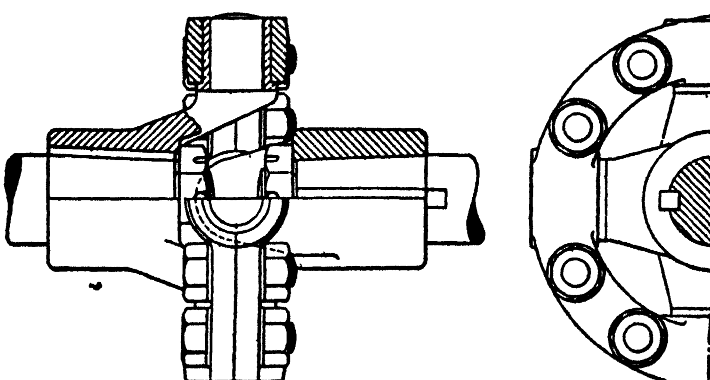


FIG. 3.—SPLIT-RING-TYPE OF UNIVERSAL JOINT.

refinements serve to ensure effective lubrication under service conditions and to enable the joints to carry greater loads; they do not change the kinematic principles involved, and it is therefore permissible to use illustrations of the simplest type to develop the mathematical theory of the device.

Periodic Speed Fluctuations—To gain an idea of the magnitude of the variations in angular velocity, we will assume a universal joint connecting two shafts in a vertical plane, the axis of the driving shaft being horizontal, as shown in Fig. 4. The axes *AA* and *CC* of the two pairs of trunnions intersect each other at right angles. When motion is being transmitted by the joint, line *AA* describes a circle in a vertical plane, while line *CC* describes a circle in a plane making with the vertical an angle ϕ equal to the angle between the shafts. These two circles are great circles of the same sphere. Points A and C always remain the same distance apart, viz., one quadrant of a great circle. The difference between the angular speeds of the driving and driven yokes is greatest when one of the trunnion axes is in the plane of the shafts and the other perpendicular thereto. With the joint in the position indicated in Fig. 4, when axis AA of the trunnions on the driving yoke is in the plane of the shafts, the angular velocity of the driven yoke is greater than that of the driving yoke, and when it is perpendicular to that plane, the angular velocity of the driven yoke is smaller. There are four points in each revolution at which the angular velocities of both yokes are equal, located midway between the points of maximum speed increase and decrease, respectively.

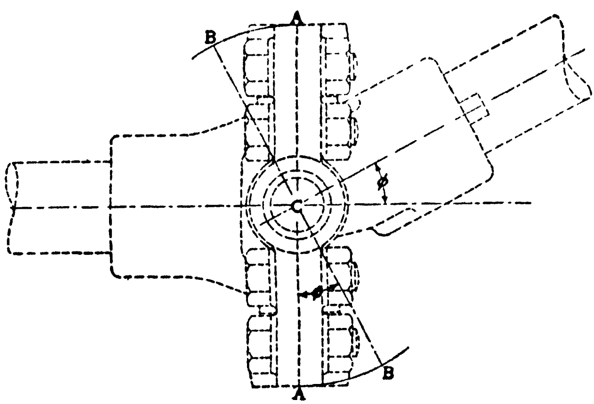


FIG. 4.

Let the two large arcs in Fig. 5 represent the great circles described by ends A and B of the two trunnion axes, respectively. Let point A travel from the point of intersection of the two arcs to point A', and point B to point B', which latter is determined by the fact that $A'B'$ must be a quadrant. Now lay off from point B on the arc of travel of that point a quadrant or 90°, which gives point C. Through A' and C draw an arc of a great circle. Angles $B'A'C$ and $B'CA'$ are both right angles, because their opposite sides are quadrants, hence angle ACA' is a right angle. We therefore have a right-angle spherical triangle $AA'C$, the angle $A'AC$ of which is equal to the angle between the connected shafts, the side AA' of which represents the angular motion of the driving shaft, and the side AC the angular motion of the driven shaft during a short period after A has passed through the point where the trunnion axis of the driving yoke is perpendicular to the plane of the two shafts.

According to a theorem of spherical trigonometry,

$$\cos A'AC = \tan AC \cot AA',$$

and since the tangent is the reciprocal of the cotangent,

$$\frac{\tan AC}{\tan AA'} = \cos A'AC.$$

Also, as for very small angles the tangents are proportional to the angles, we have

$$\frac{AC}{AA'} = \cos A'AC.$$

Therefore, when the trunnion axis of the driving yoke is perpendicular to the plane of the connected shafts, the angular velocity of the driven shaft is smaller than that of the driving shaft in the proportion of the cosine of the angle between the shafts to unity. In a similar way it can be shown that when the trunnion axis of the driving yoke is in the plane of the connected shafts, the driven shaft turns faster than the driving shaft in the inverse proportion.

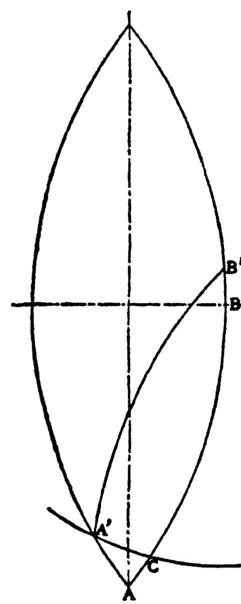


FIG. 5.

It is desirable to have an expression for the ratio of angular velocities at any point in the revolution of the driving yoke. To simplify the mathematical expressions we will denote the angle $A'AC$ by ϕ, the side AA' by a, and the side AC by b. We then have, as before,

$$\tan b = \cos \phi \tan a.$$

Differentiating we get

$$\sec^2 b\, db = \cos \phi \sec^2 a\, da,$$

and

$$\frac{db}{da} = \cos \phi \frac{\sec^2 a}{\sec^2 b},$$

which gives the ratio of instantaneous angular velocities in terms of ϕ, a and b. It is preferable, however, to express the ratio in terms of ϕ and a alone, as b is not known, and, moreover, can be readily eliminated. Squaring the equation for $\tan b$ we get

$$\tan^2 b = \cos^2 \phi \tan^2 a,$$

and adding 1 to each side of the equation,

$$1 + \tan^2 b = 1 + \cos^2 \phi \tan^2 a.$$

But since

$$1 + \tan^2 b = \sec^2 b,$$

$$\sec^2 b = 1 + \cos^2 \phi \tan^2 a,$$

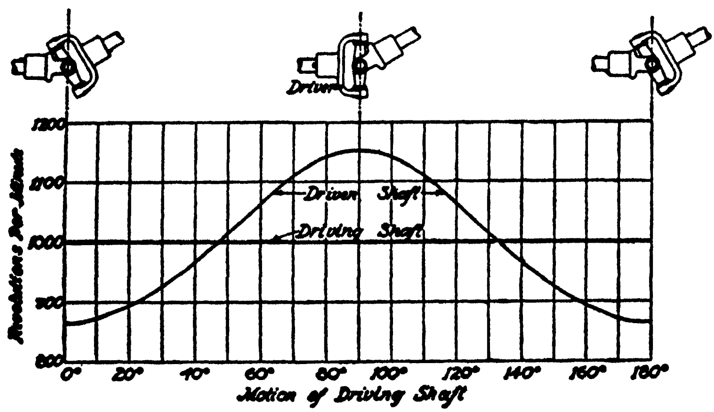


FIG. 6.—VARIATION OF DRIVEN-SHAFT SPEED (ANGLE BETWEEN SHAFTS, 30°).

and substituting this value of $\sec^2 b$ in the equation for db/da we have

$$\frac{db}{da} = \cos \phi \frac{\sec^2 a}{1 + \cos^2 \phi \tan^2 a},$$

which gives the ratio of the angular velocities after any angular motion a of the driving shaft from the zero position, in which latter the trunnion axis of the driving yoke is perpendicular to the plane of the two shafts. When $a = 0$ the equation reduces to

$$\frac{db}{da} = \cos \phi,$$

which is the same as that already found for the position of minimum speed of the driven shaft.

The curve of Fig. 6 shows how the speed of the driven shaft varies during one-half a revolution or 180°, when the driving shaft has a speed of 1000 rpm and the angle ϕ between shafts is 30°. We start from the position where the trunnion axis of the driving yoke is perpendicular to the plane of the shafts. In this position the driven shaft rotates at 866 rpm, its lowest speed. Its speed increases until after a little more than 45° motion of the driving shaft it equals the speed of the latter. It keeps on increasing, and after 90° motion, when the trunnion axis of the driving yoke is in the plane of the connected shafts, it attains its maximum value of 1155 rpm. Then it decreases, according to the same curve, until after 180°, or one-half revolution, it again attains its minimum value of 866 rpm. During one revolution the speed of the driven shaft therefore passes through two maxima and two minima. The average speed, of course, is the same as that of the driving shaft, and the speed fluctuation amounts to

$$\frac{(1155 - 866) \times 100}{1000} = 28.9 \text{ per cent.}$$

The following table gives the fluctuations of driven-shaft speed for different angles between shafts, the speed of the driving shaft being assumed to be constant:

Angle ϕ, Degrees	*Fluctuation, Per Cent*	*Angle ϕ, Degrees*	*Fluctuation, Per Cent*
2	0.15	16	7.9
4	0.5	18	10
6	1.1	20	12.4
8	2	22	15
10	3	24	18
12	4.4	26	21.3
14	6	28	25

Speed fluctuations thus produced can give rise to severe stresses. In a motor vehicle we have at one end of the transmission line the engine, whose speed is maintained substantially constant by the inertia of its revolving parts, and of the clutch and transmission; and at the other end the car which, when running at high speed, also has its speed maintained by inertia. But if transmission were effected through a single universal joint working at an appreciable angle, the speed of either the engine or the car, or both, must of necessity change considerably during a quarter revolution. The inertia of the engine and associated rotating parts would strongly resist such a change in speed, and the car inertia a change in car speed, with the result that all parts of the transmission would be subjected to severe stresses. Not the least to suffer would be the tires, which would slip on the pavement as the wheels tended to accelerate or decelerate. To minimize such stresses the drive must be so arranged that the two shafts are always nearly in line with each other. They can be completely eliminated by using two universal joints in series.

We found that the speed of the driven member is reduced in a certain ratio when the trunnion axis of the driving yoke is perpendicular to the plane of the shafts, and increased in the inverse ratio when the trunnion axis of the driving yoke is in the plane of the shafts. These two positions are 90° apart. Therefore, by arranging the universal joints in series, in such a way that the driving yoke or corresponding member of the second makes an angle of 90° with the driving yoke of the first, and so that the driving and driven shafts make equal angles with the intermediate shaft, motion will be transmitted

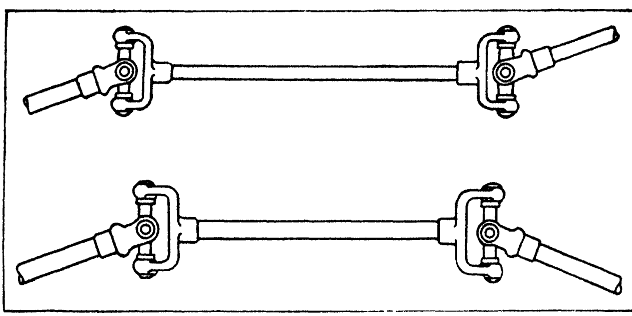


Fig. 7.—Two Arrangements of Universal Joints Ensuring Uniform Transmission of Motion.

uniformly from the driving member of the first universal to the driven member of the second. There are two possible arrangements that meet these conditions, and both are illustrated in Fig. 7. In the first the driving and driven shafts are kept parallel with each other. This is readily accomplished in cars with the so-called Hotchkiss drive, in which the only connection between the driving axle and the frame is through semi-elliptic springs. If the springs are symmetrical (both ends of equal length), the axle under spring action will move perpendicular to the plane of the frame, hence the pinion shaft can be maintained parallel to the frame, and, therefore, to the transmission main drive shaft. In the other arrangement both the driving and driven shafts deviate from the propeller shaft in the same direction, and must be so controlled or guided that both always make equal angles with it. This can be approximated by using on the rear axle a torque arm about half as long as the propeller shaft. With both arrangements the yokes at opposite ends of the propeller shaft must be in the same plane.

Only a single universal joint is used in the propeller shafts of most modern passenger cars. With the transmission combined with the engine, the propeller shaft is necessarily quite long, which tends to reduce the maximum angle between the shafts connected by the joint, and the angle can be further reduced by so mounting the engine (at a small angle to the horizontal) that with the car carrying a normal load, the crankshaft axis produced practically intersects the rear-axle axis, which gives what is known as a "straight-line drive."

Details of Design—The major parts of universal joints are drop-forged of medium-carbon steel and heat-treated. Originally the trunnion bearings were bronze-bushed, but as engine horse power increased, and greater bearing loads had to be carried, hardened-steel bushings were substituted for the bronze bushings. Still later these also proved inadequate, and at present needle roller bearings are generally fitted. The torque capacity of a universal joint usually is limited by the load capacity of its trunnion bearings; it also depends on the speed of rotation, the degree of continuity of the torque, and the angle at which the joint operates. If the connected shafts are nearly in line, there is little motion at the bearings, and power loss and heat generation are small, whereas the reverse is the case if the joint works at large angles.

For joints of similar design the torque capacity evidently varies as the cube of the linear dimensions, because the bearing area and the permissible bearing load vary as the square of the linear dimensions, and the force represented by this load acts on an arm whose length is one of the linear dimensions of the joint. A compact joint is particularly desirable in cars with torque-tube drive, where it must be enclosed in a housing at the forward end of the torque tube. There the unit bearing load under maximum engine torque sometimes runs as high as 3750 psi in direct drive, and close to 10,000 psi in low gear, even with steel-bushed universals. Joints with needle roller bearings have similar torque ratings. These high loadings, however, should be used only if the drive is substantially straight. Where needle roller bearings are used, the cross or other member carrying the trunnions is made of low-carbon alloy steel and case-hardened, and similar material is used for the outer races with which the rollers contact.

Companion Flange—Ordinarily, because the distance between the two shafts connected by the propeller shaft varies with spring action, a sliding joint must be incorporated in the drive. However, there are certain types of universal joint that permit of limited axial motion between their driving and driven members, and these make a separate sliding joint unnecessary. The universal joint at the forward end of the propeller shaft is secured to the transmission main drive shaft, or to an intermediate shaft, while that at the rear end is similarly secured to the shaft of the rear-axle drive pinion. Both of these connections usually are of the splined type, one yoke of the universal having splines broached in its bore, and the shaft on which it is mounted having splines hobbed on it. The yoke is held in a definite axial position on the shaft by a shoulder and spacers on one side of it, and a cap screw and washer on the other. Such an assembly, however, makes it difficult to remove the propeller shaft from the vehicle, and to facilitate service operations, companion flanges are now widely used. As shown in Fig. 8, a companion flange consists of a

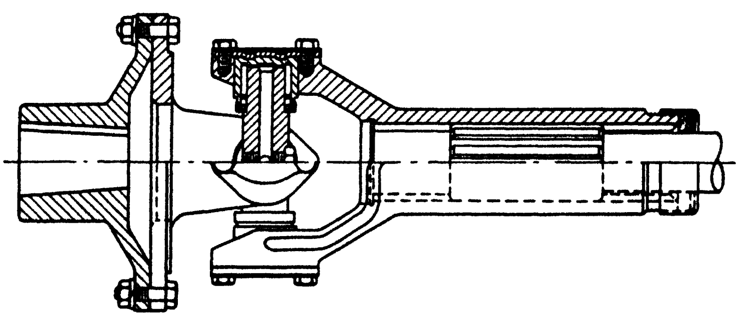


FIG. 8.—NEEDLE-BEARING TYPE OF UNIVERSAL JOINT WITH COMPANION FLANGE (BLOOD BROTHERS).

disc that is either keyed or splined to the shaft, and has a circle of bolt holes near its circumference, by means of which one yoke of the universal joint is bolted to it. This yoke of the universal then also is in the form of a disc or ring, with two lugs projecting from it laterally, bored out to accommodate the trunnions of the joint. With this construction the propeller shaft can be removed from the vehicle after the companion-flange bolts have been taken out. Certain types of universal joint have the lugs accommodating the trunnions made separate and secured to a spider by means of cap screws, and with these a separate companion flange is unnecessary.

Fig. 8 shows a needle-bearing universal joint manufactured by Blood Brothers Machine Company. A roller cup is set into the bore of each arm of the yoke, and is held against angular motion therein by a lock plate secured by cap screws. At the inner end of the rollers there are a roller retainer and a cork oil seal. The center cross is bored out to form an oil reservoir, and oil is introduced through a fitting screwed into the cross at its center. The slip yoke has ten splines, and is provided with an oil-seal disc at one end and a felt-washer oil seal at the other. Another design of needle-bearing joint was shown in Fig. 1.

Sliding-Block Joint—Another type of universal joint, in which motion also takes place around axes at right angles with each other, but which has only one pair of trunnions, is illustrated in Fig. 9. Each of the trunnions carries a ball or spherical block, mounted thereon on a needle bearing, and each ball or block is located in a bore in the body or housing of the joint, which is bolted to a companion flange. In operation, the housing rocks around the trunnion axis, and the spherical blocks slide back and forth in the bores in the housing to produce a rocking motion around an axis perpendicular

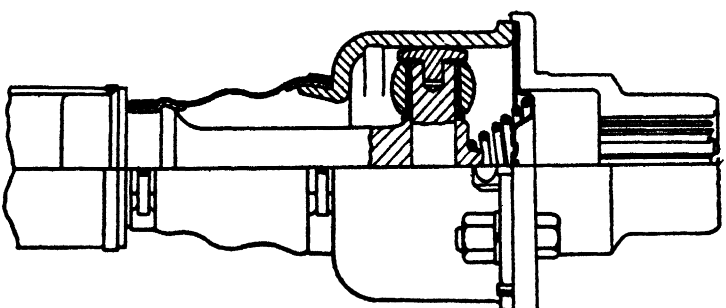


FIG. 9.—SLIDING-BLOCK JOINT.

to that of the trunnions. The principle of operation therefore is exactly the same as that of the joints previously described, but this design has the advantage that it permits a certain amount of longitudinal motion between the connected shafts, and thus obviates the need for a separate sliding joint.

Constant-Velocity Universals—The reason the Cardan type of universal does not transmit motion uniformly is that the effective lengths of the lever arms at the points of motion transfer vary continually. Referring back to Fig. 4, assume the driving (left) yoke to rotate at a constant velocity of n rpm. Then point A at the end of the trunnion axis of this yoke will have an absolute velocity of $2\pi nr$ ipm, r being the radius of point A with respect to the axis of the driving yoke. But the radius of point A with respect to the axis of the driven yoke is shorter in this position, being equal to $r \cos \phi$, and as for a given linear velocity the angular velocity is inversely proportional to the length of the radius, with the joint in this particular position the driven turns faster than the driving yoke. In order that, in a universal joint, motion may be transmitted at constant velocity, it is necessary that the point of contact, or of motion transfer, be always at equal radial distances from the axes of both connected shafts. This is the case if the point of contact always lies in the plane bisecting the angle between the connected shafts, as can be shown by the example of a pair of equal-sized bevel gears (Fig. 10). Here the pitchline, near which contact always takes place, is in the plane bisecting the angle between the axes of the two gears, and each point in the pitchline is at equal radial distances from the axes of both gears. As the radial distances are the same, the angular velocities of both gears are always the same.

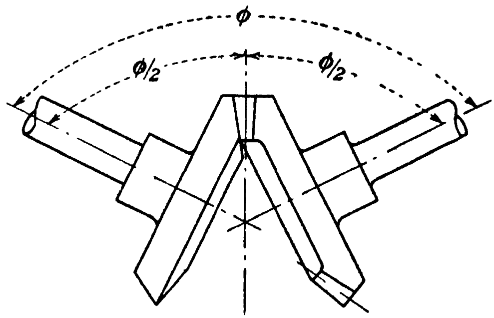


FIG. 10.—ILLUSTRATING PRINCIPLE OF CONSTANT-VELOCITY JOINT.

In a universal joint, of course, it is impossible to transfer motion directly from the driving to the driven yoke or corresponding member, because it must be possible to vary the angle between the axes of the two and still transmit the motion without appreciable backlash. The problem can be solved by means of steel balls interposed between members that are rigid angularly with the driving and driven shafts respectively. It is necessary, however, to so arrange the design that the ball centers always will lie in a plane which bisects the angle between the shafts.

Bendix-Weiss Joint—A constant-velocity joint invented by Carl W. Weiss (a pioneer in the American oil-engine industry), which is now being manufactured by Bendix Products Division of Bendix Aviation Corporation, is illustrated in Fig. 11. It consists of two spiders with two, three, or four arms each, each arm of one spider being located between adjacent arms of the other. Motion is transferred from one set of arms to the other by steel balls lodged in grooves cut in the sides of the arms. To ensure that the balls will always remain in the plane bisecting the shaft angle, the grooves are cut at an angle with the axis of the spider, the two grooves accommodating a particular ball crossing each other. The ball thus is restrained to always remain at the crossing point of the centers of the grooves. When the connected shafts are in line, the centers of all of the balls lie in a plane perpendicular to the shaft axis. If now the driven shaft is swung through a certain angle, the plane of ball centers is automatically shifted through one-half that angle, and the condition of constant-velocity transmission is thus satisfied. An incidental advantage of this type of joint is that it permits of a moderate axial motion between the connected shafts, which ordinarily makes it possible to do without a special slip joint.

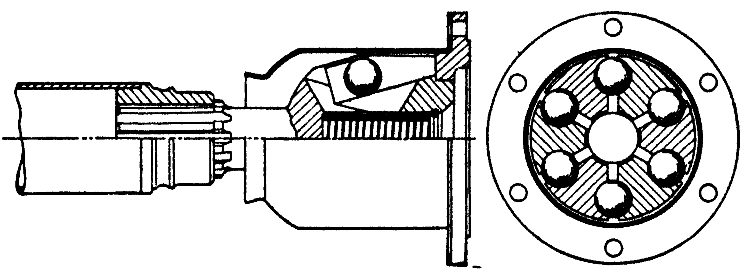


FIG. 11.—BENDIX-WEISS CONSTANT-VELOCITY JOINT.

Rzeppa Constant-Velocity Joint.—Another type of constant-velocity joint, invented by A. H. Rzeppa and manufactured by the Gear Grinding Machine Company of Detroit, is shown in section in Fig. 12. Here the steel balls are lodged in grooves cut on the inside and outside, respectively, of two concentric races with spherical surfaces, keyed or splined to the driving and driven shafts. In this case the ball grooves are parallel with each other, and they, therefore, do not control the angular position of the plane of ball centers. This is accomplished by means of a ball cage, in the shape of a spherical shell, with perforations accommodating the balls, usually six in number. The ball cage is acted upon by the connected shafts through the intermediary of a pin and a pilot, the pin having spherical connections with the pilot and both shafts. Any deviation of the shafts from the in-line position results in angular motion of the pilot and cage around their common center, and by suitably proportioning the relative lengths of the arms of the lever and the distance of its support from the axis of the joint, the angular motion of the ball cage can be made equal to one-half the angular motion of one shaft relative to the other, so that the plane of ball centers always bisects the angle between shafts.

Double Joints—Constant-velocity joints are needed most at the knuckle pivots of combined steering and driving axles, because the angle between knuckle spindle and axle shaft may approach 40°, which is very great; and, moreover, in this case it is impossible to compensate for speed fluctuations due to a universal joint at the knuckle end by placing another at the inner end of the axle shaft. Sometimes double universal joints, similar to the design illustrated in Fig. 13, are used in this location. If with the steering gear in the straight-ahead position the knuckle pivot axis passes through the center of the short intermediate member, both of the individual universals will always operate at equal angles, and the combination therefore will transmit motion at constant velocity. However, the regular constant-velocity joints permit of a more compact construction. Four-wheel-drive trucks designed for use in World War I had a single Cardan-type joint at the steering knuckle, which may have been one of the reasons why they were not very successful. All of the "Jeeps" and other vehicles with driving front wheels designed for use in World War II had constant-velocity joints at the steering knuckles.

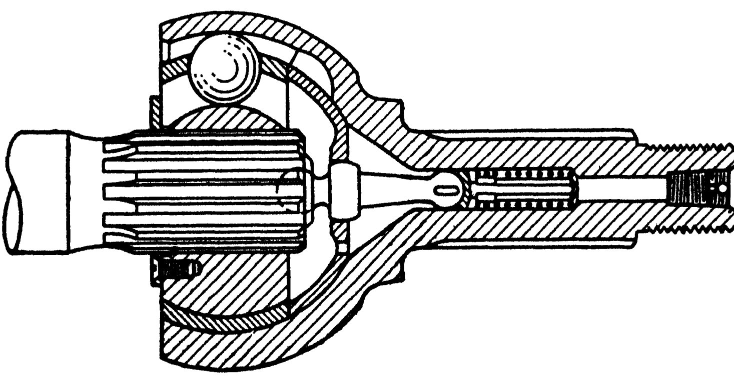


FIG. 12.—RZEPPA CONSTANT-VELOCITY JOINT.

A feature of the "Mechanics" joint shown in Fig. 13 is that the "eyes" containing the trunnion bearings are made separate from the spider, to which they are bolted. Each eye is provided with an integral key engaging into a groove or keyway machined in the arm of the spider, on which the driving force is taken. A circular flange turned on the outer ends of the spider arms keeps the joint concentric, thus preventing or minimizing runout.

Tracta Joint—A constant-velocity joint which also belongs to the "double" type is the Tracta, which was invented by Fenaille in France and is being manufactured in this country by the New Process Gear Corporation. As shown in Fig. 14, it consists of two yokes and two intermediate members, all contacting members being capable of relative motion. As contrasted with other types of universal joint, the sliding

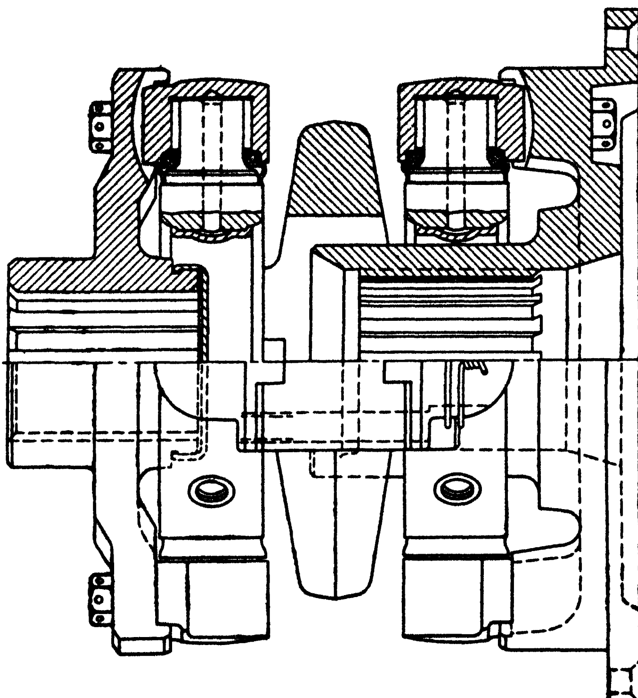


FIG. 13.—DOUBLE UNIVERSAL JOINTS FOR HIGH-ANGLED DRIVES.

motion between contacting parts is in plane instead of in cylindrical surfaces. The arms of each yoke engage into a groove in the adjacent intermediate part, and the two intermediate parts are connected in a similar manner, one being "tongued" and the other "grooved." The two yokes are in the same angular position as in a propeller shaft drive with two universal joints, and it is obvious that any irregularity introduced by one-half of the joint—one yoke and one intermediate member—is compensated for by the other half. This joint is well suited to operation at large angles, as in front-wheel drives. An advantage claimed for it is that all bearing surfaces are large; consequently, the rate of wear should be low. The parts, moreover, can be made of low-cost steel, such as No. 4027, a carburizing-type carbon-molybdenum steel.

Disc-Type Universal Joints—Considerable trouble was experienced with metal universal joints during the early years of the automobile industry, chiefly because the problem of their lubrication had not been solved. As a result, joints comprising flexible steel discs, or leather discs, were proposed,

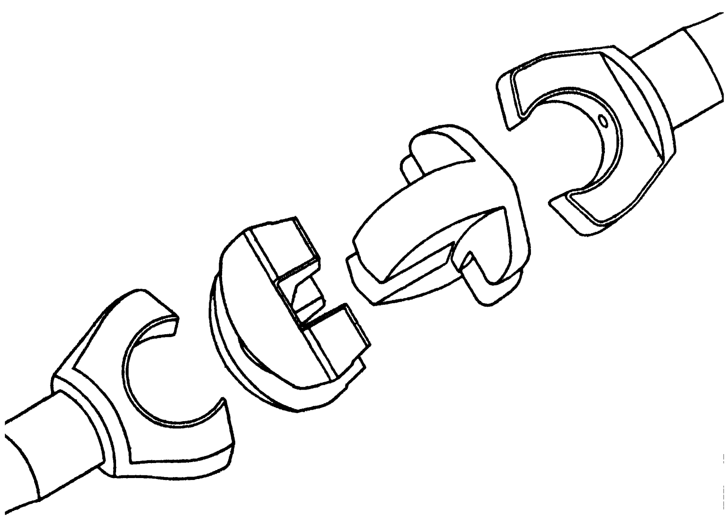


FIG. 14.—TRACTA UNIVERSAL JOINT ("EXPLODED" VIEW).

and used to a considerable extent. Later rubberized fabric discs were substituted for the leather discs, and an important step forward in this type of universal was made in England in 1911, when Edward J. Hardy introduced a special rubberized fabric disc in which the layers of fabric were so piled up that there was uniform angular spacing between the strands of cotton threads of successive layers, which ensured uniform strength of the disc in all directions. A fabric-type universal joint consists essentially of two spiders, usually three-armed, one or more fabric discs or rings between them, and bolts, nuts, and washers by means of which the discs are secured to the spiders. The form of the washers is important, as upon it depends the distribution of stress in the fabric rings adjacent to the bolts.

As compared with the metal type, the fabric-disc type of joint has the advantages that it remains silent under all conditions, and that it requires no lubrication. The type has its limitations, however, and it is no longer being used for passenger cars in this country, though in England it is still met with, especially in the drives of small cars, and even in axle shafts. In early American applications some difficulty was experienced because of excessive end thrust, which injured the discs. Later the chief difficulty arose from inability of the discs to hold the shafts permanently concentric. From what has been said before it will be realized that if the mass center of one end of the shaft is not stationary, but rotates in a circle, the tendency of the shaft to whirl will be greatly increased. This, of course, became more important as engine speeds increased. In British practice, fabric discs sometimes are provided with a metal support at the center, to prevent runout, as shown in Fig. 15. Fabric universals are suitable only for

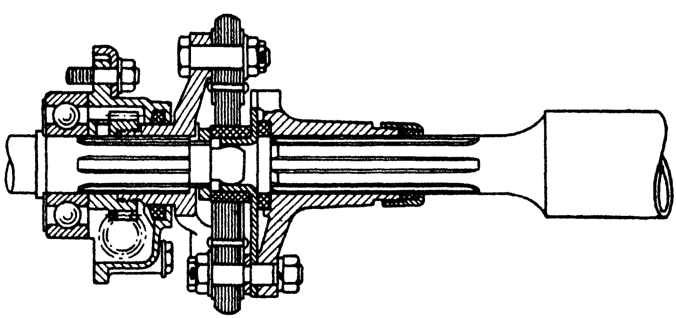


FIG. 15.—FLEXIBLE-DISC TYPE OF UNIVERSAL JOINT.

drives in which the angle between connected shaft can be kept small, the limit of angularity generally being placed at 7°

Rubber-Insulated Joint—A universal joint possessing some of the qualities of the disc type is the rubber-insulated type illustrated in Fig. 16. It originated in England, where it is known as the Lay-Rub, and is being manufactured in the United States by Thompson Products, Inc. This joint consists of two spiders *D*, two sheet-metal stampings *B* with four cup-shaped depressions in each, rubber spools *A* on metal cores *C* lodged in these depressions, and bolts and nuts by which the spools are assembled to the spiders. This joint is claimed to operate satisfactory up to an angle of 15°, and it also obviates the need for a slip joint in propeller-shaft drives, as it has some axial elasticity. It cushions the drive and insulates the car body against noise and vibration transmitted through the wheels and axle. The propeller shaft is a force fit over the shank of the spider, and is arc-welded to it.

Amidship Bearing—In trucks with forward-mounted powerplant and long wheelbase it is necessary to divide the propeller shaft, in order to avoid trouble from whirling. The original practice was to divide the shaft into three sections, a short intermediate section being carried in bearings mounted on a frame cross member and connected to the other sections by universal joints. It was later found, however, that a universal joint at the rear end of the forward section is not necessary, provided proper provisions are made in the design of the bearing and its mounting to allow for any slight lack of alignment. Misalignment caused some trouble with these bearings in the early designs, and was found to be due more frequently to inaccurate installation of the powerplant than to frame distortion. It can be allowed for by using a ball bearing of the self-aligning type, as shown in Fig. 17. How-

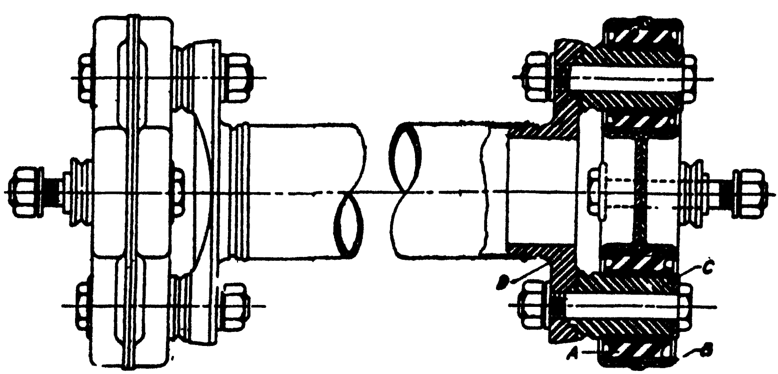


FIG. 16—RUBBER-INSULATED UNIVERSAL JOINT.

ever, a special single-row ball bearing also is available for this purpose, which permits of misalignment of the shafts up to 1.5° in all directions. Any variation in the axial dimension is allowed for by making the outer race of the ball bearing a sliding fit in its housing. In the design shown in Fig. 17, misalignment is further guarded against by providing a ball-and-socket support for the bearing housing on the frame member. One truck manufacturer takes care of both angular and axial inaccuracies by mounting the bearing housing in a block of rubber.

Careful provision must be made to keep dirt out of the bearing, because of its exposure to splashing mud. If felt seals are used, the felt rings should be quite large and flexible, and the caps should be fairly loose, and bolted down only after the housing is in place, so they can center themselves on the shaft in accordance with the tilt of the housing. Spring-loaded unit seals of either the leather or composition type are much used for this bearing. They can be pressed directly into the bore of the housing, and do not call for special machining. Such seals always should be mounted with the tips outward, so that if an excessive amount of grease is forced into the bearing by a service man, it can escape and will not cause undue heating. In addition to the oil seals, external mud slingers of the pressed-steel type are fitted, which ward off much of the splashing mud directed toward the bearing.

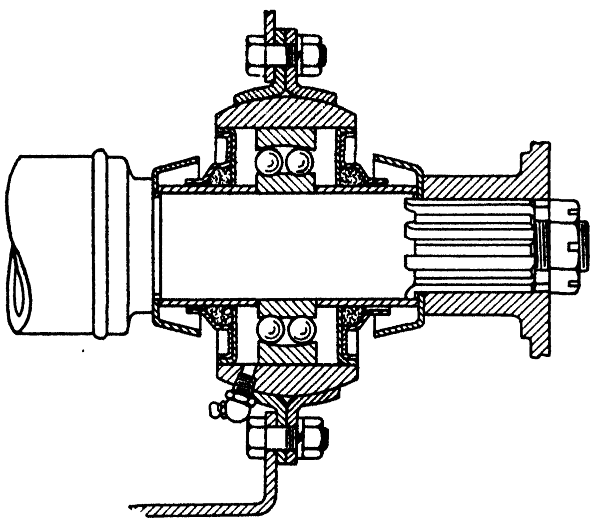


FIG. 17.—AMIDSHIP BEARING FOR PROPELLER-SHAFT.

Lubrication—Universal joints enclosed in a spherical housing immediately back of the gearcase can be lubricated from the latter by merely drilling holes in the top and bottom of the wall separating the two compartments. Other joints are provided with oil reservoirs in the cross or the trunnions. The trunnion bearings now are always closed at their outer end, the thimble-shaped bushings being held in place by either a snap ring or a bolted-on lock plate. At the inner end of the bearing the lubricant is retained by an oil seal, which is usually made of cork. Some universal joints have sufficient oil capacity so that the original supply is good for the life of the vehicle to which they are fitted, while in others the supply must be replenished after from 20,000 to 30,000 miles of service. A lubrication fitting is provided for this purpose at the center of the joint. For the lubrication of needle roller bearings a transmission oil of the 140 viscosity grade is recommended, but grease is preferable for the lubrication of slip joints. Slip joints always have large lubricant reservoirs, which are closed at one end by a disc or cap fitted to the yoke of the universal, and at the other end by a packed oil seal.

CHAPTER VII

Differential Gears

The propelling force required by a motor vehicle reaches maximum values during periods of rapid acceleration and while climbing steep grades, and is then substantially proportional to the gross weight. The maximum propelling force which the vehicle can develop in low gear usually is limited by the weight carried on the driving wheels, and since in a conventional vehicle the total weight is divided between four wheels, such vehicles must be driven by at least two wheels in order to have sufficient traction under all conditions. The driving force originates in a single engine, and when two wheels are to be driven from a single source, difficulty arises from the fact that when corners are being turned, all of the wheels follow different tracks, hence they must cover unequal distances in a given time, and therefore must rotate at different speeds. This they evidently could not do if they were rigidly fastened to a solid driving axle; one wheel or the other, or both, would then have to slip, which would be destructive to tires, and also would make steering practically impossible.

Historical—To make it possible to drive through wheels on opposite sides of the vehicle, and still permit of their turning independently of each other, use is made of a device variously known as a differential gear, an equalizing gear, or a "jack in the box." The type of differential gear in common use today, composed of an assembly of bevel gears and pinions, was invented in 1827 by Onésiphore Pecqueur, master mechanic of workshops connected with the Conservatoire des Arts et Métiers (Museum of Arts and Trades) in Paris. He is said to have conceived the idea while making a model of the Cugnot steam vehicle—the first motor-propelled road vehicle ever built. Cugnot's vehicle was both steered and driven by its single front wheel. The advantage of driving through two wheels must have been obvious, especially since at that time steel tires had to be depended on for traction, because with wheels spaced to give the normal tread of road vehicles and rigidly secured to a common axle it would have been quite impossible to make a sharp turn.

Differential gears first came into practical use on steam traction engines. The British firm of Clayton & Shuttleworth, for instance, used them for that purpose as early as 1865. Previous to that time it had been the practice to remove a driving pin from one of the road wheels whenever a fairly sharp turn had to be made. Differential gears therefore were well known when practical work on road vehicles equipped with combustion engines was started, toward the end of the nineteenth century.

The conventional differential gear has the characteristic that it applies equal torques to both of the road wheels driven through it under all conditions. This is an advantage under normal operating conditions, but may become a disadvantage when the vehicle has to proceed over slippery (muddy or icy)

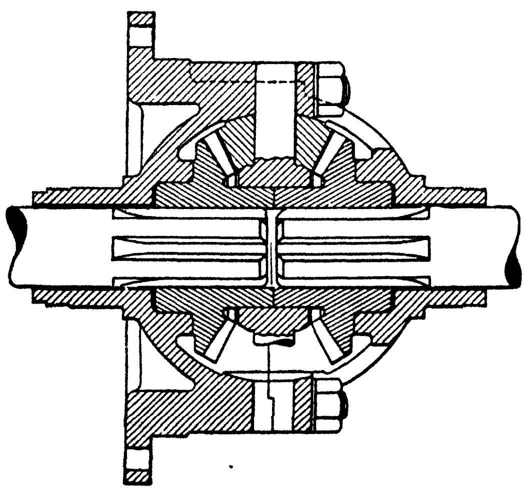


FIG. 1.—LONGITUDINAL SECTION THROUGH BEVEL-TYPE DIFFERENTIAL GEAR.

road surfaces. Various other types of differential gear have been invented, some to facilitate production, others to overcome the disadvantage of the conventional type just referred to, but their use has remained rather limited.

A bevel-type differential gear (Figs. 1 and 2) consists of two bevel gears called side gears, arranged coaxially and facing each other, and a number of bevel pinions between them and meshing with both. Generally either two or four pinions are used, though three-pinion differentials also are met with. The pinions, which are always equally spaced angularly, are capable of rotating on radial pins which at their outer ends are secured in a housing, and at their inner ends may be connected by an integral ring so as to form a spider. The housing is provided with hubs carried in ball or roller bearings in a differential carrier flange-bolted to the rear axle housing, and with a flange to which the driven bevel gear, worm gear, etc., can be secured.

Action of the Differential—Power is thus applied to the housing of the differential. The housing transmits it to the bevel pinions, the latter to the side gears, and these to the axle shafts. Under any given conditions of operation, a certain torque is impressed upon the differential housing. This torque is divided equally between the two, three, or four bevel pinions. Each bevel pinion constitutes a balance lever between the two bevel gears, and evenly divides its torque between them. Thus the total torque impressed upon the differential housing is at all times equally divided between the two side gears.

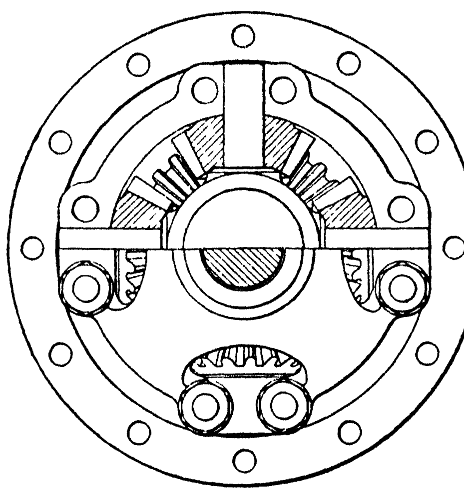


Fig. 2.—Half-Sectioned View of Bevel-Type Differential Gear.

The relative motion of the two side gears depends upon the position of the steering gear and upon the traction conditions. Suppose, first, that both driving wheels run on dry road surface so there is plenty of road adherence. Then the rate of revolution of each wheel and that of the corresponding side gear of the differential will depend upon the length of the path followed by that wheel. If the steering wheels are in the straight-ahead position, and both driving wheels have exactly the same diameter, then both will rotate at the same speed, as will the differential side gears. On the other hand, if the steering wheels are deflected from the straight-ahead position, the vehicle is constrained to travel in a curve, and the wheels on the outside of the curve will be forced to turn faster than those on the inside. Under these conditions the pinions of the differential will turn on their pins, allowing one side gear to run faster than the other. The angular speed of the differential housing is always equal to the algebraic mean of the speeds of the two side gears.

In case one of the wheels gets onto slippery ground and has insufficient road adherence, it will slip. The differential gear under these conditions also divides the propelling effort equally between the two driving wheels, and the wheel on dry surface can exert no more propelling effort than the one on slippery surface. The car will thus be stalled, and the wheel on slippery ground will be spun around at twice the rate at which it would otherwise turn with the engine running at the same speed, while the other wheel will remain stationary. This quality may be regarded as a defect of the differential gear, especially in the case of very heavy vehicles, and such vehicles sometimes are provided with a differential lock, consisting of some means for connecting the two side gears together so that they must rotate in unison.

Strength Requirements—The gears and pinions of differentials are generally made of a case-hardening alloy steel, such as the nickel steel 2320, the chrome-nickel steel 3120, the molybdenum-chromium steel 4120, or the molybdenum-nickel steels 4615 and 4815. When made of any of these steels, they combine great surface hardness with good tensile properties of the core. The two weakest elements in a bevel differential are generally the pin surfaces on which the pinions bear, and the teeth of the pinions. Calculation of these elements should be based upon the maximum rear-axle torque with the engine driving through the low gear of the transmission. Let the maximum rear axle torque (maximum engine torque $\times$ transmission low-gear ratio $\times$ rear axle reduction ratio) be designated by T; let the mean distance of the pin bearing surfaces from the axis of the differential be r, and let there be n pinions of a bore d and length of bearing l. Then the load on each pinion bearing is

$$P = \frac{T \times 12}{r \times n} \text{ lb},$$

and the unit load on the pin bearings,

$$p = \frac{T \times 12}{rndl} \text{ psi}.$$

In American practice the specific load on the differential pins is usually made about 10,000 psi. In some high-powered passenger cars in which the rear axle is designed for lubrication with E.P. (extreme-pressure) lubricants, the pin loading is carried as high as 13,500 psi.

Proportions of Gears—Since the strength of gears increases with the circular pitch, and smooth tooth action is not essential, on account of the low speed or lack of speed of these gears, a coarse-pitch tooth is generally used. In passenger-car work the pinions are made with from 10 to 12 teeth, and the gears with from 16 to 22. Passenger-car differentials are now generally made with two pinions, while differentials for heavy trucks are made with four pinions, which probably is the reason larger numbers of teeth are used in the gears of truck differentials. Thus the differential of a certain 5-ton truck has 15 and 30 teeth in the pinions and side gears, respectively, and that of a moderate-sized bus, 12 and 26 teeth. Side gears of three-pinion differentials must have a number of teeth divisible by three, and those of four-pinion differentials, a number of teeth divisible by two, otherwise they cannot be assembled.

Standard American stub teeth with 20° pressure angle are commonly used for differentials. The torque capacity of a two-pinion differential with gears made of any of the case-hardening steels mentioned in the foregoing is given approximately by the equation

$$T = \frac{10{,}000y}{P^3} N\sqrt{(N^2 + n^2)} \text{ lb-ft},$$

where y is the Lewis tooth-form factor for the number of teeth in the pinion; P, the diametral pitch; N, the number of teeth in each side gear, and n, the number of teeth in each pinion. T represents the maximum torque on the rear axle, as defined in the foregoing. The permissible size of the differential is limited by considerations of ground clearance under the rear axle. For this reason the device can be made somewhat more liberal in size in small, low-powered cars, but must be made as compact as possible in higher-powered ones.

Theoretically a four-pinion differential should have exactly twice the torque capacity of a two-pinion type, but since one cannot be sure that in a four-pinion differential the load will be absolutely equally divided between the four pinions, and also because four-pinion differentials are used chiefly in trucks, which run more in the lower gears than do passenger cars, it is well to make a slight allowance for this, and the torque capacity of a four-pinion differential is approximately

$$T = \frac{18{,}000y}{P^3} N\sqrt{N^2 + n^2} \text{ lb-ft.}$$

For a first approximation in laying out a differential gear the pitch diameter of the side gears of a two-pinion differential can be made $0.29 \sqrt[3]{T}$, and that of the side gears of a four-pinion differential, $0.24 \sqrt[3]{T}$ in.

Center of Load Distribution—Bevel gears produce both radial loads and thrust loads on the bearings supporting them. Before these bearing loads can be calculated, it is necessary to determine the center of load distribution along the face of the gear. We will assume that the tooth load is distributed along the tooth in proportion to the strength of the tooth section, which is an ideal condition. In practice, of course,

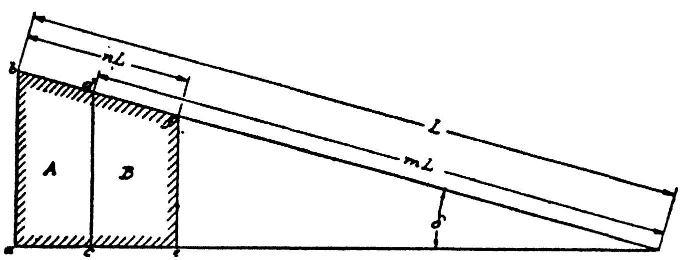


FIG. 3.—CENTER-OF-LOAD-DISTRIBUTION DIAGRAM.

much depends upon the accuracy of the gears and upon their mounting and adjustment. Let *n* be the ratio of the tooth face to the cone distance, and *m*, the proportion of the cone distance represented by the distance from the cone center to the center of the tooth load. From the well-known formula for the strength of gears we know that the strength of a tooth section is directly proportional to the circular pitch of the section, and the pitch, of course, decreases uniformly from the large end of the tooth to the cone center, where it is zero. The strength of the section of a bevel gear is also proportional to the width of that section. In Fig. 3 the vertical lines *ab*, *cd* and *ef* represent the circular pitch at the respective points, and the line *cd* is supposed to divide the entire gear into two parts of equal strength. As the strength of a gear is proportional to the product of its circular pitch into its width of face, the two areas *A* and *B* should be equal. If this is the case, then

$$\left[\frac{L \sin \delta + L(1 - n) \sin \delta}{2}\right] nL \cos \delta$$

$$= \left[\frac{L \sin \delta + mL \sin \delta}{2}\right] (L - mL) \cos \delta \times 2$$

$$\frac{L + L(1 - n)}{2} nL = (L + mL)(L - mL)$$

$$\frac{2L - nL}{2} nL = L^2 - m^2L^2$$

$$nL^2 - \frac{n^2L^2}{2} = L^2 - m^2L^2$$

$$n - \frac{n^2}{2} = 1 - m^2$$

$$m = \sqrt{1 + \frac{n^2}{2}} - n.$$

For convenience in making calculations, the values of m for a number of values of n within the usual range are tabulated below.

n..........	0.26	0.28	0.30	0.32	0.34	0.36	0.38
m..........	0.879	0.871	0.863	0.855	0.847	0.840	0.832

Bevel Gear Bearing Loads—We are now in position to calculate the radial load and the thrust load on the supporting bearings due to the tooth pressure of bevel gears. In Fig. 4, AB represents the normal pressure on the gear tooth at the pitch line. We first resolve this into a component AC in a plane tangential to the pitch cone and a component CB in a direction perpendicular to this plane. The tooth pressure is regarded as a concentrated force acting at the center of load distribution, and both components pass through this point. AD represents the component perpendicular to the plane tangent to the pitch cone, both in magnitude and direction.

$$AD = CB = AC \tan 20° = T \tan 20°,$$

20° being the pressure angle of the teeth. The load represented by AD may be resolved again into a component DE, parallel with the pinion axis (the thrust load on the pinion), and a component AE, in a plane perpendicular to the pinion axis.

$$DE = AD \sin \gamma = T \tan 20° \sin \gamma$$

$$AE = AD \cos \gamma = T \tan 20° \cos \gamma,$$

γ being the pitch angle (designated by θ in Fig. 4).

There are evidently two components of the tooth pressure in the plane perpendicular to the pinion axis, AC and AE, and their resultant, AF, constitutes the radial load on the pinion bearing.

$$AF = \sqrt{AC^2 + AE^2} = \sqrt{T^2 + (T \tan 20° \cos \gamma)^2}.$$

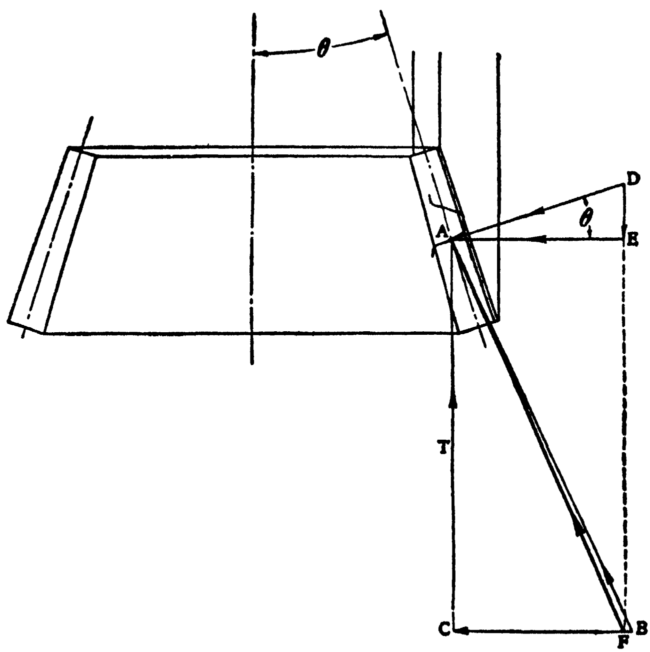


Fig. 4.—Diagram of Components of Bevel-Gear Tooth Pressure.

The same formulae are applicable to the gear, but the results are different, since the pitch angle γ, which occurs in each of them, has a different value. In the case of the differential gear it is not necessary to calculate the radial load on the pinion bearing (the pins) in this way, because these can be more simply calculated with sufficient accuracy from the rear axle torque, the mean distance of the pin bearing surfaces from the axis of rotation, and the number of pins, as already explained. In calculating the thrust load on each pinion, account must be taken of the fact that teeth on opposite sides are under pressure simultaneously. With a number of pinions equally spaced around the circumference there is no radial load on the side gears, and these therefore do not need large supporting surfaces.

Sample Calculation—In the following the pinions and gears of a two-pinion differential for a medium-sized passenger car will be calculated. We will assume the pinions to have 12 and the gears 22 teeth each. Then, if the diametral pitch is 6 and the teeth are of standard stub form, the differential will take care of a maximum torque of

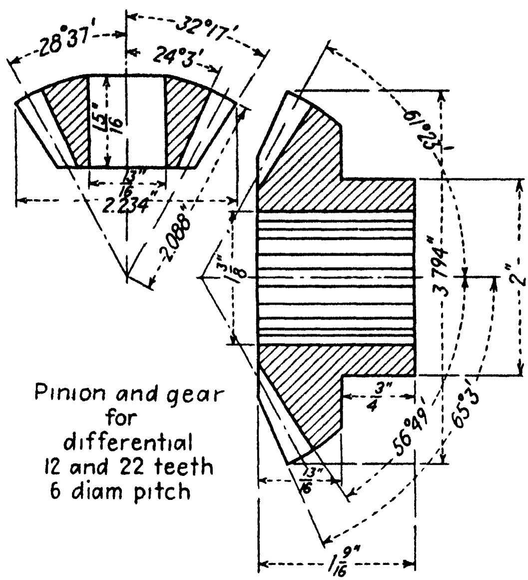


FIG. 5.—PINION AND SIDE GEAR OF CALCULATED DIFFERENTIAL.

$$\frac{10{,}000 \times 0.078}{6^3} \times 22 \times \sqrt{(22^2 + 12^2)} = 1990 \text{ lb-ft.}$$

As cars of this type usually have an overall reduction of about 11:1 in low gear, this corresponds to a maximum engine torque of 180 lb-ft.

The maximum pitch diameter of the pinion is 2 in., and that of the side gear, 3.666 in. The cone distance therefore is

$$\sqrt{\left(\frac{3.666}{2}\right)^2 + \left(\frac{2}{2}\right)^2} = 2.088 \text{ in.}$$

The face width of the gears can be made slightly more than one-third the cone distance—say, ¾ in. As the pitch radius divided by the cone distance is equal to the sine of the pitch angle, we have $\sin \gamma = 1/2.088 = 0.479$; therefore, $\gamma = 20° \, 37'$. The pitch angle of the side gear is equal to the complement of this, or 61° 23′ (Fig. 5).

The addendum for the gears is $0.8/6 = 0.1333$ in., and the dedendum $1/6 = 0.1666$ in. The addendum divided by the cone distance is equal to the tangent of the angle which must be added to the pitch angle to give the face angle, and the dedendum divided by the cone distance is the tangent of the angle which must be subtracted from the pitch angle to give the root angle:

$$\frac{0.1333}{2.088} = 0.0639 = \tan \text{ of } 3° \, 40'$$

$$\frac{0.1666}{2.088} = 0.0799 = \tan \text{ of } 4° \, 34'.$$

Consequently, the face angle is

$$28° \, 37' + 3° \, 40' = 32° \, 17' \text{ for the pinion and}$$

$$61° \, 23' + 3° \, 40' = 65° \, 3' \text{ for the gear,}$$

and the root angle is

$$28° \, 37' - 4° \, 34' = 24° \, 3' \text{ for the pinion, and}$$

$$61° \, 23' - 4° \, 34' = 56° \, 49' \text{ for the gear.}$$

Since the width of face is ¾ in. and the cone distance 2.088 in., the ratio n of face width to cone distance is 0.36. This makes the ratio of the distance between the center of tooth load and the axis of the differential to the pitch radius of the side gear

$$m = \sqrt{1 + \frac{0.36^2}{2}} - 0.36 = 0.839,$$

and the actual distance of the center of tooth load from the axis is

$$0.839 \times 1.833 = 1.536 \text{ in.}$$

Thus the total tooth load under maximum rear-axle torque (1990 lb-ft) is

$$\frac{1990 \times 12}{1.536} = 15{,}550 \text{ lb.}$$

As this is divided between two pinions and two side gears, the total tooth load on each pinion and each side gear is 7775 lb. In each case this load is divided between at least two different teeth, on opposite sides of the pinion and gear.

We can now make use of the equations for thrust load on bevel gears developed in the foregoing. The radial thrust on each pinion is

$$7775 \times \tan 20° \times \sin 28° 37'$$
$$= 7775 \times 0.364 \times 0.479 = 1350 \text{ lb},$$

and that on each side gear,

$$7775 \times \tan 20° \times \sin 61° 23'$$
$$= 7775 \times 0.364 \times 0.878 = 2480 \text{ lb}.$$

In normal operation, of course, these thrust loads are much smaller. Thus, a car having a gross weight of 4000 lb, a forwardly-projected area of 28 sq ft, and an effective wheel radius of 14 in., at 45 mph on smooth level road in still air requires a driving force of only about 160 lb, which corresponds to a rear-axle torque of 187 lb-ft, or less than one-tenth that of the 1990 lb-ft on which the foregoing calculations are based. The thrust loads, of course, are smaller in direct proportion.

Formerly it was common practice, in the passenger-car industry at least, to have the pinions and side gears of the differential bear directly on the housing, which is usually made of cast iron. Quite naturally, wear takes place on the thrust surfaces, and as a result slack developed in the drive, which was quite objectionable because of the shocks received by the driving members when the slack was taken up as the clutch was engaged, or when the brakes were applied. At present, thrust washers are placed back of both pinions and gears, which, when they are worn, can be easily replaced. That back of the pinion has the form of a spherical segment. These thrust washers are often made of plastic phenolic material. To ensure proper lubrication of the bearing surfaces, it is customary to drill an oil hole through each pinion and side gear, at the bottom of a tooth space. Alternately, oil grooves may be cut in the pins of the differential.

The pins in the differential to which the foregoing calculations apply can be made of $1\frac{3}{16}$ in. diameter, and the pinion bearings on them will be $1\frac{5}{16}$ in. long. To make these bearings as long as possible, the pinions usually are cut off flat from the small ends of the teeth, while the outer end is made spherical. The mean distance of the pin bearing from the axis of the differential is 1.60 in., hence the maximum load on each bearing will be

$$\frac{1990 \times 12}{2 + 1.60} = 7500 \text{ lb},$$

and as the bearing area on each pin is 0.763 sq in., the unit load under the worst conditions will be slightly less than 10,000 psi. While there is quite a heavy load on the pinion bearings (the pins), with a number of pinions equally spaced angularly, there is no radial load on the side gears, and these, therefore, present no special bearing problem. Hubs formed on the side gears usually are carried directly in bores in the housing.

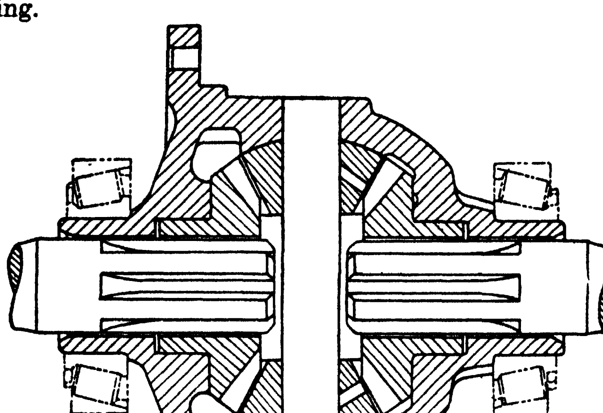


FIG. 6.—SINGLE-PIECE-HOUSING DIFFERENTIAL.

Single-Piece-Housing Differential—The simplest design of 'differential gear is one having only two bevel pinions and a single-piece housing, of which an illustration is shown in Fig. 6. The inside of the housing is finished to a spherical surface having its center at the cone center of the gears. There are large openings in the sides of the housing through which the side gears can be inserted. The two pinions are then meshed with the side gears, and the whole gear assembly is turned around the common axis of the side gears until the common axis of the pinions coincides with the axis of the holes for the pinion pin in the housing, whereupon the pin is introduced. The pin carrying the pinions is locked in position by a pin screw. In most of the newer designs a tubular spacer is placed over the pin between the two pinions to hold them the proper distance apart. This type of differential is now in almost universal use on passenger cars.

Details of Construction—Differential housings are generally cast of malleable iron, though the split type also can be drop-forged. The housing—or one-half of it—is formed with a flange to which the bevel crown gear is secured. This flange is offset from the center plane of the differential so as to bring the axis of the driving pinion into the center plane of the differential. In the case of a final drive by worm or spur gear, a flange may be provided on each half of the housing, and the worm or ring gear clamped between them. The hubs of the differential housing are turned to form seats for antifriction bearings supporting the differential in the differential carrier. While these hubs usually carry the inner race of the axle bearings, in some designs the outer races are fitted into hubs on the differential housing and the inner races are mounted on the ends of tubes extending through the axle housing. The side gears are broached out to take the splined ends of the axle shafts, which latter clear the bore of the hubs of the differential housing.

Spur-Gear Differentials—A differential gear can be built also with spur-type side gears and pinions. Such differentials were used to a certain extent at one time, but are now only rarely met with, probably because for a given torque capacity they require a housing of larger diameter, thus tending to cut down the road clearance under the axle. As shown in Figs. 7 and 8, a spur differential consists of two spur gears mounted on the inner ends of the differential shafts, of a varying number of pairs of spur pinions, and of a housing surrounding the whole. The spur pinions are of substantially twice the width of the spur gears; the latter are placed some distance apart, and the extra width of the pinions extends into this intermediate space where the two pinions of each pair mesh together. The action of this type of differential is exactly the same as that of a bevel differential.

The pinions of spur-gear differentials are made with a small number of teeth, generally about ten, because any small increase in their size entails a large increase in the diameter of the differential housing. Stub teeth are preferably used. The cases of spur differentials are made in two parts which are held together by bolts. The parts preferably should be provided with a telescoping joint, to ensure continued alignment of bearings. One part is usually made in the form of a circular plate and the other in the form of a cylinder open at one end, and the differential drive gear may be fastened to the housing by the same bolts which hold the two parts together.

Non-Slip Differentials—A large number of differentials have been invented which prevent a car from losing traction when one wheel gets onto slippery ground. Most of them involve some form of one-way transmission device, that is, a mechanism by which power can be transmitted in one direction but not in the other. With the ordinary differential, if one wheel is held from rotating and the frame or housing of

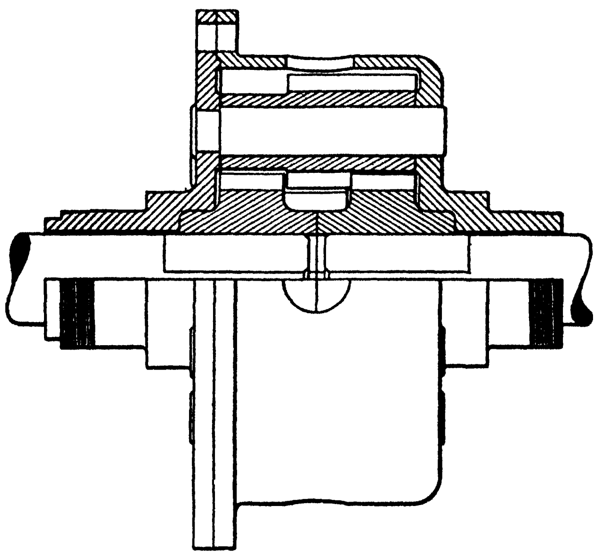


FIG. 7.—SPUR-PINION DIFFERENTIAL, SIDE VIEW AND LONGITUDINAL HALF-SECTION.

the differential is rotated, the other wheel will be rotated at twice the speed of the housing. Also, if the housing is held from rotation and one wheel is rotated, the other wheel will rotate in the opposite direction at the same speed. With one of the special differentials of the type referred to, if one wheel is locked or held from rotating, by turning the other wheel, the housing may be rotated, but it is impossible to turn the free road wheel by turning on the housing.

It is self-evident that such a differential does not equally divide the torque between the two driving wheels, for if it did, when one wheel was spinning, the other wheel would have no more torque impressed upon it than the spinning one, which would be insufficient to propel the car. The relative torques impressed upon the two wheels respectively depend upon the resistance encountered by them. Ordinarily in straight-ahead motion, both wheels encounter substantially equal resistances, and the driving torque on both is therefore the same. But in turning a corner the outer wheel is compelled to run ahead of the differential housing, and all the torque is taken by the inner wheel, the conditions then being the same as when one wheel has no traction.

One of the first differentials of the non-slipping type to appear on the market was the M. & S., illustrated in Fig. 9.

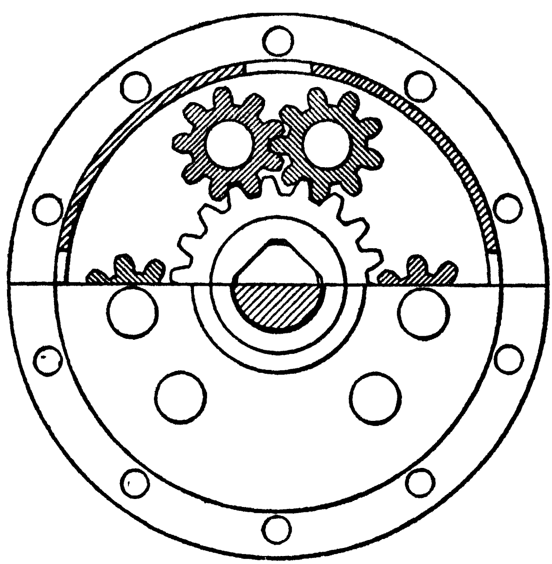


FIG. 8.—SPUR-PINION DIFFERENTIAL, END VIEW AND TRANSVERSE HALF-SECTION.

Each of the axle shafts carries a helical side gear, and the differential housing carries three helical pinions with radial axes, and six such pinions of which each one meshes with both one of the radial pinions and one of the gears on the axle shafts. It is well known that in helical gears, if the helix angle of the driving gear is very small, power cannot be transmitted through the pair in the reverse direction, because the frictional resistance is too great, and this is the principle made use of in this differential.

Multi-Pull Differential—Another, more recent type of self-locking or non-slipping differential is the Multi-Pull, illustrated in Fig. 10. Between the halves *A* and *B* of the housing is located the center drive ring *C*. Inside this drive ring there are two center rings *F*, each of which contains eight steel balls. Adapter fittings *E*, splined to the inner ends of the axle shafts, carry the side plates *D*. Each half of the housing is machined with twelve holes on the inside, into which are inserted coil springs and steel balls which tend to hold the side plates *D* in engagement with the drive ring *C*. While the vehicle is on a straight course, both side plates *D* are clutched to the drive ring *C* by the spring pressure. When the front wheels are swûng around in steering, the side plate

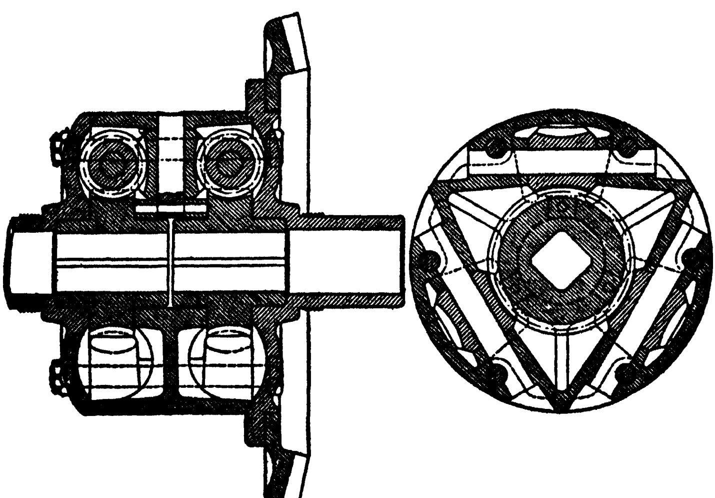


Fig. 9.—M. & S. Non-Slip Differential Gear.

to which the outer road wheel is secured moves out of engagement with drive ring *C* and slides along the splines on the outside of fitting *E*. In the illustration the left side plate is shown disengaged. Farther movement of the outside wheel causes its side plate to engage with the drive ring again, due to the pressure of the twelve springs and balls. The springs exert a pressure of 3 lb each, and have a range of motion of $\frac{5}{16}$ in.

It will thus be seen that when the vehicle describes a turn, the outer wheel is disengaged from the driving mechanism, so that the drive must be through the inner wheel alone. As in turning a curve the inner rear wheel travels a shorter distance than any of the others, the propelling force it must develop is comparatively large, and under conditions of poor adhesion it may not be able to develop sufficient traction. Propulsion on curves by the inner wheel alone is a characteristic of all differentials which do not lose traction when one of the driving wheels gets onto slippery ground. Use of this type of self-locking differential is confined largely to heavy trucks which occasionally must be operated off the hard-surfaced roads. Locking or non-stalling differentials are required most in all-wheel drive trucks, with which otherwise all traction may be lost if any one of the four or six driving wheels hits a slippery spot of pavement.

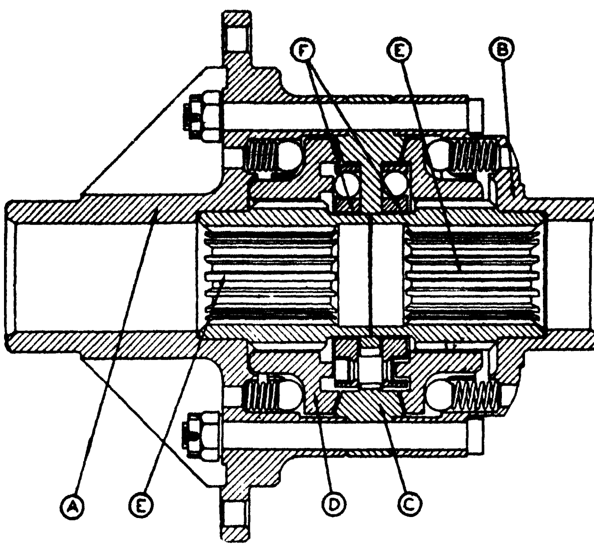


FIG. 10.—MULTI-PULL NON-SLIP DIFFERENTIAL GEAR.

Differentials with Torque Bias—In addition to the two types of differential described in the foregoing, there is a third, intermediate type which, while not entirely preventing the spinning of a wheel on slippery surface, will permit greater tractive force to be developed by the other wheel than by the one thus handicapped. With such a differential the maximum tractive force which can be developed by the wheel with plenty of road adhesion is proportional to that which can be developed by the one with little adhesion. Such differentials are particularly suitable for inter-axle installation in vehicles with two driving axles. To show the advantage of such a differential, let us assume that it can impress a maximum torque on the axle with the greater road adherence which is 1.5 times as great as that which the other axle will take, and that the operating conditions are such that the total tractive force required by the vehicle is equal to 10 per cent of the load on the driving wheels. Assuming all driving wheels to carry equal loads, with all three differentials of the conventional type the vehicle will stall as soon as the coefficient of road adhesion of one wheel drops below 0.10. On the other hand, with the special inter-axle differential the vehicle will not stall until the coefficient of road adhesion of one wheel drops below 0.075.

There are at least two principles which can be made use of in such torque-biasing differentials. One is exemplified in a design in which the differential bevel pinions, while acting as equal-armed levers under ordinary. conditions, become unequal-armed levers in the position they assume when one of the wheels tends to slip. The other is similar to that of the self-locking differential, in that the design embodies mechanism in which there is much greater friction when a force or torque is applied to it at one end than when it is applied at the other. In the self-locking differential the friction is so great that motion cannot be transmitted through the mechanism in one direction; in the torque-bias differential it is considerably less.

Timken High-Traction Differential—A torque-biasing differential is being manufactured by the Timken-Detroit Axle Company. In design it is similar to the conventional bevel-gear differential, but its pinions and side gears are cut with a relatively small number of teeth of special form. With ordinary involute teeth, the effective lever arm on which the tooth pressure acts is constant in length, whether there is contact on, above or below the pitch line, and the torque therefore is always equally divided between the two side gears. With the special teeth used in the Timken High-Traction differential, the length of the lever arm changes with the point of contact on the teeth. The principle of action may be explained by reference to the two views of Fig. 11. There lines are drawn normal to the tooth contours at the instantaneous points of contact, and perpendiculars are drawn from these lines to the axis of the pinion and to imaginary axes of the side gears. It is apparent that the lengths of lever arms L_1, L_2, L_3, and L_4 vary with the point of contact. Thus in the upper view the leverage is 1:1, while in the lower view it is 1:1.63. The ratio in which the torque is divided changes uniformly as the gears move from one extreme position to the other.

When the vehicle is being driven on a straight course, the gears assume the position in which they are shown in the upper view, because both wheels then encounter equal resistances, and if one acted on a longer lever arm, they automatically would turn until the lever arms were equal. But when—for one reason or another—one of the road wheels is unable to take one-half of the torque impressed, it will run ahead and move the gears into the position where this wheel

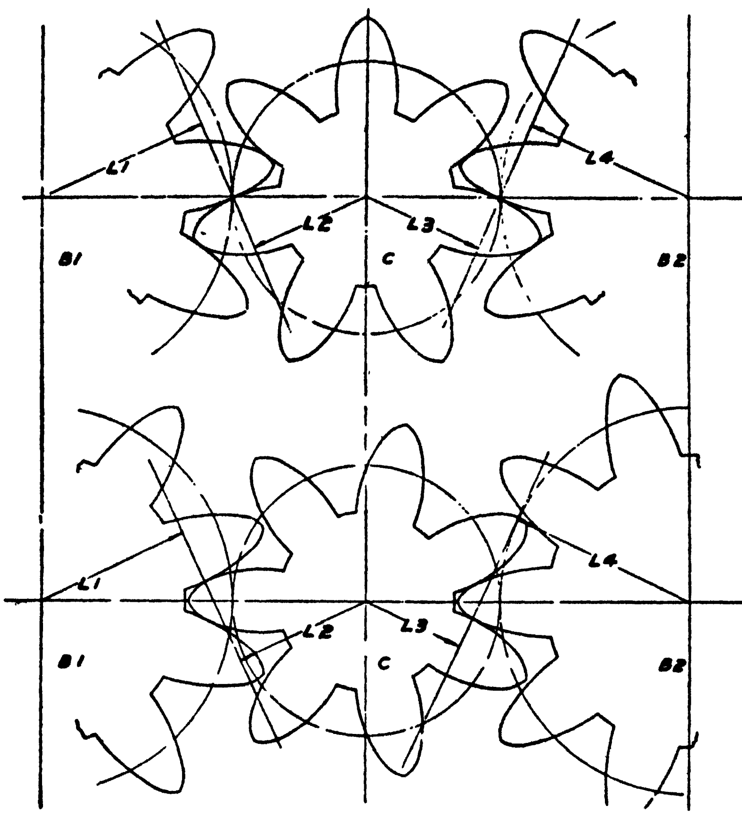


FIG. 11.—ILLUSTRATING PRINCIPLE OF HIGH-TRACTION DIFFERENTIAL.

receives less than one-half of the total torque, whereby the tendency to slip is reduced. It is claimed that with this differential 24 per cent more torque can be delivered to the wheel with better traction. Differentials of this type have been used on racing cars to reduce wheel slip and consequent tire wear.

Cam-Type Differential—Another differential of the torque-biasing type makes use of concentric cams and intermediate plungers in contact with them. As shown in Fig. 12, the driving member is a cylindrical shell or cage *A*, with slots in it which accommodate radially reciprocating plungers *B,B*. There are two rows of such plungers, but they are offset angularly, which is the reason plungers of one row only show in the longitudinal section. The two pairs of cams also are offset angularly with relation to each other. Plungers *B,B* contact both the inner cam *C* and the outer cam *D*, which are the driven members, corresponding to the side gears of a conventional differential. In Fig. 12 this differential is shown as designed for inter-axle use, the two driven members connecting two concentric shafts. The double inner cam *C* is splined to a solid shaft *E*, which at its rear end carries the driving member (pinion or worm) for the rear driving axle, while the outer cam *D* is splined to the tubular shaft *F*, which

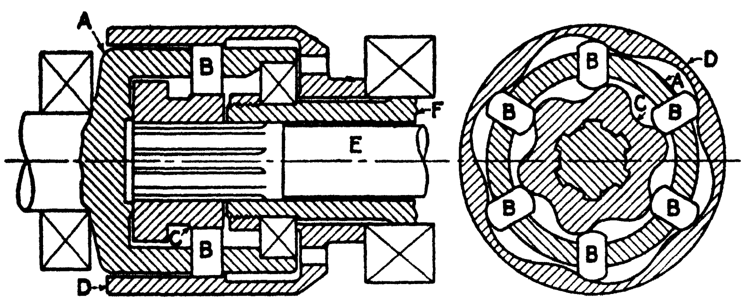


FIG. 12.—CAM-TYPE TORQUE-BIAS DIFFERENTIAL.

carries the driving member of the final-drive gear for the forward driving axle.

In operation, when there is no differential action, the plungers assume the positions relative to the cams in which they are shown in the right-hand view of Fig. 12. It might be thought that, since the point of contact on the outer cam is farther from the axis of rotation than that on the inner one, greater torque would be applied to the axle driven by the former. Such, however, is not the case. The plunger, of course, exerts a pressure on both cams, and if it is in equilibrium, the components of both contact pressures along its axis must be equal. The tangential force at each point of contact is equal to the product of the force along the plunger by the tangent of the angle of slope at the point of contact. The cam slope is considerably less at the point of contact on the outer cam, hence the tangential force on the outer cam also will be less, but since its radius or moment arm is considerably greater, the torque on the outer cam will be the same as that on the inner one.

With this type of differential, the same as with the conventional one, if one of the driven members (the cams) is held from rotation, and a torque is applied to the driving member (the cage), the other cam will turn at twice the speed of rotation of the cage. The plungers then will be carried over the humps of the cam which is being held, and will not only carry the other cam along with them, but cause it to rotate at twice its own speed, owing to the interaction between it and the reciprocating plungers. When the plungers are in contact with the crest of one cam and the bottom of the other, the tangential component of the pressure between cam and plungers is *nil,* and to prevent a dead-center effect, the cams and the rows of plungers are so staggered that when one row of plungers contacts its cams at top and bottom, the other row contacts them about halfway up the cam slopes.

Differential Lock—Some early motor trucks were provided with manually operated differential locks. With such a lock, if the truck lost traction because one wheel was on a muddy or an icy patch of road, the driver would lock his differential and then proceed, as this enabled him to apply all available torque to the other driving wheel, or at least as much as that would take, which was usually enough to get off the slippery surface. The problem of working out a neat and generally satisfactory design of differential lock is not an easy one, especially now that so much power is being transmitted to the driving wheels. Besides, with increase in the proportion of hard pavements and the provision of facilities for sanding icy parts of main roads in winter time, the need for a differential lock has decreased, and this probably explains why it is no longer used. Fig. 13 shows a differential lock which was used on some early motor trucks. This particular lock was operable from the driver's seat, while some others had to be operated from the ground.

Friction Lock—Instead of providing the differential with a jaw clutch by means of which one of the side gears (or one axle shaft) can be connected positively to the differential housing, it may be provided with a friction clutch between the two members. To facilitate installation, a metallic multiple-disc clutch would be chosen for the purpose, because its diameter can be made small. It would be kept permanently engaged by springs. With this arrangement it is possible for the wheel having good traction to take torque equal to the sum of the torques required to spin the wheel having poor traction and to slip the clutch. Since the wheel having good traction can take greater torque, it can produce more tractive effort. Such a friction-type differential lock was described in a 1951 S.A.E. paper by L. R. Buckendale and L. G. Boughner of the Timken-Detroit Axle Company, and was said to have been used in a fleet of pole-setting trucks operated by a telephone company. Unlike the positive differential lock, which would be engaged only to get out of a difficult position and unlocked immediately thereafter, the friction lock remains effective permanently, and whenever the vehicle is negotiating a turn, either the clutch or one of the wheels must slip, which entails

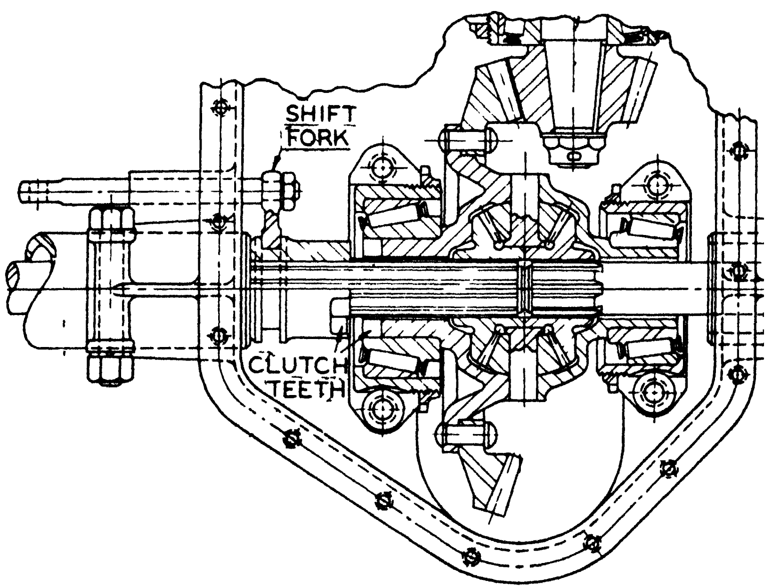


FIG. 13.—DIFFERENTIAL LOCK.

loss of power and wear and tear. Under normal traction conditions the slippage would occur in the clutch. This combination of a conventional differential with a friction clutch is merely another form of biased differential.

None of the special differentials described in the foregoing is of any use if both driving wheels have equally poor traction.

It will be seen that some of the illustrations in this chapter (Figs. 7, 8 and 9) show the inner ends of the axle shafts squared where they enter the differential side gears. This type of shaft joint was used during an earlier period, and was then considered a great improvement over the still earlier practice of using one or two keys. Actually it is a poor practice, because the square section of the shaft has little more than one-half the torsional strength of the full section. At present, shaft joints transmitting engine power are always splined, and generally the ends are upset so that the splines do not weaken the shaft at all.

Differentials for Unequal Division of Torque—In vehicles with four-wheel or six-wheel drive it is often desirable to divide the torque unequally between the driving axles. In a four-wheel-drive truck, for instance, if the rear wheels are equipped with dual tires, they are able to carry much more weight than the single-tired front wheels, and there is good reason for dividing the weight on a 1:2 basis. In such a case, if an ordinary differential were used between the front and rear axles, only ne-half of the load on the rear wheels would

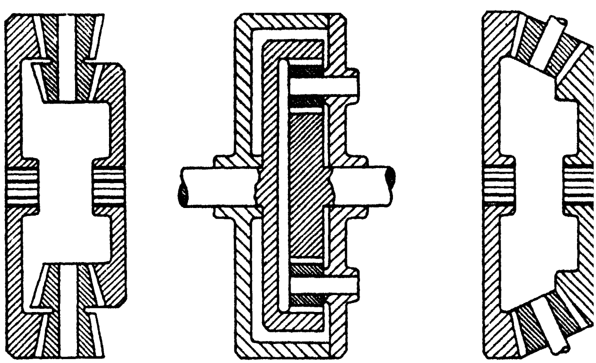


FIG. 14.—DIFFERENTIALS DIVIDING TORQUE UNEQUALLY.

be available for traction purposes, because no more torque could be applied to the rear than to the front wheels, that is, about one-half as much as the rear wheels would stand. Therefore, in a truck with two or more driving axles, the torque should be divided between these axles in the proportion of the loads supported by them when under maximum driving torque. Driving torque transfers load from the front to the rear axle, and the load division then is no longer the same as under static conditions. Diagrams of three types of differential for unequal torque division are shown in Fig. 14. The one on the left probably is not very practical, as with two pinions arranged in line and sufficiently far apart so they can be cut with conventional tooling, the diameter of the housing would have to be quite large. The design in the middle has advantages if the ratio of torques to the two axles is to be fairly large, while the one on the right can be designed for either small or large torque ratios, as the side gears can be made with tooth numbers differing by one, two, three, or any larger number.

One fact that must be kept in mind by the designer when contemplating the use of either a non-slip differential or a differential lock is that with these the whole of the engine power may under certain conditions have to be transmitted by one axle shaft and one road wheel, and these parts may have to be made stronger than would otherwise be necessary. Of course, the torque load on these parts is still limited by the maximum ground adhesion of the wheel.

CHAPTER VIII

Rear Axles

Rear axles are divided into "dead" and "live" axles. A "dead" axle is one which merely supports the chassis and has no part in the transmission of power to the rear wheels, while a "live" axle carries chassis weight and also transmits driving power. In motor vehicles, dead axles are used only in combination with side-chain drive to rear wheels, or with front-wheel drive, and are therefore quite rare.

Live axles usually comprise separate driving and load-carrying members, and are then referred to as "floating" axles. Non-floating axles, in which the same unit serves to carry load and transmit driving power, were used to a small extent during the pioneer days of the industry, when front and rear axles were connected by "reach rods" or by a triangular structure secured to the bearings of the rear axle and having a pivotal connection with the front axle at its center. As shown in Fig. 1, such an axle consists simply of a straight cylindrical bar extending through bearings secured to the chassis springs and the reach rods. One end is reduced in diameter and carries a differential gear, the side gears of which

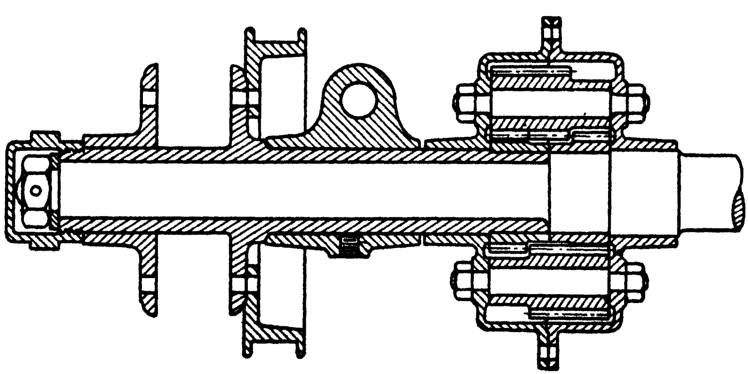


FIG. 1.—SECTION OF NON-FLOATING DRIVING AXLE (FRAGMENT).

are keyed or otherwise secured to the axle and to a sleeve on the reduced portion thereof, respectively. The road wheels also are secured to the axle and the sleeve. Non-floating axles were discarded because they did not permit of enclosure of the driving mechanism and of its effective lubrication.

Parts of Floating Axle—The part of the floating axle which supports the load is known as the housing, while the parts which transmit the power (and which may also assist in carrying the load) are known as the axle shafts. The axle housing comprises a central enlarged portion enclosing the driving gears and differential, and lateral extensions thereof which, if they are separate parts, are known as axle tubes. The driving-gear housing also has a forward extension, in which there are bearings for the drive-pinion shaft. To the outer ends of the axle tubes are secured the brake backing

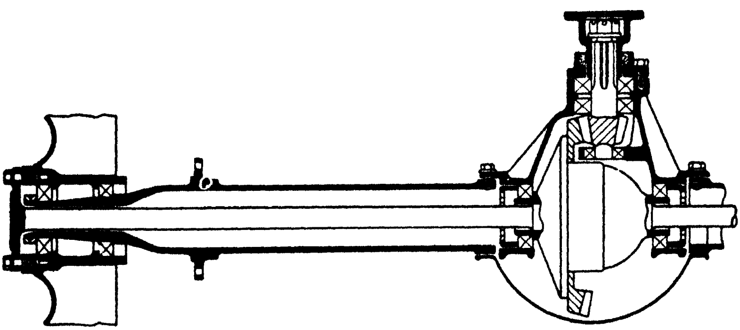


FIG. 2.—FULL-FLOATING AXLE.

plates, and between these and the center housing, spring seats may be either formed on the housing or secured thereto by welding or riveting.

Stresses in Axle Parts—A rear axle is subjected to both bending and torsional moments. The weight supported by it subjects it to bending moments and shear in the vertical direction, the axle acting as a simple beam supported at its ends and loaded at intermediate points. Driving and retarding (braking) forces on the wheel rims produce bending moments and shear in the horizontal fore-and-aft direction. Driving torque produces shear in the axle shafts and the axle housing; braking torque, in the housing alone, provided the brakes act directly on the wheels. In addition to these vertical and horizontal fore-and-aft forces, which are active as long as the vehicle is in motion (except for the braking forces), there are occasional transverse forces on the axle, as when

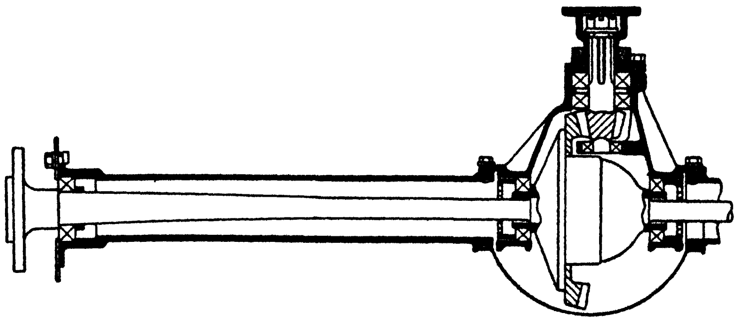


FIG. 3.—SEMI-FLOATING AXLE.

the vehicle turns a corner (centrifugal force), when the wheel hits a road obstruction while skidding, or when the vehicle is being backed into a curb at an angle.

Classification of Live Axles—A live axle in which the axle shafts are subjected to torsional stresses only is called a full-floating axle (Fig. 2). This involves that the axle housing extend through the road wheels, the latter having bearings on the outside of the housing. An axle in which the wheel is secured to the axle shaft and the latter has a bearing in the end of the axle housing, so that the chassis weight is transferred first from the axle housing to the axle shaft and then to the wheel, is known as a semi-floating axle (Fig. 3). With axles of this type the axle shafts, in addition to being subjected to torsion, are subjected to bending moments by the load on the axle and by forces on the wheel in the fore-and-aft as well as in the transverse direction. There is a third type of axle, known as the three-quarter floating type (Fig. 4), in which there is a single bearing on the outside of the axle

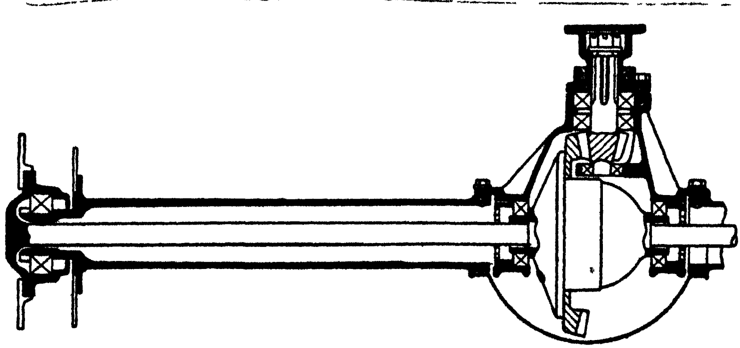


FIG. 4.—THREE-QUARTER-FLOATING AXLE.

housing in the center plane of the wheel. It is obvious that with this construction, forces in the plane of the wheel, whether vertical or horizontal, cannot produce any bending moment on the axle shafts, but transverse forces on the wheel can. In order that the axle shafts may be completely protected against bending stresses, it is necessary that the connection between the axle shaft and the wheel should not be rigid in any plane in which a bending moment might act. This would necessitate the use of a dog clutch between the shaft and the wheel hub. Heavy commercial vehicles now usually have rear axles in which each wheel is mounted on two antifriction bearings on the outside of the axle housing, and the axle shaft is forged with an integral flange at its end, by means of which it is bolted to the wheel hub. This construction has sometimes been referred to as a seven-eighths floating axle. The distinction, however, is of no great importance, and axles of this type are generally classed as full-floating. The differences between the three types are confined to the methods of mounting the wheels on the axle, and the center portion of the axle may be the same for all.

The differential gear is always supported in antifriction bearings in the center housing, and the axle shafts have splined connections with the side gears of the differential, in which they are a sliding fit. In built-up axles the center housing is a malleable-iron casting with a large opening in the back through which the driving pinion with its stub shaft and the differential gear can be introduced. This type of axle, however, is not much used any more. Conventional axles of modern road vehicles are of the "banjo" type, in which the entire axle housing, comprising both the center housing and the axle tubes, is in a single piece. The center housing, however, has large openings in both front and rear. The front opening is closed by what is known as the gear carrier or the differential carrier, while the rear opening is closed by a cover plate, which usually has the shape of a spherical segment.

There are three other types of rear axles that deserve mention, although they are rarely used in American practice. With one of these there is a final gear reduction directly at each wheel, by either a pair of spur gears or a spur pinion and an internal gear. In this case the axle shafts, that is, the shafts carrying the spur pinions, are eccentric with relation to the carrying member of the axle, and the latter is generally made in the form of an I-section drop forging. The center housing, containing the differential and a pair of bevel gears, may be secured to either the forward or rear side of the axle beam. If it is secured to the rear, the forwardly projecting part of the housing, containing the bearings for the pinion shaft, must extend through an opening in the web of the beam. The axle shafts are enclosed in tubes carried by the center housing at their inner, and by a gear housing rigidly secured to the carrying member at their outer, ends, hence the driving shafts always remain parallel with the carrying member.

De Dion-Type Axle—Another type, which dates back to the very infancy of the industry, is known as the De Dion axle (Fig. 5). It was first used in a very small passenger car, while more recently it has been used in some heavy buses. With this type of axle, the center housing enclosing the differential and driving gears is supported on the chassis frame. The wheel spindles are tubular and are carried at the ends of a large-diameter tube which is curved in the horizontal plane, to clear the center housing. Since the center housing is spring-supported, and therefore is free to move vertically with relation to the road wheels, it is necessary to incorporate two universal joints in each of the shafts connecting the differential to the road wheels. These shafts are not surrounded by tubes, but the universal joints at their ends may be enclosed in the center housing and in the wheel hub. In the original De Dion axle, shown in Fig. 5, the universal joint at the wheel end was located on the inner side of the wheel, while in the more recent exemplifications of this type the wheel spindles are tubes of rather large diameter (sometimes trumpet-shaped) and the universal joint is located at the outer end of the spindle, to increase the length of the axle shaft, thereby reducing the angularity of the joints. Each wheel hub is mounted on two bearings on the outside of the tubular spindle. An advantage of the De Dion type of axle is that it reduces the unsprung weight (because the center housing with the differential and driving gears is spring-supported), which improves the riding qualities of the vehicle.

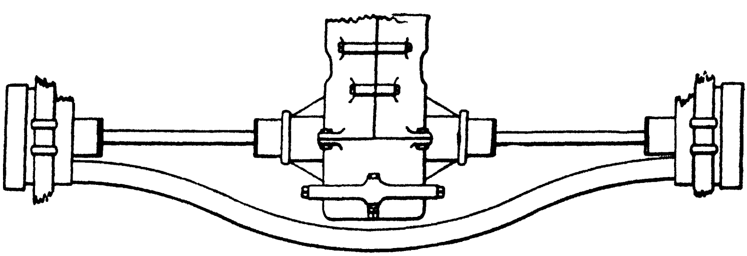


FIG. 5.—DE DION-TYPE DRIVING AXLE.

Articulated Axles—Finally, we have what are variously known as "swinging," "oscillating," or "articulated" axles, which are used on some cars having independent suspension at the rear. With this type, also, the center housing is supported on the chassis frame, and "swinging axles," comprising both an axle tube and an axle shaft, extend from the center housing to the road wheels. The tubes are mounted on the center housing in such a way that they can swing around the axis of the driving-pinion shaft. In one particular design, each axle shaft is driven through a separate pair of bevel gears, the two pairs being of different size but having the same ratio. The differential gear in that case is mounted on the driving shaft between the two bevel pinions. With an axle of this type, no universal joints are required in the axle drive shafts, but since the axles swing in the vertical plane, the wheels on opposite sides of the vehicle do not remain parallel, and the tread changes, which has a tendency to cause scuffing of the tire tread.

Torsional and Bending Moments—As mentioned previously, the only axle types in extensive use in American automotive practice are the "floating" axles, the full-floating type predominating in heavy commercial vehicles, the semi-floating in passenger cars. Whatever the axle type used, the overall loads are the same. That is to say, the load supported by the axle always will create the same bending moments and shear, the power transmitted will produce the same torsional moment, and any lateral forces or shocks also will produce the same bending moments and thrusts. However, in the different axle types these various loads are differently divided between the housing and shafts, and the resulting bearing loads differ materially. It is readily understood that if the axle shafts are subjected to torsion only, they can be made smaller in diameter than if they have to withstand bending moments in addition. On the other hand, in a full-floating axle, two bearings are required for each wheel, where one suffices in the case of semi-floating and three-quarter-floating axles, and though the two bearings of the full-floating axle need not carry any more load than the single bearing of either the semi-floating or three-quarter-floating axle, their cost naturally is greater. One definite advantage of the full-floating axle is that if an axle shaft breaks, it does not let the axle down, and the vehicle therefore can be towed without special equipment to support the broken end of the axle. The chief difference between the various axle types seems to be in the magnitudes of the loads imposed on the bearings and the axle shafts by lateral impacts on the wheels.

Bearing Loads Due to Lateral Forces—Fig. 6 is a diagram of a full-floating axle in which the wheel is shown to be subjected to a lateral force F at the rim. Let R be the radius of the wheel; D, the distance between centers of the wheel bearings; L_1 and L_2, the radial reactions of the wheel bearings on the wheel hub, and T, the thrust reaction of the bearing which takes the thrust load. Then, since the moments around any axis must vanish, we have

$$FR = L_1D \quad \text{and} \quad FR = L_2D.$$

$$L_1 = \frac{R}{D}F = L_2.$$

In practice the expression R/D has a value of about 4, hence, when the wheel receives a lateral shock, each of the two wheel bearings is subjected to a shock load about four times as great as the shock on the wheel. In the case of the inner bearing this adds to the normal load, whereas in the case of the outer bearing the shock load and the normal load are opposite in direction, and the resultant load on the bearing is the difference between the two. Incidentally it may be pointed out that the thrust load on the bearing is equal to the shock, since $T = F$.

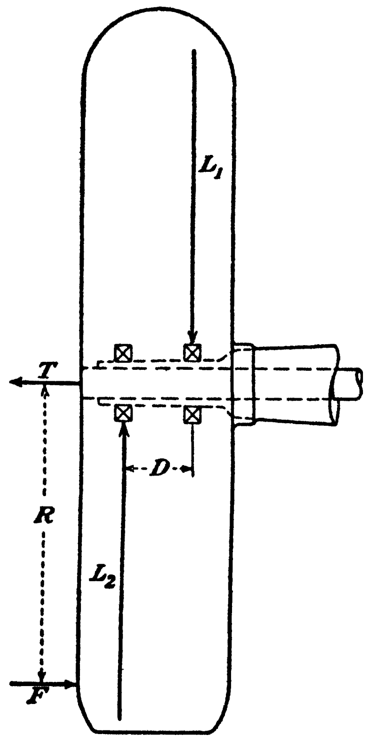


Fig. 6.—Bearing Loads Due to Side Thrust on Wheel in Full-Floating Axle.

The semi-floating axle is shown in diagram in Fig. 7. In this case

$$L_1D = FR = L_2D \quad \text{and} \quad L_1 = \frac{R}{D}F = L_2.$$

In practice $R/D = 0.6$ approximately, so that L_1 and L_2 are each equal to only about 60 per cent of the shock F. We found that in the case of the full-floating axle they are equal to about four times the shock, hence the bearing loads due to lateral shocks on the wheel are roughly seven times as great in the full-floating type as in the semi-floating and three-quarter-floating types of axle.

Axle-Shaft Materials—Owing to the fact that the axles represent unsprung mass, which it is desirable to keep down to a minimum, axle shafts always are made of high-grade steel, and heat treated, and the factor of safety allowed in calculating them is quite low, especially if the shafts are subjected to pure torsion. Either carbon or alloy steel is used, the nominal carbon content usually being either 0.40 or 0.45 per cent. When a shaft of circular section is subjected to torsion, the stress in its material is greatest at the surface, and decreases uniformly toward the center, where it is *nil.* Torsion subjects the material to shear. The ability of a shaft to withstand torsional moments therefore depends on the shearing strength of its material, and steels for such shafts

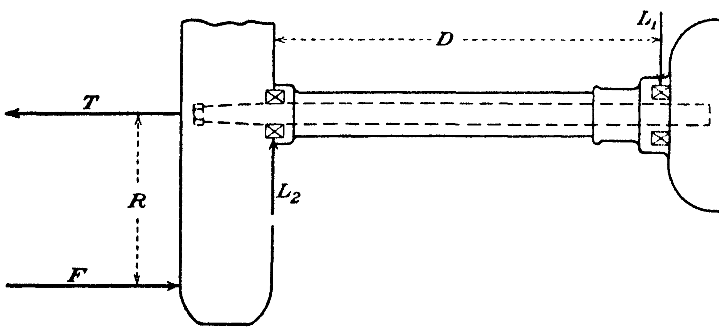


Fig. 7.—Bearing Loads Due to Side Thrust on Wheel in Semi-Floating Axle.

generally are rated according to their elastic limit in shear. The shearing strength of a steel—the same as its tensile strength—varies with its composition and its Brinell hardness, and for a steel of given composition the hardness can be varied within wide limits by changing the drawing or tempering temperature. As the stress in the shaft is greatest at the surface, the shearing strength (and therefore the hardness) of the surface layer is of greatest importance, and it has long been known that when shafts are made of low-carbon (carburizing) steel, their resistance to torsional moments can be greatly increased by case-hardening them.

Heat Treatment of Axle Shafts—Alloy steels of medium carbon content are used most extensively. After a preliminary normalizing or annealing operation, the shafts are quenched in oil from a temperature slightly above the critical range of the steel, and are then tempered or drawn to show the desired hardness. Sometimes the shafts are drawn at a sufficiently high temperature to permit of machining after heat treatment. In that case the heat treatment usually is applied at the steel mill, while the machining is done at the axle plant. This makes it necessary to keep the Brinell hardness down to about 300, which, of course, limits the shearing strength. If greater strength is needed, the shafts are machined before heat treatment, and only certain operations calling for great accuracy, such as cutting of keyways and splines, are left until after the heat treatment. The Brinell hardness can then be carried to 350 or 375. For the greatest strength, shafts of alloy steel are fully machined in advance and drawn at a low temperature to give a Brinell hardness of close to 400. This calls for cold straightening after heat treatment. Alloy-steel axles are always quenched in oil, and the mild action of this quenching medium results in a moderate hardness, which seldom exceeds 500 Brinell. After being hardened, the shafts must be tempered, and even if tempering is effected at a quite low temperature, it will reduce the Brinell hardness to about 440. On the other hand, carbon steel of the 1045 grade can be quenched in water, and such steel, after being drawn at 400 F, will still show a Brinell hardness of about 600. Carbon steel does not possess the same degree of hardenability as alloy steel; that is to say, when test specimens of carbon and alloy steel, respectively, are given a similar hardening treatment, in the carbon-steel specimen the hardness will drop more rapidly with increase in depth below the surface. But in a shaft subjected to approximately equal torsional moments in both directions, the surface hardness is paramount, and the lesser hardenability of the carbon steel therefore is of no great importance.

Torsion Tests—Shearing strengths or torsional strengths are determined in torsion testing machines. In such a machine a specimen in the form of a cylindrical rod or bar is subjected to successively increased torsional moments, and the angular deflection produced by each moment is measured. From the data thus obtained, a stress-strain diagram, or a diagram of torsional moment vs angular deflection, can be drawn. At first the angular deflection is directly proportional to the moment, and the graph therefore is a straight line, but after the elastic limit has been reached the deflections increase more rapidly than the moments, and the line therefore begins to curve. The exact point where this occurs, which marks the true elastic limit, is rather difficult to locate, and it has therefore become customary to locate the point at which the rate of deformation, that is, the increase in deflection for a small definite increase in moment, is 50 per cent greater than at the beginning of stress application, the stress at this point being known as the Johnsonian elastic limit.

Safety Factor—It is customary in engineering to make the maximum working stress in a mechanical part a definite fraction of either its elastic limit or its ultimate strength, the ratio between either of the latter and the former being known as the factor of safety. Half a century ago it was the universal practice to base strength calculations on the ultimate strength of the material. Later it was argued that, since any part subjected to repetitive stresses beyond its elastic limit must eventually fail, the maximum safe stress depends really on the elastic limit, and it then became customary to base strength calculations on this property of the material.

During the past quarter century very extensive researches have been made on the fatigue of metals, and the fatigue limits or endurance limits of various metals and alloys have been determined. This is the maximum repetitive stress which the material will withstand indefinitely without failure. It is determined in fatigue-testing machines, in which the specimen is subjected to a constant succession of stress cycles until it fails. At the start the stress cycles are made to extend well beyond the elastic limit, so that failure occurs relatively early, and the test is then repeated with other stress cycles of gradually reduced range, until the specimen withstands ten million cycles, which is regarded as indicating that it will stand up indefinitely under this cycle. The maximum stress of the cycle is then called the endurance limit of the material. Alternately, the maximum stress of any cycle may be plotted (on the axis of ordinates) against the number of cycles to failure (on the axis of abscissas), which gives a curve convex to the axis of abscissas, and the value of the ordinate at which the curve becomes parallel to the axis of abscissas represents the endurance limit.

It has been found that the endurance limit, which is evidently the maximum safe stress, is more nearly proportional to the ultimate strength and the hardness than to the elastic limit of the material, hence for parts subjected to repetitive stresses at least, the ultimate strength would seem to be the most logical basis for strength calculations in design. Unfortunately, the ultimate strength of a material in shear cannot be accurately determined by a torsion test, hence there is no question but that in strength calculations for members subjected to torsion the elastic limit in shear should be the basis.

The factor of safety applied in calculating axle shafts on this basis ranges between 2 and 2.5.

Relation Between Shear Strength and Other Properties—While the Brinell hardness is more closely related to the ultimate strength than to the elastic limit, there is a fairly close relationship between the Brinell number and the elastic limit in shear of any particular grade of steel, which can be made use of if no torsion-test results for the steel to be used are available. From data collected by the author it appears that the elastic limit in shear, in psi, is equal to approximately 225 times the Brinell number for carbon steels, to 275 times that number for low-alloy steels such as the 3100 and 4100 series, and to 325 times that number for the higher alloy steels of the 3400 and 4300 series. Another relationship sometimes made use of when actual torsion-test data are lacking is that the elastic limit in shear is about 60 per cent of the elastic limit in tension. Of course, no high degree of accuracy can be claimed for any of these empirical relationships, and actual test results should be used whenever possible.

Determining Shearing Stress from Observed Data—When a torsional specimen of diameter d in. is subjected to a moment M lb-in., the shearing stress induced in the fibers at the surface of the specimen is given by the equation

$$S = M/(0.196d^3) \text{ psi},$$

in which the expression $0.196d^3$ represents the polar section modulus of the circular section. This equation is based on the assumption that the stress in the material varies in direct proportion to its distance from the axis (or from the neutral fiber). This holds true as long as the stress at no point of the section exceeds the elastic limit. When the moment required to stress the outermost fibers to the elastic limit is inserted in the above equation, it gives the stress corresponding to the elastic limit in shear. If the moment causing failure of the specimen is inserted, the equation gives what is known as the apparent ultimate shearing strength, which is always greater than the actual shearing strength.

When a shaft of diameter d in. and length l in. is subjected to a torque of M lb-in., the resulting deflection is

$$\theta = \frac{583.6Ml}{Gd^4} \text{ deg},$$

where G is the modulus of rigidity of the steel. An average value of G for steel is 11,500,000 psi, and if this value is inserted the equation for the deflection becomes

$$\theta = \frac{Ml}{19{,}700d^4} \text{ deg.}$$

The shearing stress is related to the shaft dimensions and the deflection by the following equation

$$S = \frac{\theta dG}{114.3l} \text{ psi,}$$

which for $G = 11{,}500{,}000$ psi becomes (substantially)

$$S = 100{,}000 \frac{d}{l} \theta \text{ psi.}$$

Pertinent data for a number of axle-shaft steels are compiled in the following table:

Table I—Properties of Axle-Shaft Steels

S.A.E. Number	*Quenching Medium*	*Drawing Temperature*	*Brinell Hardness*	*E.L. in Shear*	*App. Ult. Strength*
1045	Water *	1200 F †	207	47,000	
1045	Water *	400 F	600	140,000	
3145	Oil	850 F	325	94,400	
3435	Oil	850 F	387	127,200	144,000
4140	Oil	1050 F	300	82,300	127,000
4340	Oil	850 F	400	130,000	

* Ten per cent caustic solution.
† This steel was tested with a view to its use in propeller shafts, and not in axle shafts.

Torsional Fatigue Tests—In the past, fatigue tests, or tests to determine the endurance limits of materials, have been applied chiefly to tensile specimens, but they are now being made also on torsion members. The Timken-Detroit Axle Company applies a torsion fatigue test directly to axle shafts, rather than to small-size specimens, and the author is indebted to R. W. Roush, chief metallurgist of the company, for the following information on the testing machine and the testing method, as well as for some data obtained in this test. The machine, of which a photograph is reproduced in Fig. 8, is hydraulically operated, comprising a hydraulic cylinder of 3 in. bore and 24 in. stroke. The piston of this cylinder connects through a piston rod and connecting rod to a lever arm secured to the splined end of the axle shaft, the flanged end of the shaft being rigidly held in a fixture on the machine.

It is possible to control the applied moment by limiting either the load or the length of stroke. To produce a fiber stress of 60,000 psi in a shaft of 1¾ in. diameter and 40 in. long, it must be deflected about 13 deg from the neutral position. In the test the shaft is deflected through this angle alternately to opposite sides of the neutral position, and this stress cycle is repeated at the rate of 30 per minute until failure occurs.

FIG 8—TORSION-FATIGUE TESTING MACHINE FOR AXLE SHAFTS.

Endurance Limits of Axle-Shaft Steels—The tests conducted with this machine might be described as accelerated endurance tests. With the rotating-beam type of machine, used to determine the endurance limit in tension, the tests usually are continued until the specimens withstand 10 million stress cycles. With this large and relatively slow machine it evidently would be impractical to continue the test for so long a time, and it is therefore limited to about 200,000 cycles. This practice is justified by the fact that in automotive service, the moments on which stress calculations are based, corresponding to full vehicle load and maximum engine torque, occur only rarely.

All of the popular axle-shaft materials have been tested on this machine, and results obtained with five different grades of steel, each drawn to come within a particular Brinell-hardness range, are plotted in Fig. 9. Strange as it may seem, the greatest endurance is shown by the carbon steel 1045. This is due to the fact that this steel is being quenched in a caustic solution and drawn at the relatively low temperature of 400 F, which results in a far greater surface hardness than obtained with any of the other steels.

The experimental results shown by the chart have been confirmed by service experience. In a number of cases where failures with shafts of alloy steels were relatively high, these shafts were replaced with others of 1045 carbon steel, caustic-quenched and drawn at 400 F, and the trouble ceased. In the caustic-quenched carbon-steel shafts the surface hardness extends to a depth of from $\frac{1}{8}$ to $\frac{3}{16}$ in. It has been found that the carbon-steel shafts are not highly sensitive to surface imperfections, unless these extend all the way through the hardened shell. Surface imperfections often prove disastrous to alloy-steel shafts, in which the hardness is more nearly uniform throughout. Most alloy steels have some "banding" and directional properties, which it is difficult to eliminate by processing. Under torsion, incipient fractures advance longitudinally, and if the shaft is hard throughout, it finally splits. It has been shown by Almen that the excellent performance of water-quenched shafts is due to a high residual compressive stress in the surface material.

Increases in the endurance limits of axle shafts can be achieved also by shot-peening and rolling—processes which put the surface layer of the material under compression.

Strength of Splined Shafts—As axle shafts have splined connections at either one or both ends, the question of the strength of splined shafts arises. Extensive tests bearing on the relative strengths of splined shafts and solid shafts of an outside diameter equal to the diameter at the bottom of the splines were made by C. W. Spicer and reported by him in an S.A.E. paper in 1927. The tests were made on ten-spline shafts of the proportions used for permanent connections, of 1045 steel, and the Johnsonian elastic limit was used. It was found that the elastic limit in shear of the splined shafts averaged 8 per cent lower than that of solid round shafts of an outside diameter equal to the diameter at the bottom of the splines. The apparent ultimate strength of the splined shafts in all cases was materially greater than that of the solid shafts with which they were compared.

Designers usually assume that a splined section has approximately the same strength in torsion as a full round section of a diameter equal to that at the bottom of the splines, and splined shafts have their ends upset sufficiently so that the diameter at the bottom of the splines is equal to the basic diameter of the shaft. Splines usually are cut by means of hobs, hence the spline grooves run out gradually, and the transition from the upset to the basic diameter also is made gradual, to minimize stress concentration. When splined shafts first came into use in automotive practice, about 1910, four- and six-splined designs were used predominantly; at present ten splines are in common use for passenger-car axles,

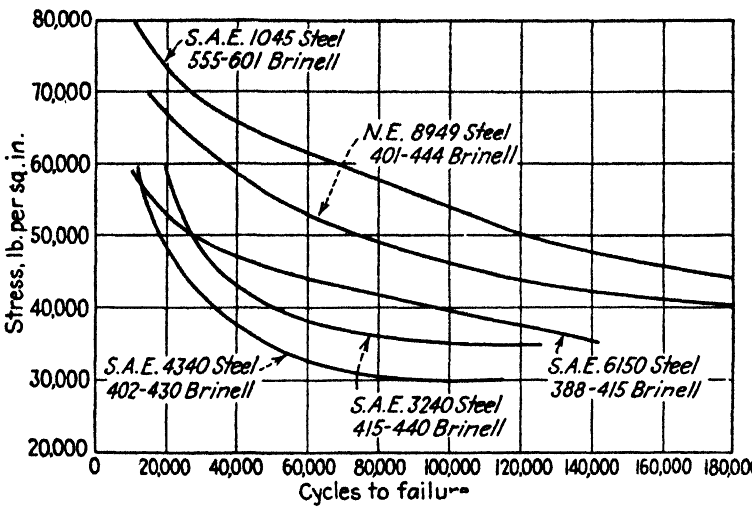


FIG. 9.—FATIGUE-LIFE CURVES FOR AXLE-SHAFT STEELS.

and 16 and even 18 splines are cut on the axle shafts of heavy commercial vehicles. The greater the number of splines, the smaller the depth necessary to give a certain contact area per unit of length, and multiple-splined joints save both material (cost) and weight.

Where multiple-splined shafts are used it is now customary to make the bottom of the spline grooves in the shaft semi-circular, instead of flat or concentric with the shaft. This eliminates concentration of stress at sharp corners, and also tends to conserve the cutting tools. Some manufacturers use involute instead of flat-sided splines. The strength of splined shafts is largely a matter of fatigue rather than of static strength, and the ratio of fillet radius to shaft diameter is important. If the ends are shot-peened, the shot must have

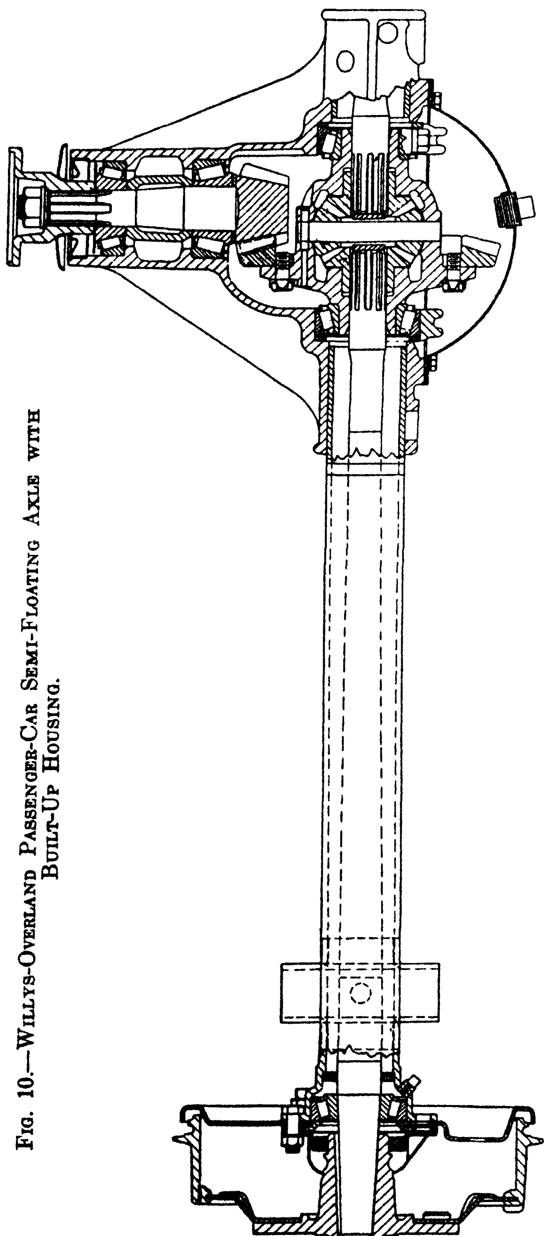


Fig. 10.—Willys-Overland Passenger-Car Semi-Floating Axle with Built-Up Housing.

a smaller radius than the fillet, to enable it to impinge effectively on the fillet.

Types of Axle Housing—There are at least four types of axle housing in general use today. The oldest is the built-up type, of which the Willys, shown in section in Fig. 10, is an example. It consists of a center housing of malleable cast iron, with bosses on the sides into which seamless steel tubes are fitted. At their outer ends the tubes are expanded to accommodate the outboard bearings of the axle, and provided with a flange to which the brake backing plate is bolted. The same bolts hold a bearing-retainer ring in place. Sometimes the bearing-housing and brake-backing-plate flange is welded to the tube. Spring rests or spring pads are welded to the axle tubes. It will be noted that in the Willys axle the center housing is provided with substantial ribs between the forwardly extending member and the lateral bosses, to stiffen it in the plane of the gear axes.

A more common housing for semi-floating and three-quarter-floating axles is the pressed-steel type, which is made up of two mild-steel stampings welded together by either the acetylene torch, the electric arc, or flash-welding. The welds generally are at top and bottom. End fittings are butt-welded to the axle tubes. Large circular openings are formed in the center housing at the front and rear, and the edges of these openings are reinforced by turning the metal over in the press, or by placing reinforcing rings of flat stock on the inside. Such reinforcement is necessary also to provide sufficient thread for the cap screws which hold the gear carrier in place. The reinforcing rings, of course, also add to the stiffness of the housing in the vertical plane.

Pressed-steel axle housings are produced in a number of different designs. A plain pressed-steel housing is shown in Fig. 11. In this design the central "banjo" is reinforced by an inner shell of pressed steel which fits it snugly. The tubular portions of the housing increase in section slowly in the horizontal and more rapidly in the vertical plane as the center housing is being approached. The flange at the forward side of the center housing is cut away at certain points to permit of the introduction of the differential carrier with its crown gear and differential-housing bosses, with a minimum outside diameter of the shell, and therefore maximum ground clearance. The openings in the inner shell through which the axle shafts pass are not much larger than the upset ends of the shafts, and the walls of the shell serve as oil dams. In a Cadillac axle the spherical rear cover extends entirely through

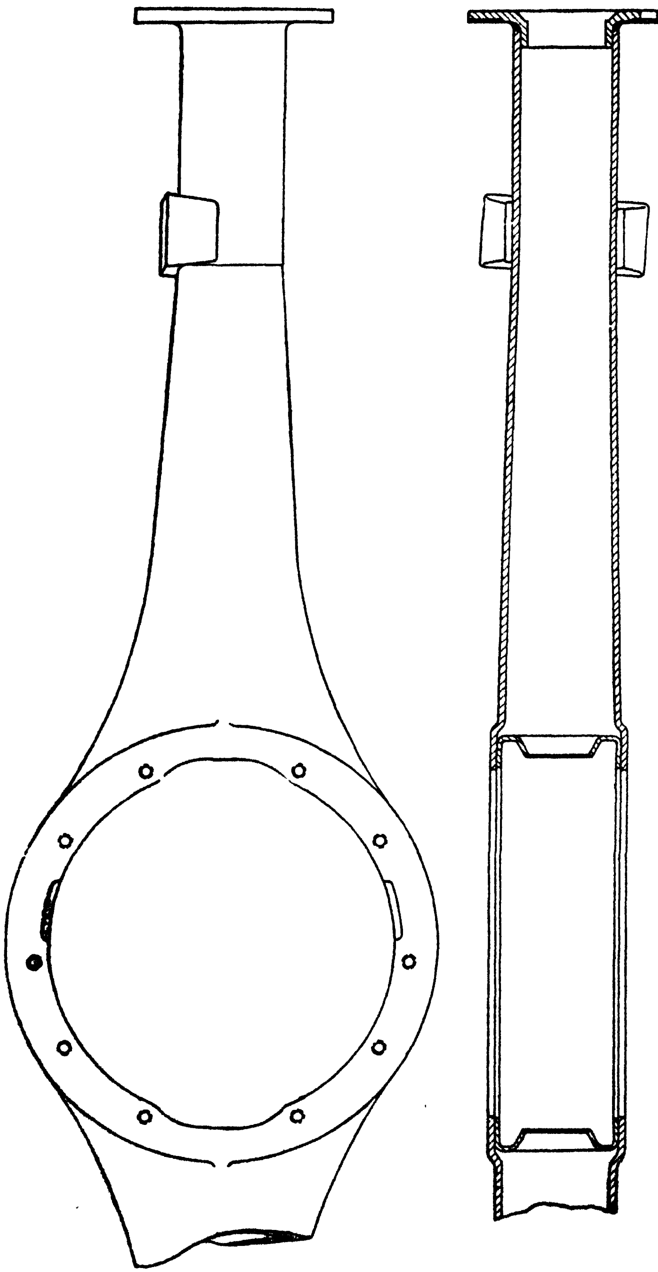


FIG. 11.—PRESSED-STEEL AXLE HOUSING.

the center housing, as shown in Fig. 12. It is welded to the rear and bolted to the forward flange, and therefore serves as a reinforcement for the housing. Some other pressed-steel

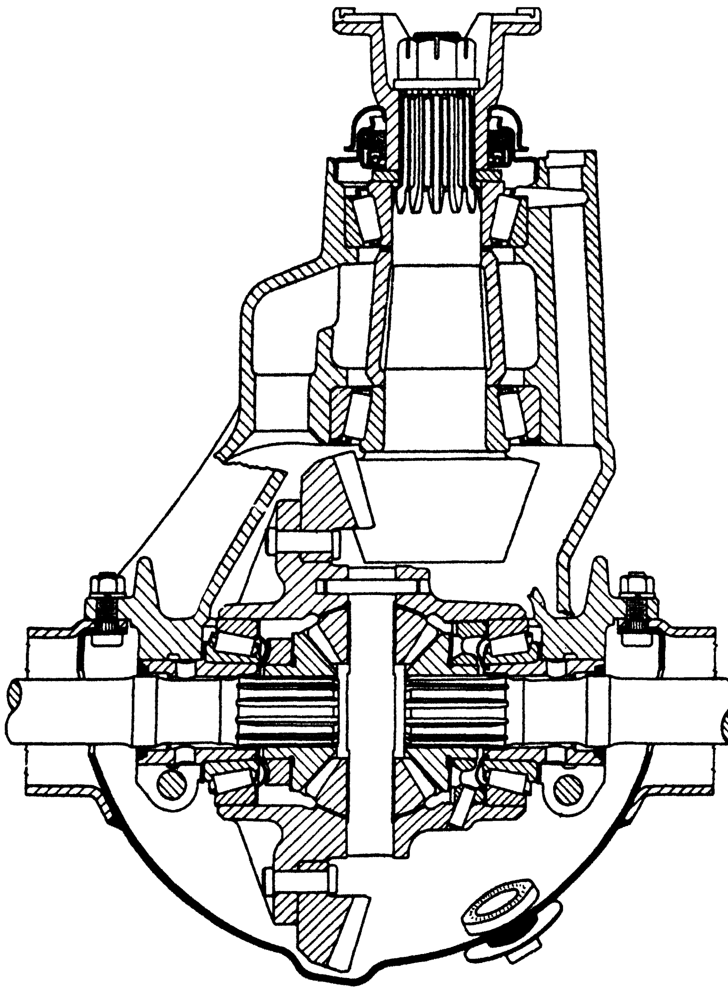


FIG. 12.—AXLE HOUSING REINFORCED BY WELDED-ON COVER PLATE (CADILLAC).

axle housings also have the cover plate welded to the housing, instead of having it secured thereto by cap screws.

Cast Axle Housings—Cast axle housings are extensively used in heavy commercial vehicles. These housings are really

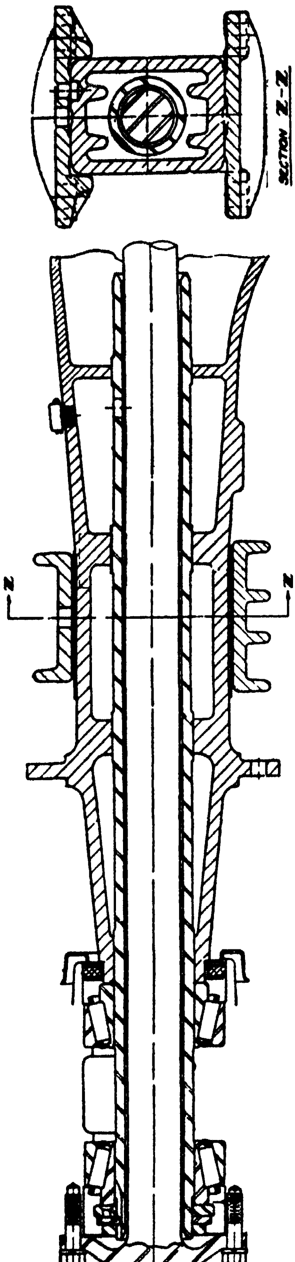


Fig. 13.—Malleable-Iron Axle Housing with Steel Sleeves (Timken-Detroit).

of composite type, comprising a central cast member into which seamless steel tubes are forced from both ends. Fig. 13 shows the housing of a Timken axle cast of malleable iron, with sleeves of alloy-steel tubing pressed into it. Like the pressed-steel housings, it is of the banjo type, the differential carrier being secured to the center housing with cap screws. The arms extending from the center housing are of rectangular section, and provided with internal ribs which are bored out to receive the sleeves, the latter being undercut at other than the seating portions, to facilitate their introduction into the housing. The cast portion of the housing extends only as far as the wheel hubs, all of the wheel bearings being mounted on the sleeves. As one of the bearing cones bears against the end of the housing and the corresponding cone of the other bearing against an adjusting nut on the sleeve, there is a tendency to withdraw the sleeves from the housing, and the former are held in the housing against axial forces by dowels or pin screws. Flanges for the attachment of brake backing plates are cast integral with the housing, while the spring pads are separate and are clamped and doweled to the housing. Separate spring pads, of course, are a convenience in a stock axle, as with them the distance between spring centers can be easily changed. If radius rods are to be used, each spring pad is provided with a lug to which the rod can be pin-jointed.

Another design of composite axle housing, used on an English truck, is illustrated in Fig. 14. The center portion of the housing is a steel casting, and its arms extend only as far as the brake backing plates, which latter are clamped between flanges at the ends of the casting and what may be described as extensions of the housing. Each sleeve at this point is provided with a small external flange which fits a counterbore in the end of the large casting, whereby it is held positively against axial movement. Angular movement of the sleeve is prevented by a key in the housing extension. There is an axial extension also on the wheel hub, which telescopes that on the axle housing, and the joint between the two extensions is provided with a grease seal. There is an additional seal in the wheel-hub extension. The axle shafts of this axle drive to the wheel hubs through dog clutches, and the axle therefore is truly full-floating.

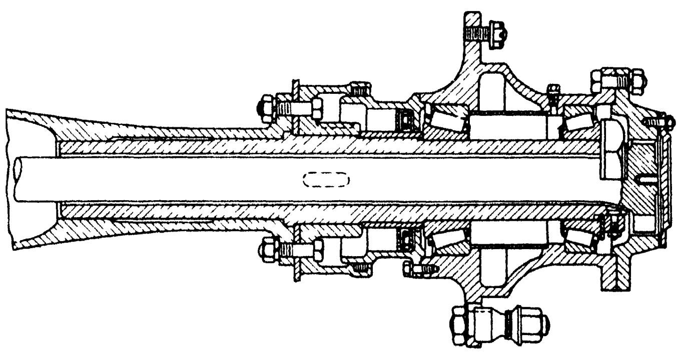


FIG. 14.—BRITISH TRUCK AXLE WITH COMPOSITE HOUSING.

Forged Housings—Axle housings can be produced also by a forging process, and important advances in this method of production are due to the Clark Equipment Company of Buchanan, Mich., who produce such housings from tubular blanks. The production process actually starts with a plate of 1035 steel, which is rolled into a tube and then welded. Various steps in the production of the housing are illustrated in Fig. 15, but the first two stages, the steel plate and the rolled, unwelded tube, are there omitted. After being welded, the tube is perforated at the center, as shown. Next each end of the tube adjacent to the central perforated section is reduced in size, to thicken the section of the arms to which the spring pads are welded. In the next two operations the banjo portion is formed, and then the ends of the arms are

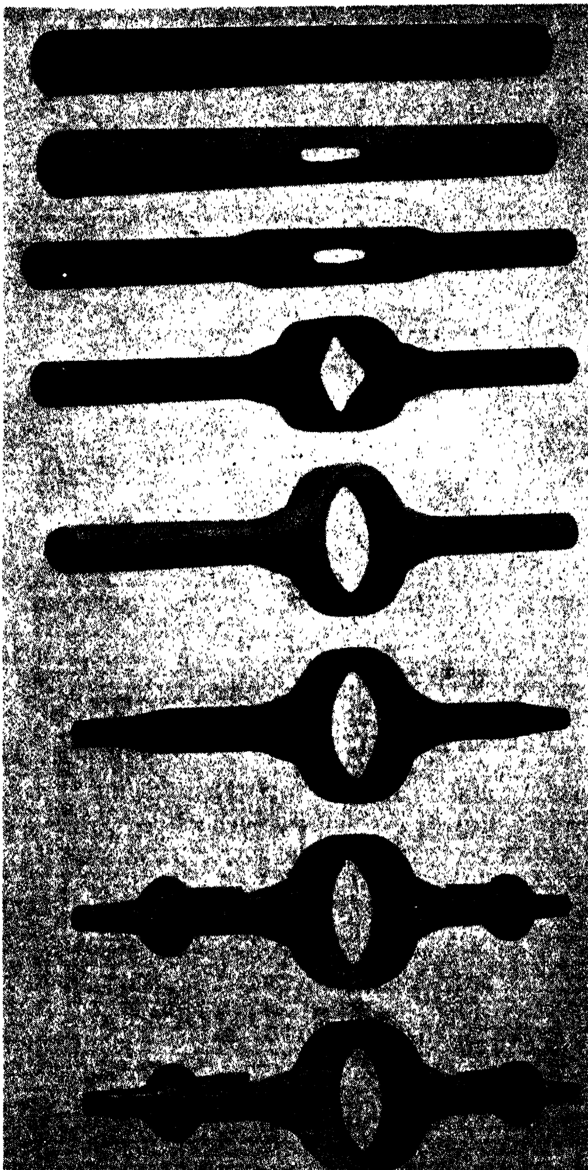


FIG. 15.—STEPS IN THE PRODUCTION OF AXLE HOUSINGS FROM TUBULAR BLANKS.

swedged down. Spring pads and flanges for the backing plates are welded on, together with reinforcing rings on the faces of the banjo, which also serve to provide sufficient stock for studs on which to mount the differential carrier. During the forming operations the material is thoroughly worked, and its mechanical properties are further improved by suitable heat treatment. After the housing has been heat-treated, a number of machining operations are performed on it. This is a housing for a full-floating axle, and it is in wide use for trucks of all sizes. It is a one-piece housing of the banjo type, and it is claimed that the section is so controlled that the material is stressed substantially uniformly throughout.

Pinion-Shaft Mountings—The pinion of the final-drive gears can have either an overhung or a straddle mounting. The former predominates in passenger-car, the latter in heavy-duty bus and truck axles. With an overhung mounting the stub shaft of the pinion is carried in two bearings, and in order that the pinion may be rigidly supported, these bearings should be located a considerable distance apart. A spacing of bearing centers equal to at least 2.5 times the overhang of the pinion center over the center of the near bearing is recommended. Provision must be made for taking thrust in both directions, and with the overhung mounting it is customary to use two angular-contact bearings, either ball or roller, back to back, that is, with the cups or female cones toward each other. One male cone rests against a shoulder on the pinion shaft, the two female cones rest against internal flanges in the bearing housing, and the remaining male cone is supported by an adjusting nut or a spacer. A washer may be placed between the pinion and the adjacent bearing for purposes of gear adjustment. It was formerly customary to provide special adjusting nuts for the bearings, independent of the companion flange for the universal joint at the forward end of the shaft, but at present many designers use the splined hub of the companion flange as a spacer, and one and the same nut to adjust the bearings and hold the companion flange in place.

The antifriction bearings, instead of being mounted directly in the center housing, may be mounted in a flanged sleeve which is set into a bore of the housing, in which case pinion-adjustment is accomplished by means of shims between the housing and the flange on the sleeve.

With the straddle mounting the arrangement of the bearings back of the pinion is similar, except that the two are located close together, on opposite sides of a narrow internal flange in the bearing housing, on which thrust loads are taken.

The bearing at the inner end of the pinion takes radial loads only, and is usually of the cylindrical-roller type. Thrust loads, of course, are the same with both mountings, but radial loads on the bearings are materially less with the straddle mounting. However, the chief advantage of the straddle mounting is that it reduces the deflection of the pinion under load. An oil seal is always provided at the forward end of the pinion-bearing housing.

Rear-Axle Torsion and Thrust—All of the forces and moments or couples originating in the rear axle give rise to reactions. There are two moments or torques, namely, the torque on the pinion shaft and the torque on the axle shafts, and each of these produces a torque reaction which is equal to it in magnitude but opposite to it in direction. Referring back to Fig. 12, the driving bevel pinion, when looked at from the rear of the car, turns left-handedly and presses down on the teeth of the bevel gear. At the same time the shaft of the pinion presses upward on the bearings supporting it, with an equal force. The direct pressures of the pinion teeth on the gear teeth and of the pinion shaft on the bearings constitute a couple which is balanced by another couple due to an increase in the upward reaction of the pavement on the left, and a decrease in the reaction on the right rear wheel. In other words, the effect of the torque on the pinion shaft is to increase the load on the left rear wheel and to decrease that on the right rear wheel. The weight removed from one and added to the other wheel can be readily found by dividing the pinion-shaft torque, in lb-in., by the wheel tread in in. This transfer of weight reaches its maximum when the rear axle torque is so great that the wheels begin to slip on hard, dry pavement, in which case, in a car with standard tread, it is equal to about 7 per cent of the weight originally on each rear wheel. The balance of the car therefore is not disturbed, and it is not necessary to take any steps to compensate for the effects of propeller-shaft torque.

It is different with the reaction to the torque on the axle shafts, which is greater than the propeller-shaft torque in the proportion of the rear-axle reduction ratio to unity. When the rear wheels are being turned around or urged around by the engine in one direction with a given torque, the axle housing is urged around in the opposite direction with the same torque. As the axle housing is freely supported in the wheel hubs or on the axle shafts, on antifriction bearings, there is nothing to prevent it from turning, except the members which connect it to the vehicle frame. One or more of these connections must be so designed that they are capable of resisting the maximum torque that may be impressed on the housing.

The magnitude of the torque on the axle housing has been often misjudged. A good idea of it can be obtained by reflecting that to balance it at the end of an arm equal in length to the wheel radius, requires a force equal to the propelling effort exerted by the two wheels.

In addition to the torque, the driving force must be provided for. All of the driving effort originates at the contact of the driving wheels with the ground, whereas much of the resistance to motion originates at the front wheels and on the surfaces of the body (air resistance), and the rear axle virtually must push the car body and front axle forward. In the opposite case, when brakes are applied to the rear wheels, the driving effort originates mainly in the chassis (due to its inertia) while the rear axle tends to remain behind, and then the chassis pulls the rear axle instead of being pushed by it. Hence it is necessary to have a connection or connections between the axle and the chassis frame which can transmit forces in both directions, or in other words, transmit both a pull and a push.

The Hotchkiss Drive—In many vehicles both the rear-axle torque and the propelling and retarding forces are taken up by the rear springs, the method of construction making this possible being known as the Hotchkiss drive. Cars with Hotchkiss drive always have half-elliptic springs, which are rigidly secured to spring seats on the axle housing and are pinned to the chassis frame at their forward end. The Hotchkiss drive calls for springs that are substantailly flat when under load. The springs naturally yield under severe torsion, which is of advantage in that it tends to protect the transmission members, the wheels and tires against shock. Of course, the springs are rather severely stressed by their double duty of cushioning road shocks and counter-acting rear-axle torque, particularly the main leaves, and these should always be made of high-quality spring steel.

Torque Tubes—Another means of taking care of the torque reaction and the driving thrust is the torque tube, which surrounds the propeller shaft. The torque tube is fitted rigidly to the rear-axle center housing and eliminates the need for a universal joint at the rear end of the propeller shaft, which latter is connected to the bevel-pinion shaft by a splined and pinned coupling. At the forward end the torque tube is supported from either a cross-member of the frame or from the rear end of the transmission case. A method of supporting the forward end of the torque tube by a ball-and-socket joint is illustrated in Fig. 16. There one half of the socket is a casting which bolts to the rear of the transmission housing, and also serves to hold the outer race of the main-drive-shaft bearing in place. The other half is a sheet-metal drawing. Both halves are flanged, and they are held together by bolts. The ball is formed on a sleeve fitting over the forward end of the torque tube. The universal joint is located centrally within the ball and socket, and the chief advantage of this type of forward-end support is that it provides a dust- and oil-proof housing for the universal joint, which latter, therefore, need not be of the enclosed type. In this particular design the space within the socket communicates with the interior of the transmission housing, and the same supply of lubricant serves for both compartments. An oil guard in the ball keeps oil out of the torque tube.

Instead of by a ball and socket, the forward end of the torque tube may be supported by a forked connector which swivels on the tube and is pin-jointed to forked lugs on a

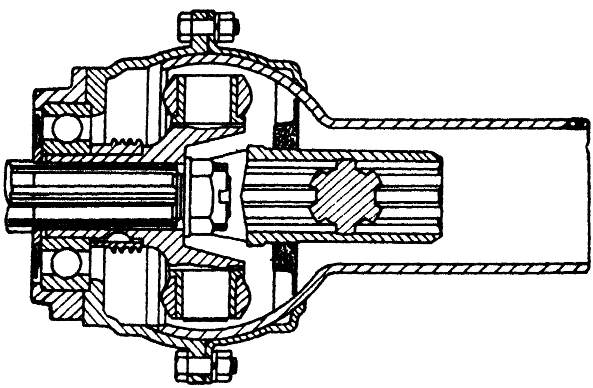


FIG. 16.—BALL-AND-SOCKET SUPPORT FOR FORWARD END OF TORQUE TUBE.

frame cross member. The axis of articulation, of course, must pass through the center of the universal joint. This method of support, which is illustrated in Fig. 17, is more or less obsolete, its disadvantage, as compared with the ball-and-socket support, being that with it there are three additional bearings requiring lubrication. When torque-tube drive is used, the spring seats may be made free to rotate on the axle housing. However, it is possible also to use spring seats rigid with the axle, together with unsymmetrical springs, so proportioned that they are not subjected to additional stresses due to the torque. In the Chevrolet the front end of the torque tube is adapted to slide, and therefore transmits no driving thrust. The latter is taken by the springs, which are swiveled on the axle and shackled only at the rear.

Diagonal Brace Rods—In any vehicle in which the driving thrust and brake pull are taken on a central torque tube and the rear springs are connected to the car frame by shackles throughout, the rear axle is subjected to important bending moments when one rear wheel strikes a large road obstruction or only one of the rear brakes is effective. Such moments are generally taken care of by providing diagonal braces between the spring seats or brake supports on the rear axle housing and the forward end or an intermediate point along the length of the torque tube. A rear-axle construction with diagonal

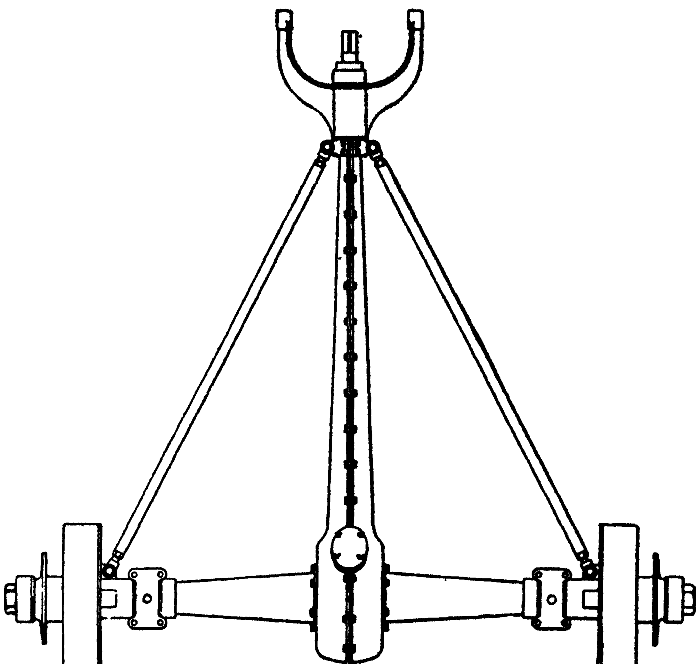


FIG. 17.—DIAGONAL BRACE RODS AND YOKED SUPPORT FOR TORQUE TUBE.

brace rods is shown in Fig. 17. The braces are generally tubular and may be provided with pinned joints at both ends, or they may have a pinned joint at the rear end while at the forward end a threaded rod welded to the tube may extend through a hole in a lug on a torque tube fitting and provided with nuts on both sides of the lug. In the Buick, which has torque-tube drive and rear-suspension on coil springs, the braces are pressed-steel channels with the flanges turned outward, and they, together with the spring rests, are clamped to the axle housing.

Torque Arms—Instead of a tube surrounding the propeller shaft, one or two arms or bars may be used to take the torque reaction. A single torque arm would be mounted on the center housing of the axle and have a pivotal or link connection to the frame at its forward end. Double torque arms, which are sometimes used in connection with rear suspension on coil springs, are mounted on the axle tubes at the spring seats and are made to approach each other toward their forward ends, where they may be supported on brackets secured to a cross member back of the tunnel in the X member of the frame. A drawing of such a torque arm is shown in Fig. 18. If the torque arms have fixed pivots on the frame, the axle will swing around these pivots, and one of the yokes of the universal joint at the forward end is then arranged so it can slide on its shaft, to compensate for changes in the distance between the transmission main drive shaft and the bevel-pinion shaft with spring action. Such torque arms in passenger cars are provided with rubber bushings at both ends,

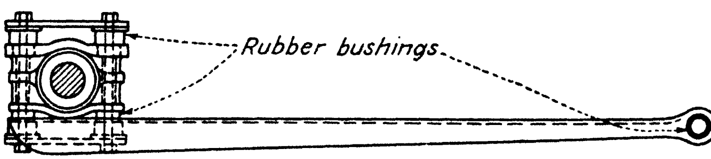


FIG. 18.—PRESSED-STEEL TORQUE ARM.

to prevent the transmission of shocks and noises from the axle to the frame and body.

With rear suspension on coil springs there are usually three other connections between the axle and frame. One of these is a track bar extending parallel with the axle and having a pivotal or universal connection to the axle at one end and to the frame at the other. Its chief object is to prevent lateral motion of the body relative to the axle (swaying) on curves. Another is a stabilizer bar, which also extends parallel with the axle and is carried in bearings on either the frame or the axle. It is provided with arms at or near its ends which connect by links to the axle or frame. The third connection is through the shock absorbers, which parallel the springs. Track bars, stabilizers, and shock absorbers will be discussed in the chapters on suspension.

Radius Rods—Radius rods are struts extending between the axle and a frame member, having pivotal connections with both, so that they are unable to take torque reaction. They came into use first in connection with side-chain drive, where they are needed for the adjustment and maintenance of the distance between the axes of chain sprockets. They are no longer being used on passenger cars, but some heavy trucks still have them. The object of radius rods is to transmit the driving and braking forces. If the rear springs are located outside the chassis frame, the radius rods can conveniently be installed below the frame side rails. Connections to the frame and axle housing preferably should be of the universal type, so that the rods and their connections will not be unduly stressed by side sway of the body. Fig. 19 shows a typical design of radius rod.

It may help to give an idea of the latest practice in providing for rear-axle torque and driving and braking forces to state that of 39 passenger-car models in production in the United States in 1950, 30 had Hotchkiss drive, 6 had torque tubes taking both torque and thrust, 1 a torque tube taking torque only, the driving thrust being taken on the springs, and 2 had torque arms. Of more than 400 truck models listed, 75 per cent had Hotchkiss drive, 19 per cent had radius rods, and the remainder had torque arms.

Axle Shafts for Semi-Floating Axles—In semi-floating axles the shafts are subjected to both torsion and bending moments. The bending moments vary along the length of the shaft, and the shaft diameter should vary correspondingly. Stresses in the shaft become a maximum when the wheels are either being slipped on hard, dry pavement by a sudden application of power, or when they are locked on the same kind of pavement by the brakes. It is generally assumed that the friction coefficient between tire and road under these conditions is 0.6, and the frictional force on the tire tread when the wheel is either being spun by power or is locked by the brakes is $0.6W$, where W is the load supported by the wheel. In passenger cars, at least, the engines usually have sufficient power to spin the wheels on hard, dry pavement when the power is applied through the low gear.

Suppose the engine develops a maximum torque of T lb-ft, that the low-gear ratio of the transmission is r_t, the rear-axle ratio r_a, and the effective rolling radius of the wheel R. Then the maximum torque which the engine can impress on each rear wheel is $Tr_tr_a/2$ lb-ft, or $6Tr_tr_a$ lb-in., and the maximum tractive force on each wheel is $6Tr_tr_a/R$ lb. Even if this should be less than the frictional force $0.6W$, the wheels might still be spun momentarily by the inertia of the flywheel and other revolving parts, if the clutch were allowed to engage suddenly. What limits the tractive force in that case is not the maximum engine torque, but the maximum torque capacity of the clutch, which is usually some 30 per cent greater.

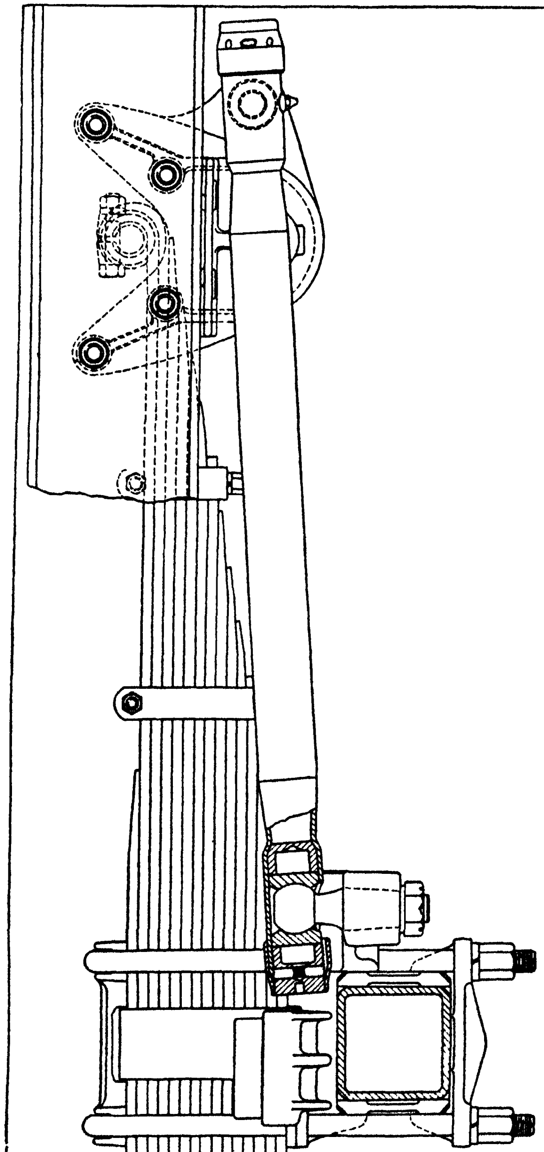


FIG. 19.—TUBULAR RADIUS ROD EXTENDING BETWEEN BALL STUDS ON SPRING PAD ON AXLE HOUSING AND ON SPRING BRACKET ON FRAME RAIL.

Therefore, it is the safest plan to base axle calculations on the torque required to spin the wheels on dry, hard pavement, even if the engine torque should fall short of this by as much as 25 per cent.

Now let l be the distance between the center planes of the road wheel and the outboard bearing of the axle, respectively. Then the reaction W of the pavement on the wheel produces a bending moment Wl on the axle at the center of the bearing. But if the wheel slips on the pavement there is a frictional force $0.6W$ acting on the tire tread which, in addition to a torque $0.6WR$, produces a bending moment $0.6Wl$ on the shaft at the center of the outboard bearing. We therefore have two bending moments on the shaft at the outboard bearing, a vertical moment Wl and a horizontal moment $0.6Wl$, the resultant of which is

$$M_b = \sqrt{(Wl)^2 + (0.6Wl)^2} = 1.17Wl.$$

The resulting stress in the shaft is

$$S_t = \frac{M_b c}{I} = \frac{1.17Wlc}{I},$$

where c is the distance from the neutral axis to the outermost fiber of the shaft, and I the moment of inertia of the shaft section. The torque or torsional moment on the shaft is $0.6WR$, and the resulting shearing stress in the shaft,

$$S_s = \frac{Tc}{J} = \frac{0.6WRc}{J},$$

where J is the polar moment of inertia of the shaft section ($= 2I$). According to the well-known formulae for combined stresses, the maximum tension (or compression) in the shaft is

$$t = \tfrac{1}{2}S_t + \sqrt{S_s^2 + \tfrac{1}{4}S_t^2},$$

and the maximum shear,

$$s = \sqrt{S_s^2 + \tfrac{1}{4}S_t^2},$$

and substituting the values of S_t and S_s found in the foregoing we get

$$t = 0.585\frac{Wlc}{I} + \sqrt{\left(\frac{0.6WRc}{2I}\right)^2 + \frac{1}{4}\left(\frac{1.17Wlc}{I}\right)^2}$$

and

$$s = \sqrt{\left(\frac{0.6WRc}{2I}\right)^2 + \frac{1}{4}\left(\frac{1.17Wlc}{I}\right)^2}.$$

Now, for a circular section $c = D/2$ and $I = \pi D^4/64$. Making these substitutions and simplifying we get

$$t = \frac{32W}{\pi D^3}(0.585l) + \sqrt{0.09R^2 + 0.34l^2}$$

and

$$s = \frac{32W}{\pi D^3}\sqrt{0.09R^2 + 0.34l^2}.$$

In making use of these equations, instead of assuming the weight W carried by one wheel with the car fully loaded, we can use a weight W under which the engine would just be able to spin the wheels in low gear with a friction coefficient of 0.6. Suppose, for instance, that the engine can develop a maximum effective torque of 200 lb-ft, that the low-gear ratio of the transmission is 2.68, the rear-axle ratio 4.25, and the effective wheel radius 13 in. Then the maximum torque on each wheel would be

$$\frac{200 \times 2168 \times 4.25 \times 12}{2} = 13{,}660 \text{ lb-in.},$$

and the maximum tangential force on the wheel tread,

$$13{,}660/13 = 1050 \text{ lb},$$

and with a friction coefficient of 0.6 this corresponds to a maximum wheel load of 1750 lb. Now suppose we have an axle steel with an elastic limit in shear of 98,000 psi, so that with a factor of safety of 2 the permissible shearing stress is 49,000 psi. Then, applying the formula for the torsional strength of shafts,

$$13{,}660 = 0.196 \times 49{,}000 \times d^3,$$

hence

$$d^3 = \frac{13{,}660}{0.196 \times 49{,}000} = 1.42$$

and

$$d = 1.125 \text{ in.}$$

This should be the minimum diameter of the shaft, near the differential end, where the load on it is nearly entirely torsional. Now assume that a bearing of 35 mm (1.378 in.) bore is to be used as the outboard bearing, where the bending moments are at their maximum. There the maximum tensile and shearing stresses would be

$$= \frac{32 \times 1750}{3.14 \times 1.378^3} (0.585 \times 2.5) + \sqrt{0.09 \times 13^2 + 0.34 \times 2.5^2}$$

$$= 38{,}300 \text{ psi}$$

and

$$s = \frac{32 \times 1750}{3.14 \times 1.378^3} \sqrt{0.09 \times 13^2 + 0.34 \times 2.5^2} = 28{,}360 \text{ psi.}$$

Thus both the tensile stress and the shear in the shaft at the outboard bearing are quite moderate, and there would seem to be sufficient margin for any additional stress due to lateral shocks to the wheels.

Provision for Thrust Loads—Most passenger and light commercial vehicles now are equipped with disc wheels, and with semi-floating and three-quarter-floating axles the hubs or centers of these wheels may be either made separate, as in the Willys axle shown in Fig. 10, or they may be forged integral with the axle shaft, as indicated in Figs. 3 and 4. In the former case the outboard bearing can be threaded over the axle shaft from its outer end, and its inner race may then be seated on a tapered section of the shaft, or rest against a shoulder. A tapered seat seems to be preferred, because it obviates the need for a sudden change in the section at a critical point. If the axle shaft is forged with an integral flange at its outer end to serve as a wheel center, the outboard bearing must be threaded over it from its inner end, which makes it necessary that the bore of the bearing be slightly larger than the diameter over the splines at the differential end of the shaft. The bearing is then fixed axially by a shoulder on the shaft on its outer, and by a nut or collar thereon on its inner side, the collar being shrunk on. If the outboard bearing is a single angular-contact bearing, it can take thrust loads in one direction only, and thrust in the opposite direction—toward the differential—must then be transferred through the two axle shafts to the outboard bearing on the opposite side of the vehicle. To this end a thrust block is placed between the adjacent ends of the two shafts, at the center of the differential gear, with a hole through it to allow the shaft of the differential pinions to pass. End play of the axle shafts is adjusted by means of shims between the outer races of the outboard bearings and the bearing retainers.

Stresses on Axle Housing—In a semi-floating axle the static load on the outboard bearing is equal to the load on the adjacent wheel minus the weight of the wheel and the axle shaft. With the vehicle in motion, this load is increased by the driving force or the retarding (braking) force, which forces attain their maximum values when the wheel is either "spun" by the engine or locked by the brake. In accordance with the result arrived at in the foregoing, the bearing load under these conditions may be assumed to be $1.17W_1$, where W_1 is the static load. The driving and braking forces, of course, are proportional to the wheel load W, rather than to the bearing load W_1, but since these two loads are nearly the same, it is permissible to apply the coefficient 1.17 to the bearing load. The bearing load produces a bending moment on the axle housing which increases uniformly from zero at the center of the bearing to $1.17W_1l_1$ at the center of the spring seat, and remains constant between spring seats. It is here assumed that the driving force is transmitted to the chassis frame by the springs or by radius rods located directly below the springs. Therefore, the maximum bending moment on the axle housing is $1.17W_1l_1$. However, the axle housing, the same as the axle shafts, is subjected to combined stresses —bending and shear. The housing is subjected to torque under all operating conditions. When the car is being driven, torque is applied to it through the bearings of the pinion shaft, and when the car is being braked, torque is applied through the brake backing plate or brake support. The maximum possible torque in each case is $0.6WR$ lb-in. The stress in the material of the housing due to maximum bending moment is

$$S_t = \frac{1.17W_1l_1c}{I}$$

and that due to the maximum torque,

$$S_s = \frac{0.6WRc}{J}.$$

Insérting these values in the equations for combined stresses and substituting the values of c, I and J, we get for the maximum tensile stress

$$t = \frac{0.585W_1l_1c}{I} + \sqrt{\left(\frac{0.6WRc}{J}\right)^2 + \frac{1}{4}\left(\frac{1.17W^1l^1c}{I}\right)^2}$$

and for the maximum shearing stress,

$$s = \sqrt{\left(\frac{0.6WRc}{J}\right)^2 + \frac{1}{4}\left(\frac{1.17W_1l_1c}{I}\right)^2}.$$

Now let us assume we have a semi-floating axle for a passenger car in which the maximum load W on each rear wheel is about 1250 lb and the maximum load on the outboard bearing about 1200 lb. Also, let the effective wheel radius R be 13 in., and the center distance l_1 between the outboard bearing and the spring seat, 9 in. Assume that the axle housing has an outside diameter of 3 in. at the spring seats and is made of No. 7 gauge (0.180 in.) stock, so that the inside diameter is 2.64 in. Then

$$c = 1.5 \text{ in.}, \quad I = 1.591 \text{ in.}^4 \quad \text{and} \quad J = 3.182 \text{ in.}^4,$$

and inserting values in the equations for maximum tensile and shearing stresses,

$$t = \frac{0.585 \times 1200 \times 9 \times 1.5}{1.591} + \sqrt{\left(\frac{0.6 \times 1250 \times 13 \times 1.5}{3.182}\right)^2 + \frac{1}{4}\left(\frac{1.17 \times 1200 \times 9 \times 1.5}{1.591}\right)^2}$$
$$= 12{,}063 \text{ psi}$$

and

$$s = \sqrt{\left(\frac{0.6 \times 1250 \times 13 \times 1.5}{3.182}\right)^2 + \frac{1}{4}\left(\frac{1.17 \times 1200 \times 9 \times 1.5}{1.591}\right)^2}$$
$$= 6106 \text{ psi}.$$

Pressed steel housings usually are made of No. 1020 steel, which has an elastic limit in tension of about 40,000 psi, and an elastic limit in shear of about 25,000 psi. The factor of safety therefore would be a little more than 3 with respect to tension, and about 4 with respect to shear, both based on the elastic limit.

Calculation of Full-Floating Axles—Trucks and buses of large capacity, say over 15,000 lb gvw, do not have sufficient engine power to spin the rear wheels under full load on hard, dry pavement, and the maximum torque on each axle shaft should be calculated from the maximum engine torque, the transmission low-gear ratio, and the axle ratio. A loss of 10 per cent in the transmission and final drive can be figured with. The brakes also are not sufficiently powerful to lock the wheels under full load on hard, dry pavement. In fact, the maximum braking force is not likely to exceed 30 lb per 100 of gross vehicle weight, which corresponds to a deceleration of 9.66 ft per sec^2, and a stopping distance of 44.5 ft from an initial speed of 20 mph.

The shaft diameter required can be calculated in the same way as the minimum shaft diameter for a semi-floating axle. As already pointed out, the shaft is of constant diameter, except at the inner, splined end, where it is upset, and close to the flange at the outer end, where the diameter is increased gradually to avoid a sudden change in section and consequent concentration of stress. With the better grades of alloy steel used for axle shafts, a shearing stress of 45,000-50,000 psi can be figured with.

In the full-floating axle the axle housing itself transfers the load to the road wheels, and therefore acts as a beam freely supported near its ends and loaded at intermediate points—at the spring seats. While the axle housing itself represents distributed load, we will be justified in considering this also concentrated at the spring seats, as this simplifies the calculations, and the resulting error is insignificant. Let the maximum load on each wheel be represented by W, and the center distance between the wheel and the spring seat by l. Then the reaction of the ground on the wheel produces a bending moment Wl on the axle housing at the center of the spring seat. If the maximum propelling or retarding force is represented by aW, the combined bending moment due to the vehicle weight and the propelling or retarding force is $\sqrt{1 + a^2}W$. This moment results in a bending stress

$$S_t = \frac{\sqrt{1 + a^2}Wlc}{I} \text{ psi.}$$

The maximum torque aWR produces a shearing stress

$$S_s = \frac{aWRc}{J} \text{ psi.}$$

If the housing is of annular form, the polar moment of inertia J is equal to $2I$, where I is the moment of inertia around a diameter of the annulus. By now substituting the above values of S_t and S_s in the formulae for combined stress and replacing J by $2I$, we get for the maximum tensile stress

$$t = \frac{\sqrt{1 + a^2}Wlc}{2I} + \sqrt{\left(\frac{aWRc}{2I}\right)^2 + \left(\frac{\sqrt{1 + a^2}Wlc}{2I}\right)^2}$$

and for the maximum shear

$$s = \sqrt{\left(\frac{aWRc}{2I}\right)^2 + \left(\frac{\sqrt{1 + a^2}Wlc}{2I}\right)^2}.$$

In these equations we can replace c/I by $1/Z$, Z being the section modulus, and we can then take $W/2Z$ out of the individual terms, as it appears in all of them. This gives

$$t = \frac{W}{2Z}\left[\sqrt{1 + a^2}\,l + \sqrt{a^2R^2 + (1 + a^2)l^2}\right] \text{ psi,}$$

and

$$s = \frac{W}{2Z}\sqrt{a^2R^2 + (1 + a^2)l^2} \text{ psi.}$$

We will now apply the foregoing to the calculation of an axle housing for a truck of 16,000 lb gvw, equipped with an engine having a maximum torque of 200 lb-ft. We will assume that the total reduction ratio between engine crankshaft and drive wheels in low gear is 37.5, and that the wheels have an effective radius of 17 in. At full load approximately 75 per cent of the total load is on the driving wheels, hence each wheel then carries 6000 lb. Allowing 10 per cent for loss in the transmission, the maximum propelling effort is

$$\frac{200 \times 12 \times 37.5 \times 0.9}{17} = 4765 \text{ lb}$$

or roughly 30 per cent of the gross vehicle weight. Hence, a in the formula is equal to 0.3 and $\sqrt{1 + a^2} = 1.044$. Let the center distance l between wheel and spring seat be 13.5 in. Then, inserting values in the equations for t and s, we get

$$t = \frac{6000}{2Z}\left(1.044 \times 13.5 + \sqrt{0.3^2 \times 17^2 + 1.09 \times 13.5^2}\right)$$

which when simplified gives

$$t = \frac{87{,}300}{Z}.$$

Similarly we find that

$$s = \frac{45{,}000}{Z}.$$

As the elastic limit in shear of steels is at least 60 per cent of their elastic limit in tension, and in this case the shearing stress is only

$$\frac{45{,}000 \times 100}{87{,}300} = 51.5 \text{ per cent}$$

of the tensile stress, the latter determines the dimensions required.

If the axle housing is to be forged of 1035 steel, which when heat-treated and drawn at 1000 F has a tensile strength of about 90,000, and an elastic limit of 60,000 psi, a tensile stress of 20,000 psi can be figured with. We then have

$$Z = \frac{87{,}300}{20{,}000} = 4.365 \text{ in.}^3$$

Now assume that the outside diameter of the housing is set down as 4¼ in. Then, calling the inside diameter d, the section modulus

$$\frac{\pi}{32}\left(\frac{4\frac{1}{4}^4 - d^4}{4\frac{1}{4}}\right) = 4.365,$$

which when solved gives $d = 3.423$ in. If the inside diameter d is made 3⅜ in. the wall thickness will be 7/16 in. and the stresses will be slightly less than assumed.

Typical Truck-Axle Design—The wheel end and center housing of a modern full-floating truck axle (International Harvester Company) are shown in Figs. 20 and 21, respectively. The axle housing, which is of the Clark Equipment

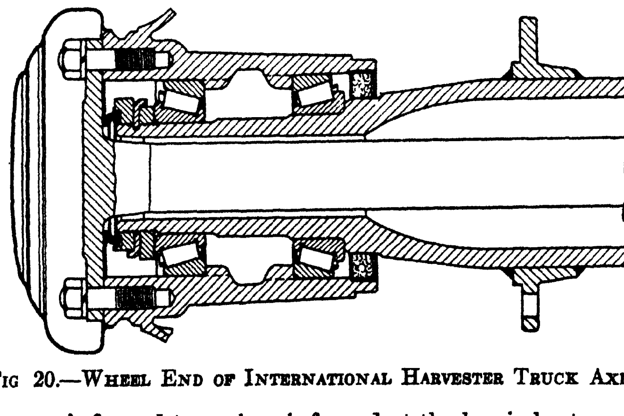


FIG 20.—WHEEL END OF INTERNATIONAL HARVESTER TRUCK AXLE.

Company's forged type, is reinforced at the banjo by two rings of ⅜-in. plate steel, which are welded to the housing at both their inner and outer edges. The differential gear is mounted in the carrier unsymmetrically, the bearing back of the bevel gear being much closer to the center plane of the differential than the other one. This arrangement undoubtedly was adopted because it divides the bevel-gear tooth load more nearly equally between the two bearings than a symmetrical design would. The spiral-bevel pinion is straddle-mounted, the two taper roller bearings back of it being seated in a flanged sleeve, and pinion adjustment is made by means of shims under the flange. An integral driving flange is formed on the outer end of each axle shaft, and the shaft is rigidly secured to the wheel hub by means of cap screws. The flange for the brake backing plate is welded to the axle housing.

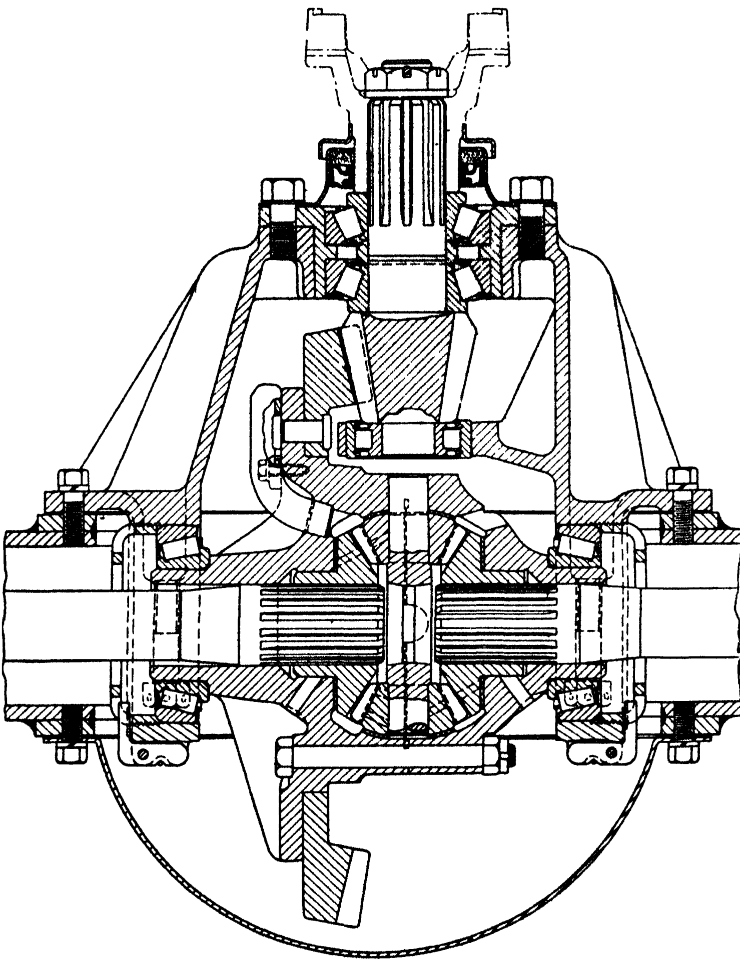


FIG. 21.—CENTER PORTION OF INTERNATIONAL HARVESTER TRUCK AXLE.

Trussed Gear Carrier—Where trouble has been experienced with rear-axle drive gears, it usually has been possible to trace it to excessive deflections of the gears under extreme load, which results in concentration of load at one end of the gear teeth. To keep down the deflections, the mounting of the gears must be made very rigid, and one method of increasing the rigidity without adding materially to the weight and bulk of the carrier consists in supporting it at both the

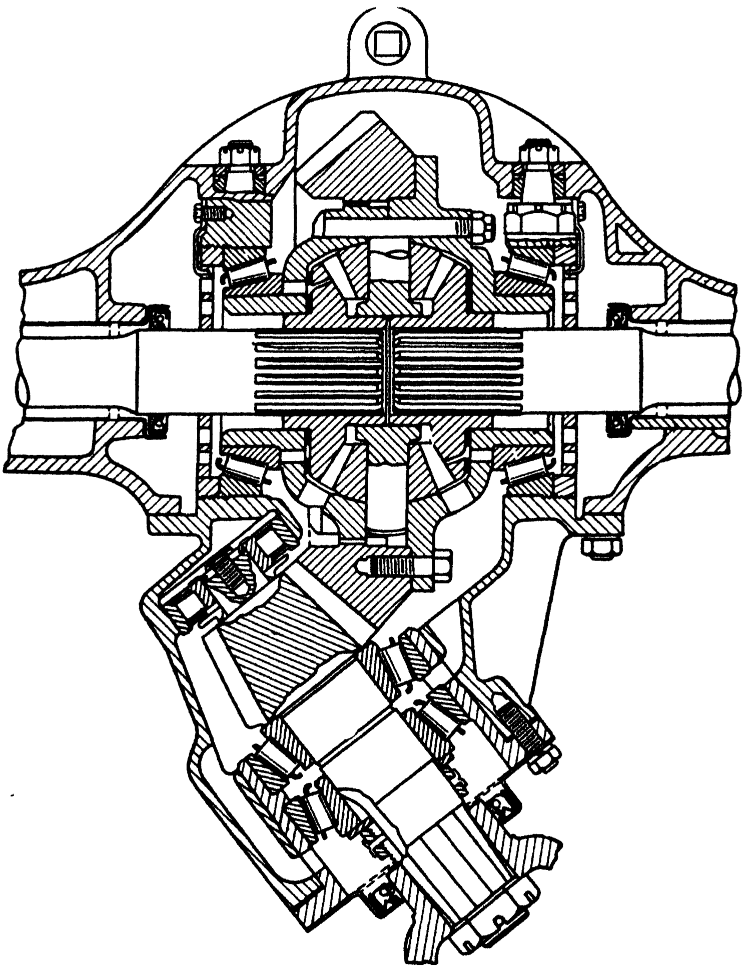


Fig. 22.—Center Section of Bus Axle with Angular Drive and Trussed Differential Carrier (Yellow Coach).

front and the rear of the center housing. This is illustrated in Fig. 22, which shows the center housing of a Yellow Coach rear axle in section. The studs which hold the bearing caps of the gear carrier in place have tapered extensions which pass through holes drilled in the back wall of the housing.

CHAPTER IX

Spiral Bevel- and Hypoid-Gear Drives

Most motor vehicles have the powerplant mounted longitudinally on the frame and require a right-angled drive to the axle. In the earliest shaft-driven vehicles the drive was by ordinary bevel gears with straight teeth. As engine speeds and car speeds increased, these gears proved rather noisy, and in 1913 spiral bevel gears (Fig. 1) were introduced to ensure more nearly noiseless operation. The machinery required to cut the gears was developed by the Gleason Works of Rochester, N. Y., and the first automobile manufacturer to install them was the Packard Motor Car Company of Detroit.

Spiral bevel gears with axes at right angles to each other bear the same relation to straight bevel gears as helical gears with parallel axes to straight spur gears. Their chief advantage is their comparatively noiseless operation at all speeds, which is due to the fact that each pair of teeth, instead of

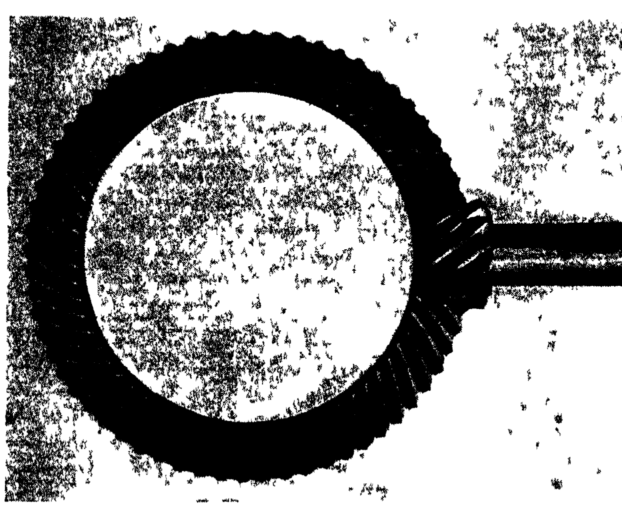


FIG. 1.—SPIRAL BEVEL GEAR AND PINION.

in connection with other types of gearing, was adopted. The addendum of the pinion is made much greater than the dedendum, and the dedendum of the gear correspondingly greater than its addendum. The sum of addendum and dedendum constitutes the whole depth, and in the Gleason system this is made smaller—for a given pitch—the smaller the number of pinion teeth. The difference between the whole depth and the working depth is the clearance, which is independent of the tooth numbers; it is made equal to 6 per cent the circular pitch for passenger-car gears, and 5 per cent the circular pitch for truck gears.

Whereas in conventional bevel gears all elements of the tooth converge toward the cone center, in the Gleason system the face elements of one member of a pair are made parallel with the root elements of the mating member, so that the clearance is constant over the face width. Tooth proportions of Gleason gears for a diametral pitch of 1 are given in Table I. For any other pitch, all linear dimensions given

Table I—Spiral Bevel Tooth Proportions

1 D.P.

Passenger Cars—Generated or Formate

No. of teeth in pinion	8	9	10	11	12	13
Working depth, in....	1.550	1.570	1.600	1.640	1.700	1.700
Whole depth, in......	1.738	1.758	1.788	1.828	1.888	1.888
Gear addendum, in...	0.235	0.240	0.250	0.270	0.310	0.370
Circular thickness, in.	From tooth layout for equal fatigue life.					
Pressure angle, deg..	16	16	16	16	16	16
Spiral angle, deg.....	35–45		35–40			

Automotive Truck—Generated or Formate

No. of teeth in pinion	6	7	8	9
Working depth, in....	1.500	1.550	1.620	1.700
Whole depth, in......	1.657	1.707	1.777	1.857
Gear addendum, in...	0.230	0.280	0.340	0.410
Circular thickness, in.	From tooth layout for equal fatigue life.			
Pressure angle, deg..	20	20	20	20
Spiral angle, deg.....	35–45			

in the table must be divided by the particular diametral pitch.

In conventional gears the circular thickness of the tooth is the same for the pinion as for the gear, each being equal to one-half the circular pitch. When the pinion has relatively few and the gear a large number of teeth, this makes the pinion teeth consideraby thinner at the root than the gear teeth, and therefore weaker. In Gleason gears the circular thicknesses of pinion and gear are so chosen that the fatigue lives of both are substantially equal. The proper circular thickness for the pinion tooth is found by means of a formula which includes an item that is dependent on the tooth numbers of both the pinion and gear, and which is obtained from a table.

Tooth Curvature—In straight bevel gears the tooth elements of both pinion and gear meet at the cone center, and this ensures contact over the entire face width, provided the axes of both pinion and gear are in the same plane. To obtain the same result with spiral bevel gears, the radii of curvature of the mating surfaces on pinion and gear would have to be equal. However, in actual service it is impossible to keep the axes of both members of a pair in the same plane, as the housing will distort and the shafts will flex under load, and if the teeth made contact over their entire face width under light load, there would be load concentration, and consequent stress concentration, at one end under heavy load. To prevent such stress concentration in spiral bevel gears, the radius of curvature of the concave side of the tooth is made slightly greater than that of the convex side. Contact then does not extend over the entire face width, but only a certain distance to both sides of the point of tangency of the circular elements, the actual width of contact increasing with the load. Where the pinion overhangs its bearings, the teeth are so cut that under light load the center of tooth contact will be close to the small end. Then, as the load increases, it shifts toward the opposite end, and at full load it is near the middle, so that the load is well distributed over the whole tooth. The more rigid the mounting, the more nearly equal the radii of curvature of both tooth surfaces can be made, and the better the load distribution will be.

Spiral Angles—The spiral angle, that is, the angle between a pitch-cone element and an intersecting tooth element, is so chosen as to give a certain pitch overlap, and it therefore varies inversely as the number of teeth in the pinion. Table II gives the spiral angles used in spiral bevel gears for both passenger cars and trucks.

Gear Materials—Rear-axle gears, as a rule, are made of a carburizing grade of steel, and case-hardened. Among the steels used are the 3½-per cent nickel steels 2315 and 2320, the 5-per cent nickel steels 2512 and 2515, the chrome-nickel steels 3115 and 3312, and the nickel-molybdenum steels 4615, 4620 and 4820. According to Almen and Boegehold of General Motors Research Laboratories, who made numerous tests on rear-axle assemblies, in rear-axle gears the properties of alloy steels as determined by the usual laboratory tests are

Table II—Spiral Angles for Spiral Bevel Gears

PASSENGER CARS

No. of pinion teeth..	8 and 9	10 and over
Spiral angle, deg.	40–45	40

TRUCKS

No. of pinion teeth..	5	6	7	8	9 and over
Spiral angle, deg ...	42–45	40–42	38–40	35–38	33–35

not fully realized, and steels for such gears should be selected primarily on the bases of warping tendencies, machining qualities, and cost, rather than on the basis of mechanical properties. In the finished gears these properties are obscured by the much greater effects of stress concentrations having their origins in tooth shape, machining scratches, gear deflections, eccentric mountings, warping during heat treatment, etc. Of course, the composition of the steel and the heat treatment have some effect on performance, and of six different steels in axle gears fatigue-tested, No. 2515 showed the highest and No. 4615 the lowest fatigue strength.

Production of Gear Blanks—Blanks for bevel pinions with their integral shafts are forged from bar stock, the pinion itself being formed by upsetting. As a rule, three passes are required to upset the blank for the pinion and to ensure satisfactory grain flow. Blanks for pinions designed to be straddle-mounted are hammered down at both ends. Gear blanks also can be conveniently produced from round bar stock. Discs cut from such stock are hammered flat, and the gear ring is then formed with a die in a hammer. The ring is forged with a web with a central "tong hold" on it, by means of which it can be manipulated, this web and the flash on the circumference being later removed with a trimming die. Rings forged in this manner can be rolled to near finish size, and they will then have uniform, substantially radial grain flow, which tends to give stronger teeth.

Heat Treatment—Gears and pinions, after being completely machined, are carburized either packed in boxes containing granular carburizing or pack-hardening material, or in gas-carburizing furnaces. Carburizing is effected at temperatures between 1675 F and 1700 F. The pinions usually are quenched in oil as they are taken from the carburizing box. Gears also may be quenched directly from the box, but in some plants they are allowed to cool in the box, are reheated to somewhere between 1400 F and 1475 F, and quenched in oil. Gears are always quenched in quenching presses, in which they are held in dies, to minimize distortion. If gears are quenched directly from the box, they must be cleaned of carburizing material with a wire brush. When pinions and gears are gas-carburized, they are passed through furnaces containing an atmosphere of prepared gas mixtures, which give up carbon to the red-hot metal surfaces. Gas-carburized gears and pinions also are quenched directly from the furnace. Sometimes gears are drawn or tempered after being case-hardened, and the drawing temperature then must be held sufficiently low (275-300 F) so the tooth surfaces after the draw will withstand the file test, which is made with a Nicholson "XF" file or its equivalent. Many plants now dispense with the drawing operation.

Strength of Spiral Bevel Gears—The results of fatigue tests usually are presented in the form of "*S-N* curves" in which the number *N* of stress cycles to failure is plotted against the stress *S*—the maximum value of the stress cycle. When fatigue-test data are thus plotted on so-called log-log section paper, on which the scales on both axes are divided logarithmetically, they yield a straight line. Almen and Boegehold plotted the results of their fatigue tests on rear-axle assemblies on such section paper. In the case of gear teeth, of course, the maximum stress cannot be calculated with the same degree of accuracy as in that of a simple test specimen, such as a rotating beam of circular section, and if the stress data are incorrect, the plotted "observation points," instead of falling on a straight line, will scatter. Almen and Boegehold applied different gear-tooth beam-strength formulae to their data, and found that the least scattering occurred when tooth stresses were calculated by a method first described by F. E. McMullen and T. M. Durkan in a paper read before the American Gear Manufacturers Association in 1922. This method takes account of the fact that a gear tooth is most heavily loaded at the moment when the tooth next in order is about to come into mesh, as it then carries the entire tooth load. This is illustrated in Fig. 3 which shows a section of a 10-tooth spiral bevel pinion in mesh with a 47-tooth gear. The pinion tooth at the left is just coming into contact with a gear tooth, and the one at the right therefore carries the entire load. This is the highest point on the pinion tooth profile at which the entire load is carried by a single tooth, and with the pinion and gear in the relative position shown, the pinion tooth stress is a maximum. The tooth pressure acts along the inclined line passing through point *A*. Designating the height *AD* (Fig. 3) by *h* and the root thickness *BC* by *t*, the maximum stress in the tooth due to the imposed bending moment is given by the equation

$$S = \frac{6hF}{t^2W'} \text{ psi},$$

where F is the tangential force acting at point A under maximum low-gear torque, and W' the equivalent face width, that is, the face width of a spur pinion of the same diametral pitch and number of teeth, and having the same strength. The equivalent face width W' is smaller than the actual face width W, firstly because the tooth section becomes weaker [the value of the expression $(t^2/6h)$ becomes smaller] as the small end of the

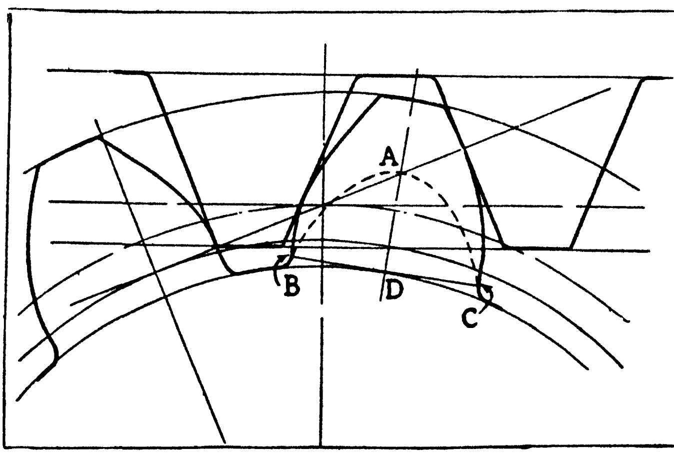


FIG. 3.—TEN-TOOTH SPIRAL BEVEL PINION MESHING WITH 47-TOOTH GEAR

pinion is approached, and, secondly, because at the same time the tangential force corresponding to a given torque becomes greater, due to the decrease of the radial distance of the point of contact from the axis. If we base our calculations on the tooth section at the large end and the tangential force at point A there, then it follows from the expression for the ideal center of load distribution in a bevel pinion given on page 193, that

$$W' = W[1 + (n^2/2) - n],$$

where n is the ratio of the face width to the cone distance. Inserting this value of W' in the equation for the tooth stress we have

$$S = \frac{6h}{t^2} \times \frac{F}{W[1 + (n^2/2) - n]} \text{ psi}.$$

Allowable Stress in Gear Teeth—Calculating the stress in the gear teeth by the McMullen and Durkan method, Almen and Boegehold found that they could represent the results of their fatigue tests on rear-axle assemblies by the S-N graph Fig. 4, which corresponds substantially to the equation

$$\log N = 39.86 - 7.54 \log S.$$

By checking with service results obtained with axles of the same design as those used in the tests, they arrived at the conclusion that 100,000 stress cycles in the fatigue-testing machine is substantially equal to normal service for the life

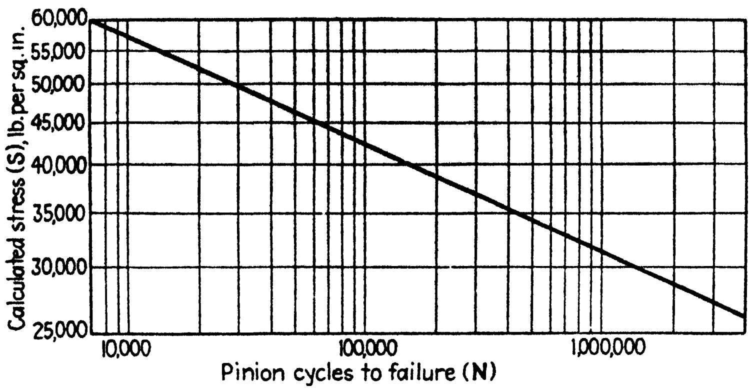


Fig. 4.—Fatigue-Life Chart for Spiral Bevel Axle Gears.

of the car, one cycle in the machine causing about as much fatigue as 1000 pinion stress cycles in normal road operation. If we insert 100,000 for N in the above equation, we find that the tooth stress which gives this fatigue life is about 42,000 psi, and Fig. 4 confirms this result. This Almen and Boegehold regard as the maximum permissible stress in passenger-car rear-axle gears, and they say that ordinarily a somewhat lower stress, 37,000 psi, is allowed, which corresponds to a fatigue life of 250,000 stress cycles:

$$S = \text{antilog} \frac{39.86 - \log 250{,}000}{7.54} = 37{,}000 \text{ psi (appr.)}.$$

Fig. 5 gives the standard bevel-gear nomenclature and symbols.

Sample Calculation—We will now calculate a spiral-bevel gear set for a truck equipped with an engine developing a maximum torque of 272 lb-ft, and with a transmission having a low-gear ratio of 7.42. This makes the maximum pinion torque 24,216 lb-in. Let us assume that a reduction ratio of approximately 6.5 is needed to give the required acceleration and hill-climbing ability. A combination with seven teeth in the pinion and 45 in the gear will then serve. Road clearance requirements put a limit on the pitch which can be used. A gear pitch diameter of 15 in. is usually about the largest permissible, and a diametral pitch of 3 will hold the diameter

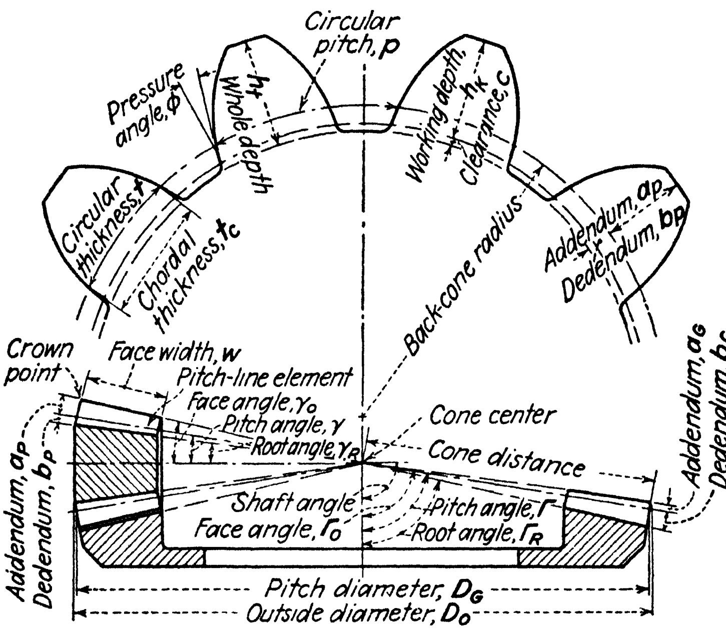


FIG. 5.—NOMENCLATURE AND SYMBOLS FOR BEVEL GEARS.

to this limit. With this pitch the pitch radius of the gear will be 7.5 in., that of the pinion, 1.1667 in., and the cone distance,

$$\sqrt{7.50^2 + 1.1667^2} = 7.5902 \text{ in.}$$

Circular Thickness—As already mentioned, whereas in conventional gearing the circular thicknesses of pinion and gear are made equal, in Gleason spiral bevel gears they are so proportioned that the fatigue lives, expressed in pinion stress cycles or in hours of operation, are approximately the same for both pinion and gear. In terms of tooth-stress cycles the fatigue life of the gear is then less than that of the pinion in the proportion of the tooth numbers. In our example, if we assume the pinion to be designed for a fatigue life of 100,000 cycles, which according to the fatigue chart corresponds to a maximum stress of about 42,000 psi, the fatigue life of the gear can be made

$$(7/45) \times 100{,}000 = 15{,}500 \text{ cycles},$$

which corresponds to a stress of 54,000 psi. As the stress can be higher in the gear tooth than in the pinion tooth, the beam strength of the former need not be as great as that of the latter, hence the root thickness of the gear tooth can be less than that of the pinion tooth. As in the Gleason system the pitch line is near the tip of the gear tooth and there is an included angle between tooth flanks of twice the pressure angle, the circular thickness of the gear tooth is further reduced. The proper division of the circular pitch between the circular thicknesses of pinion and gear is arrived at to a certain extent by trial and error, by applying the stress formula already given, but with a little experience it is possible to arrive at the required proportion between the two almost immediately. In the present case it is found that the circular thickness of the pinion must be 0.7467 in., and that of the gear, 0.3005 in., the circular pitch being 1.0472 in.

Bending Stresses—A layout of the gears (Fig. 6) shows that when there is contact at the highest point at which one tooth takes the whole load, the height h of the inscribed parabola is 0.496 in. The root thickness of the pinion tooth is 0.754 in., and the radius of the point of load application (the apex of the parabola), 1.494 in. The tangential load on the pinion therefore is

$$\frac{24{,}216}{1.494} = 16{,}210 \text{ lb.}$$

In calculating the stress for purposes of fatigue-life determination, a correction factor is applied to take account of the methods of pinion and gear mounting. If either the pinion or gear overhangs its bearings, this factor is 1.00, whereas when both pinion and gear are straddle-mounted, it is 0.80. This latter type of mounting is generally used for truck drives, and we can therefore use the "mounting factor" 0.80 in the stress formula. Allowing a pinion fatigue stress of 42,000 psi and inserting known values in the formula, we have

$$42{,}000 = \frac{0.80 \times 6 \times 0.496}{0.754^2} \times \frac{16{,}210}{W[1 + (n^2/2) - n]}$$

from which it follows that

$$W[1 + (n^2/2) - n] = 1.616.$$

Now, n is equal to $W/7.5902$, and substituting this value in the above equation we get

$$W + W^3/115.2 - W^2/7.5902 = 1.616,$$

which when solved gives $W = 2.13$ in. The face width therefore can be made 2.125 in.

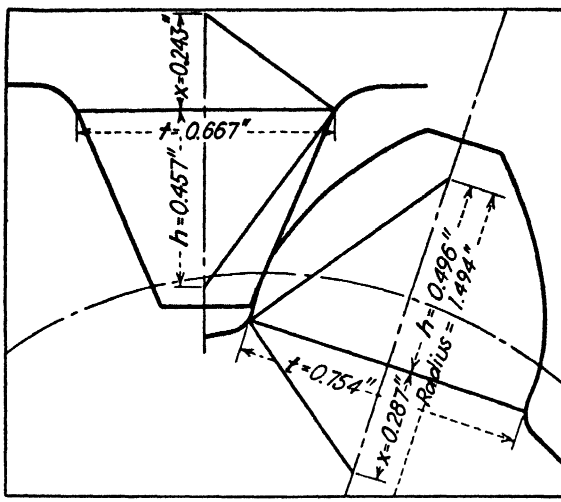


FIG. 6.—TOOTH LAYOUT FOR SPIRAL BEVEL TRUCK GEARS CALCULATED.

Pinion and Gear Dimensions—Methods of calculating the dimensions of the pinion and gear are illustrated by the following table, where they are applied to the combination which was calculated for strength in the foregoing:

Number of pinion teeth, $n = 7$
Number of gear teeth, $N = 45$
Diametral pitch, $P = 3$
Circular pitch, $p = 3.1416/3 = 1.0472$
Face width, $W = 2.125$ in.
Working depth, h_k (from Table I) $= 1.550/3 = 0.517$ in.
Clearance, $c = 0.05 \times p = 0.052$ in.
Whole depth, $h_t = h_k + c = 0.569$ in.
Pressure angle, ϕ (from Table I) $= 20°$
Spiral angle ψ_p (from Table I) $= 37°$
Cone distance, $A_o = \sqrt{D^2 + d^2}/2 = 7.5902$ in.

Pinion

Pitch diameter, $d = n/P = 2.3333$ in.
Addendum, $a_p = h_k - a_G = 0.424$ in.
Dedendum, $b_p = h_t - a_p = 0.145$ in.
Pitch angle, γ ($\tan \gamma = n/N$) $= 8° 51'$
Dedendum angle, δ_p ($\tan \delta_p = b_p/A_o$) $= 1° 6'$
Face angle $(90° - \Gamma_R) = 12° 26'$
Root angle, $\delta_R = \gamma - \delta_p = 7° 45'$
Outside diameter, $d_o = d + 2a_p \cos \gamma = 3.171$ in.
Cone center to crown $= D/2 - a_p \sin \gamma = 7.435$ in.
Circular thickness, $t = p - T = 0.7467$ in.

Gear

Pitch diameter, $D = N/P = 15.000$ in.
Addendum, $a_G = 0.280/P = 0.093$ in.
Dedendum, $b_G = h_t - a_G = 0.476$ in.
Pitch angle, $\Gamma = 90° - \gamma = 81° 9'$
Dedendum angle, δ_G ($\tan \delta_G = b_G/A_o$) $= 3° 35'$
Face angle $= 90° - \gamma_R = 82° 15'$
Root angle $\Gamma_R = \Gamma - \delta_G = 77° 34'$
Outside diameter, $D_o = D + 2a_G \cos \Gamma = 15.029$ in.
Cone center to crown $= d/2 - a_G \sin \Gamma = 1.075$ in.
Circular thickness, $T = 0.3005$ in.

Direction of Thrust Loads—Whenever the tooth pressure is at an angle to a plane normal to the pinion (or gear) axis, it has a component parallel with the axis, and creates thrust. In a spiral bevel gear the tooth pressure is inclined to the normal plane by an angle which depends on the spiral angle, the pressure angle, and the pitch angle. When a car is being driven forward, the pinion turns right-handedly (looked at from the front), and with a left-hand spiral bevel pinion the spiral angle creates end thrust which tends to force the pinion away from the cone center. End thrust due to the pressure angle is in the same direction, while the pitch angle has a modifying effect on the thrusts produced by the other two. Thus under the conditions mentioned, the end thrusts due to the spiral angle and pressure angle combine. When the car is being backed, the end thrust due to the spiral angle reverses in direction, while that due to the pressure angle does not. The former predominates, and the actual end thrust in reverse therefore is opposite in direction, but smaller, as compared with that in forward gear.

With a right-hand pinion the end thrust in forward gear is of the same magnitude as with a left-hand pinion in reverse, and is also in the direction toward the cone center, while in reverse it corresponds in magnitude and direction to that with a left-hand pinion in forward gear. As the car is being driven forward most of the time, and end thrust in forward drive is least with a right-hand pinion, the latter would seem to have an advantage. But with it the thrust is toward the cone center, that is, the pinion tends to be drawn into the gear, and it is easier to provide for thrust in the opposite direction, for which reason left-hand spiral pinions are generally used in automotive drives.

Bearing Loads with Spiral Bevel Gears—We found in the chapter on The Differential Gear that the ideal center of load distribution in a bevel gear is located at a distance mL from

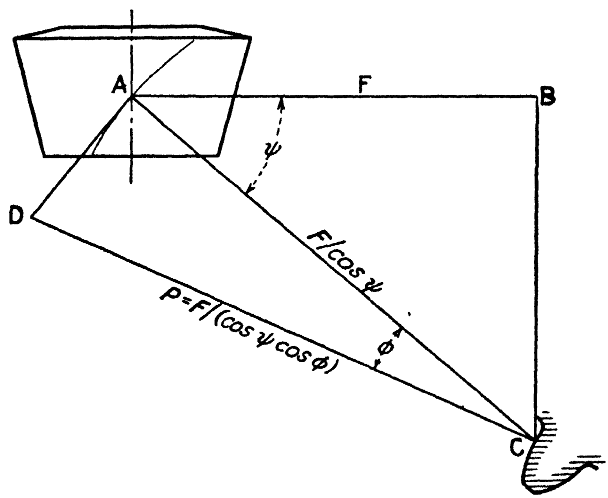


FIG. 7.—DIAGRAM SHOWING RELATION BETWEEN TANGENTIAL FORCE AND NORMAL TOOTH PRESSURE.

the apex of the pitch cone, where L is the cone distance and m is given by the equation

$$m = \sqrt{1 + \frac{n^2}{2}} - n,$$

n being the proportion of the gear face to the cone distance. A tabulation of values of m for different values of n within the usual range was given on page 193 in that chapter.

In determining the bearing loads we have to first determine the tangential force or tooth load on the pinion and its location. The mean effective pitch diameter d_p of the pinion is found by multiplying the maximum or nominal pitch diameter by the value of m. We then get for the tangential force

$$F = \frac{24T}{d_p}.$$

The normal load on the teeth is considerably greater than the tangential force, owing to the inclination of the tooth elements to the pitch-cone elements (angle of spiral) on the one hand, and to the inclination of the tooth flank (pressure angle) on the other. If we designate the angle of spiral by ψ and the normal pressure angle of the tooth by ϕ, the normal pressure on the tooth is

$$P = \frac{F}{\cos\psi\cos\phi},$$

as may be readily seen from Fig. 7 in which AB represents the tangential force on the pinion, AC the force in a plane tangent

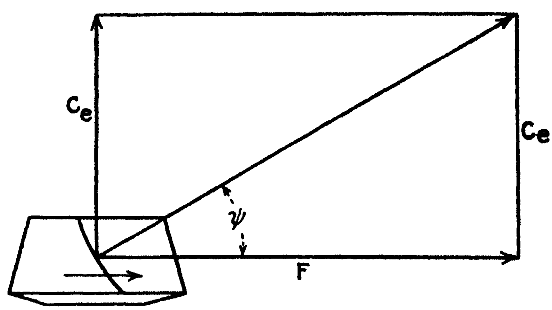


FIG. 8.—DIAGRAM SHOWING RELATION BETWEEN TANGENTIAL FORCE AND COMPONENT ALONG PITCH-CONE ELEMENT.

to the pitch cone and perpendicular to the tooth element, and CD the force normal to the contact surfaces. The problem now is to resolve the pressure P into a component parallel to the pinion axis (thrust load) and another component perpendicular to the pinion axis (radial load).

We first find the components of the tooth pressure along an element of the pitch cone and perpendicular to that element, respectively. In Fig. 8 is shown a plan view of a right-hand pinion supposed to rotate right-handedly. The horizontal arrow represents the tangential force F on the inclined tooth and the vertical arrow the resulting pressure along the pitch cone element. It will be seen that

$$\frac{C_e}{F} = \tan\psi,$$

hence

$$C_e = F \tan \psi.$$

This pressure along the pitch cone element can again be resolved into two components, as shown in Fig. 9, one parallel to the pinion axis and the other perpendicular thereto. It is here necessary to take account of the direction of the forces and we will call axial forces in the direction away from the cone center positive, and those in the opposite direction, negative. Radial forces from the point of contact toward the axis will be called positive, and those oppositely directed, negative.

Resolving the force along the pitch-cone element we get for the axial component

$$C_a = -F \tan \psi \cos \gamma,$$

and for the radial component

$$C_r = F \tan \psi \sin \gamma.$$

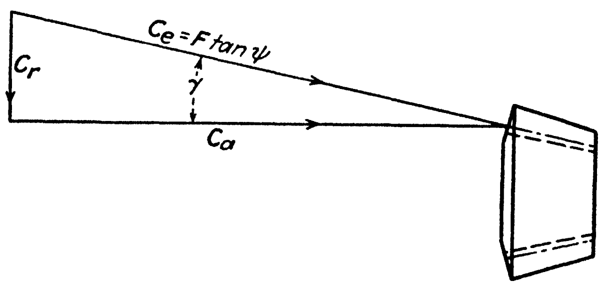


Fig. 9.—Force Along Pitch-Cone Element Resolved into Axial and Radial Components.

We next take up the component of the normal tooth pressure perpendicular to the pitch-cone element through the point of contact. From Fig. 10 it can be seen that this component C_p is equal to

$$P \sin \phi = \frac{F \sin \phi}{\cos \psi \cos \phi} = F \frac{\tan \phi}{\cos \psi}.$$

This component perpendicular to the pitch-cone element also may be further resolved into axial and radial components, as illustrated in Fig. 11, the axial component being

$$C_{aa} = F \frac{\tan \phi \sin \gamma}{\cos \psi},$$

and the radial component

$$C_{rr} = F \frac{\tan \phi \cos \gamma}{\cos \psi}.$$

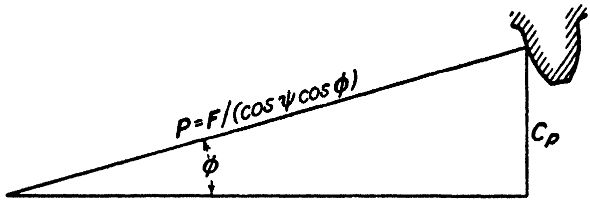


FIG. 10.—COMPONENT OF NORMAL TOOTH PRESSURE NORMAL TO PITCH-CONE ELEMENT.

We now add like components of the forces along the pitch-cone element and perpendicular to that element, respectively, and as tan = sin/cos, we can simplify the resulting equation by taking $1/\cos\psi$ out of the parentheses, which gives for the end thrust on a right-hand spiral bevel pinion turning right-handedly forward drive)

$$L_a = \frac{F}{\cos\psi}(\tan\phi\sin\gamma - \sin\psi\cos\gamma)$$

and for the radial load on such a pinion

$$L_r = \frac{F}{\cos\psi}(\tan\phi\cos\gamma + \sin\psi\sin\gamma).$$

These same equations apply to the case of a left-hand pinion turning left-handedly (reverse motion). If a left-hand pinion turns right-handedly (forward motion), the component along the pitch-cone element is in the direction away from the cone center, and the sign of its axial component becomes positive. The same applies to a right-hand pinion turning left-handedly. However, the radial component of the force acting along the pitch-cone element in this case is directed from the axis through the point of contact, and is therefore negative. We therefore have for the axial and radial forces on a left-hand spiral bevel pinion turning right-handedly (forward drive) or a right-hand spiral bevel pinion turning left-handedly (reverse drive):

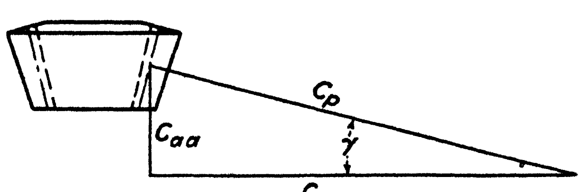


FIG. 11.—FORCE NORMAL TO PITCH-CONE ELEMENT RESOLVED INTO AXIAL AND RADIAL COMPONENTS.

$$L_a = \frac{F}{\cos \psi} (\tan \phi \sin \gamma + \sin \psi \cos \gamma)$$

and

$$L_r = \frac{F}{\cos \psi} (\tan \phi \cos \gamma - \sin \psi \sin \gamma).$$

Axial and Radial Loads on Gear—As action and reaction are equal and opposite, the axial load or end thrust of the gear is equal and opposite in direction to the radial load on the pinion, and the radial load on the gear is equal and opposite in direction to the axial load on the pinion. There is therefore no need to calculate the bearing loads of the gear separately. Of the above four equations the first two apply to assemblies comprising either a right-hand pinion turning right-handedly or a left-hand pinion turning left-handedly. For such assemblies the first equation gives the end thrust of the pinion and the radial load of the gear, while the second gives the radial load on the pinion and the end thrust of the gear. The third and fourth equations apply to assemblies comprising either a right-hand pinion turning left-handedly or a left-hand pinion turning right-handedly, the third giving the end thrust on the pinion and the radial load on the gear, and the fourth the radial load on the pinion and the end thrust of the gear. It may be recalled that in these equations F is the tangential force at the mean effective pitch radius of the pinion, ψ the spiral angle, ϕ the normal pressure angle, and γ the pitch angle *of the pinion.*

Hypoid Gears—In hypoid gears, as already explained, the pinion axis is offset with relation to the gear axis. It can be either above or below the gear axis, but in automobile drives, where the object is to lower the propeller shaft, it is always below. The direction of offset determines the hand of spiral, and when the pinion is below the gear axis it should have left-hand spiral teeth. With such teeth and right-hand rotation (as in automobile drives), the end thrust on the pinion during forward drive tends to force it away from the gear center, and end thrust in that direction can be readily provided for in the mounting.

An important item in the design of hypoid gears is the offset. It determines the extent to which the propeller shaft can be lowered, and it causes sliding motion at the tooth-contact surfaces in the direction of tooth length, which sliding motion increases with the offset. To permit of this endwise sliding motion, the teeth must have a special form in the lengthwise direction, and this imposes restrictions on the pitch lines that do not apply in the case of ordinary bevel gears. In passenger-car drives the offset usually is made nearly 1/5 the gear diameter (which is considered the practical limit), to be able to drop the propeller shaft as much as possible. In truck drives, on the other hand, where gear strength is a primary consideration, the offset usually is limited to about 1/8 the gear diameter.

Sliding Motion Along Teeth—In hypoid gears the motion of any contact point on either pinion or gear can be resolved into two components, one along the tooth element through that point, the other along a line normal thereto. Usually sliding motion along the pitch element is referred to, and in that case the pitch-line motion is the item resolved. Owing to the fact that the spiral angles of pinion and gear are different, their pitch-line velocities also differ. The normal-velocity com-

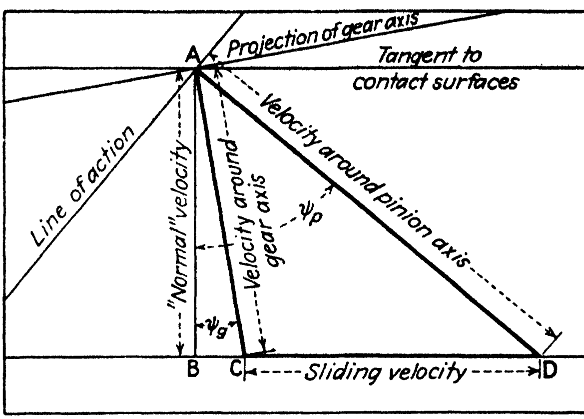


Fig. 12.—Diagram of Sliding Velocity Along Teeth in Hypoid Gears.

ponent is the same for both pinion and gear, because the teeth of both are in contact, and there can be no relative motion normal to the line of contact; but the components along the tooth elements are different.

In Fig. 12, A represents a point of contact between a pinion tooth and a gear tooth, and the horizontal line through this point is tangent to tooth elements of both pinion and gear. The normal velocity is there represented by AB, the gear pitch-line velocity by AC, and the pinion pitch-line velocity by AD. It will be seen that the "normal velocity" deviates from the pitch-line velocities by the respective spiral angles ψ_g and ψ_p. The component of the pinion pitch-line velocity along the pitch element is BD, and that of the gear pitch-line velocity, BC. Hence there is relative motion (sliding motion) of velocity CD between the contact points on the gear and the pinion.

From the diagram it follows that

$$CD/AB = \tan \psi_p - \tan \psi_g$$

and since $AB = AC \cos \psi_g$

$$CD = AC \cos \psi_g (\tan \psi_p - \tan \psi_g).$$

Average values of ψ_g and ψ_p for automotive type hypoid gears are 10° and 50°, respectively, so that on the average

$$CD = AC \times 0.985(1.192 - 0.176) = 1.0076AC.$$

Thus the velocity of longitudinal sliding motion at any contact point is practically equal to the velocity of that point around the pinion axis. The above expression for the longitudinal sliding velocity in hypoid gears was first given in an S.A.E. paper in 1926 by A. L. Stewart and Ernest Wildhaber.

Profile Sliding Motion—In addition to sliding motion along the teeth there is—as in all spur and bevel gears—sliding motion in a direction transverse to the tooth elements, commonly referred to as profile sliding. The velocity of this sliding motion varies throughout the period of action. When there is contact on the pitchlines it is nil, because at that moment the points of contact on pinion and gear move in the same direction, along the tangent to both pitch circles at the pitch point, and at the same velocity—the pitch-line velocity. At any other point of the line of action the contact points on pinion and gear move in different directions, and also at different velocities. The velocity of sliding motion varies linearly with the distance from the pitch point, and it therefore suffices to determine it for any one point of the line of action other than the pitch point to obtain values for the whole period of action. It reaches its maximum value at the end of the period of action, and we will determine its magnitude for that point, in terms of the pitch-line velocity.

Fig. 13 corresponds to the pair of spiral bevel gears calculated in the foregoing (7 and 45 teeth, 3 diametral pitch). That part of the diagram to the right of the vertical center line is a section through the pinion in a plane normal to the axis and through the point of contact B at the end of the line of action GB, which we will call the pinion normal plane. Contact point B on the pinion moves in the direction of a tangent at that point to a circle around the pinion axis, and its velocity BC is greater than the pitch-line velocity in the same proportion as OB is greater than the pitch radius OA.

$$BC = 1.5865v/1.1667 = 1.360v.$$

Contact point B on the gear tooth is at the bottom of the working depth, and its velocity therefore is slightly less than pitch-line velocity. In this particular case the radius of point B on the gear tooth is 7.435 in., and as the pitch radius is 7.5 in., the velocity is $7.435/7.5 = 0.9913v$. We must also take account of the fact that this point does not move in the pinion normal plane but at a slight angle thereto. The angle between its direction of motion and the pinion normal plane is θ_r/r, where θ_r is the arc of recess, in this case 26° 17′, and r the gear ratio, 45/7, hence the angle of deviation figures out to 4° 5′. If point B on the gear moved in the pinion normal plane we would plot

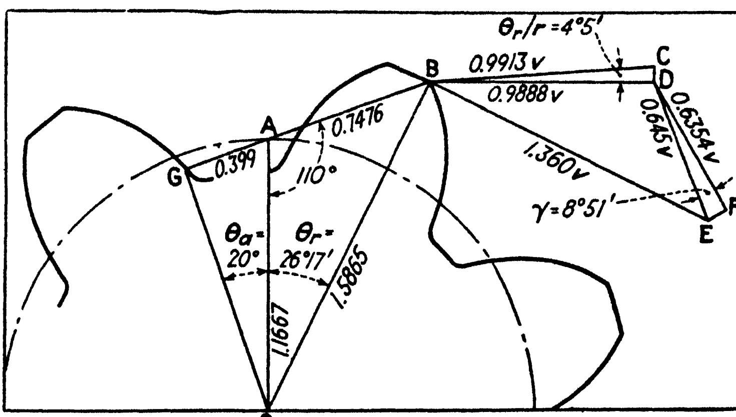


FIG. 13.—DIAGRAM OF PROFILE SLIDING VELOCITY IN SPIRAL BEVEL GEARS.

its velocity along a horizontal line extending to the right from it. As it is, we plot the velocity along a line making an angle of 4° 5′ with the horizontal, and then project the resulting vector BC on to the horizontal. This gives a velocity component of point B on the gear in the pinion normal plane (represented by BD) of $0.9888v$. If now we connect points D and E, the length of line DE represents the velocity of sliding motion along the outline of the tooth section in the pinion normal plane. We can determine this velocity by measurement on the diagram, but the result can be obtained with a greater degree of precision by trigonometric methods. This is possible because the lengths of two sides of the triangle BDE are known, as is also the included angle *EBD*, which is equal to the arc of recess, 26° 17′. We thus find that *DE* is equal to $0.645v$.

The value thus found is the sliding velocity along the tooth outline in the pinion normal plane, which makes an angle γ (= 8° 51′) with the tooth profile. The tooth profile is not as deep as the section in the pinion normal plane, and the sliding velocity along it is less in the proportion of unity to cos 8° 51′ (0.9881). Therefore, the sliding velocity along the tooth profile is

$$0.9881 \times 0.645v = 0.637v.$$

As the end of the line of action is 0.7476 in. from the pitch point, and the beginning of the same line, 0.399 in., the sliding velocity at the beginning of tooth action is

$$(0.399/0.7476) \times 0.637v = 0.340v.$$

The values thus arrived at graphically check quite well with results obtained when a method based on an analysis by Allan H. Candee of the Gleason Works is applied to the problem. Mr. Candee, in an Appendix to a paper on "Bevel Gears in Aircraft," *Transactions of the A.S.M.E.*, May, 1943, gave the following equation for the sliding motion in *spur gears* corresponding to an infinitesimal motion dx along the pitch circles:

$$dS = (1/R_1 + 1/R_2)s\,dx,$$

where R_1 and R_2 are the pitch radii, and s is the distance of the point of contact from the pitch point. In the case of bevel gears the back-cone distances must be substituted for the pitch radii, and the equation then becomes

$$dS = (\cos\gamma/R_1 + \cos\Gamma/R_2)s\,dx.$$

The sliding velocity is

$$v_s = dS/dt = (\cos\gamma/R_1 + \cos\Gamma/R_2)s\,dx/dt.$$

Now, dx/dt is the pitch-line velocity v, hence

$$v_s = (\cos\gamma/R_1 + \cos\Gamma/R_2)sv.$$

The following data apply to the gear set represented by Fig. 13:

$$\gamma = 8^\circ\,51'$$

$$\cos\gamma = 0.988$$

$$\Gamma = 81^\circ\,9'$$

$$\cos\Gamma = 0.154$$

$R_1 = 1.1667$ in.

$R_2 = 7.5$ in.

$s = 0.399$ at the beginning and 0.7476 at the end of the period of action.

We therefore have

$$v_s = (0.988/1.1667 + 0.154/7.5) \times 0.399v = 0.339v$$

for the sliding velocity at the beginning of tooth action, and

$$v_s = (0.988/1.1667 + 0.154/7.5) \times 0.7476v = 0.636v$$

at the end.

The above calculation applies to a pair of spiral bevel and not directly to hypoid gears, but there can be little doubt that the profile sliding velocity is substantially the same in hypoid as in spiral bevel gears of the same pitch and the same pinion pitch-line velocity. The longitudinal and profile sliding velocities in hypoid gears are at right angles to each other and can be readily combined by means of the parallelogram of motions.

Comparison with Spiral Bevels—An important difference between spiral bevel and hypoid gears is that whereas in the former the spiral angles of pinion and gear are equal, in hypoids the pinion has a much larger spiral angle than the gear. For proper mesh the normal circular pitch must be the same for both, and as for a given normal pitch the circular pitch is greater the greater the spiral angle ($p = p_n/\cos \psi$), it follows that for a given gear diameter and a given ratio the pinion pitch diameter is larger in the case of hypoids, usually by about 20 per cent. Moreover, since the working depth is made proportional to the normal pitch, that, too, is greater in hypoids, and hypoid gears have a materially greater "surface capacity" than spiral bevels of the same gear diameter. On the other hand, the greater sliding velocity in hypoid gears tends to reduce the ratio of load-carrying capacity to "surface capacity." The relative sliding velocities for spiral bevel and hypoid gears, respectively, over the entire arc of action, are shown in Fig. 14. The areas enclosed between the vertical axis and the two curves, respectively, evidently are measures of the total sliding motion during a period of action.

The two factors favoring the hypoid gear with respect to load capacity, as compared with spiral bevels, are the greater diameter of the pinion and the smaller spiral angle of the gear. With a spiral angle of 10° and a pressure angle of 20° the normal tooth load in hypoid gears is only 8 per cent greater

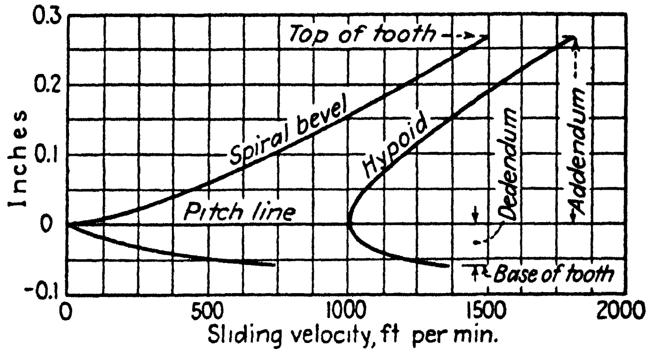


FIG. 14.—COMPARISON OF SLIDING VELOCITIES IN SPIRAL BEVEL AND HYPOID GEARS.

than the tangential force. Hypoid gears usually have a pressure angle of 20° on the driving side and 25° on the coasting side. When hypoid gears are substituted for spiral bevels in passenger cars, the gear diameter usually remains about the same. Gear pitch diameters of hypoid gears for passenger-car axles are about $0.56\sqrt[3]{T}$ in., where T is the maximum torque on the rear axle, in lb-in. A comparison between a number of passenger-car rear-axle gears of the spiral-bevel and hypoid types, respectively, with the same tooth numbers and the same gear diameters, showed that with hypoid gears the normal tooth load averaged about 12 per cent less, the tangential pressure about 16 per cent less, and the end thrust in forward drive, 8 per cent less.

Bearing Loads with Hypoid Gears—With hypoid gears the loads on the bearings of the pinion and the gear must be calculated separately, as the axes of pinion and gear do not intersect, and the thrust load on one therefore is not equal to the radial load on the other, as in the case of bevel gears. The pressure angle ϕ, of course, is the same for both pinion and gear, but the spiral angles and pitch angles are different. Spiral angles of pinion and gear will be designated by ψ_p and ψ_g respectively, and pitch angles by γ and Γ. In the following equations for axial and radial loads the plus sign denotes that the thrust is away from the cone center, while the minus sign denotes that it is toward the center. These equations refer to a combination comprising a left-hand pinion and right-hand gear, the pinion turning right-handedly, as well as to one comprising a right-hand pinion and left-hand gear, the pinion turning left-handedly. There is a similar set of four equations for other combinations, but as these combinations are not used in automotive drive, the equations need not be given here.

HYPOID PINION LOADS

$$T_p = \frac{F}{\cos \psi_g} (\tan \phi \sin \gamma + \sin \psi_p \cos \gamma) \text{ lb.}$$

$$R_p = \frac{F}{\cos \psi_g} (\tan \phi \cos \gamma - \sin \psi_p \sin \gamma) \text{ lb.}$$

HYPOID GEAR LOADS

$$T_g = \frac{F}{\cos \psi_g} (\tan \phi \sin \Gamma - \sin \psi_g \cos \Gamma) \text{ lb.}$$

$$R_g = \frac{F}{\cos \psi_g} (\tan \phi \cos \Gamma + \sin \psi_g \sin \Gamma) \text{ lb.}$$

With hypoid gears the spiral angle of the pinion is made 50° for up to 13 teeth, 45° for 14 and 15 teeth, and 40° for more than 15 teeth.

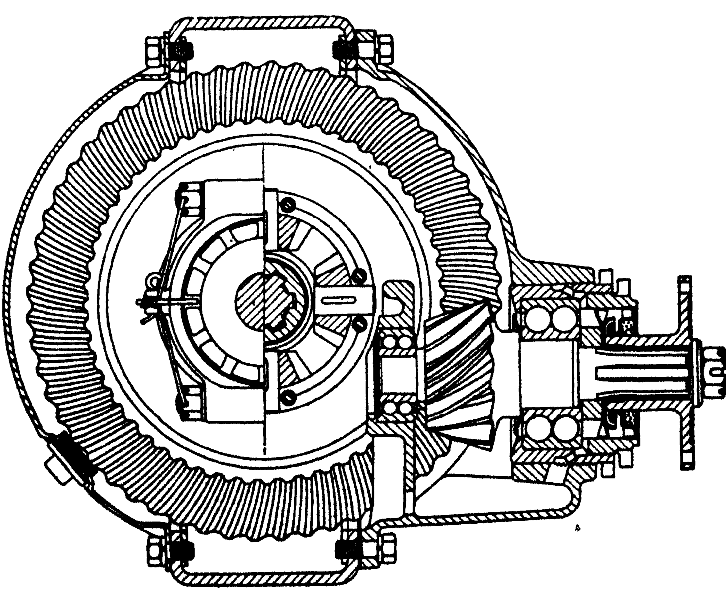


FIG. 15.—HYPOID-GEAR ASSEMBLY WITH STRADDLE MOUNTING.

Gear Mountings—For satisfactory operation it is necessary that the gears be rigidly mounted. This calls for the use of bearings of adequate size, properly spaced. Bearing sizes are chosen in accordance with the calculated bearing loads, as explained further on. In axles for heavy, powerful vehicles the pinions usually are straddle-mounted, as shown in Fig. 15, while in most passenger-car axles they overhang their bearings, as in Fig. 16. Gleason Works specify that under maximum torque the pinion must not lift or sag more than 0.003 in., and not yield more than 0.003 in. axially. To ensure such rigidity, the center spacing of bearings in overhung mountings should be at least 2.5 times the distance of the pinion center from the center of the adjacent bearing. With a mounting in which the differential bearings are equally

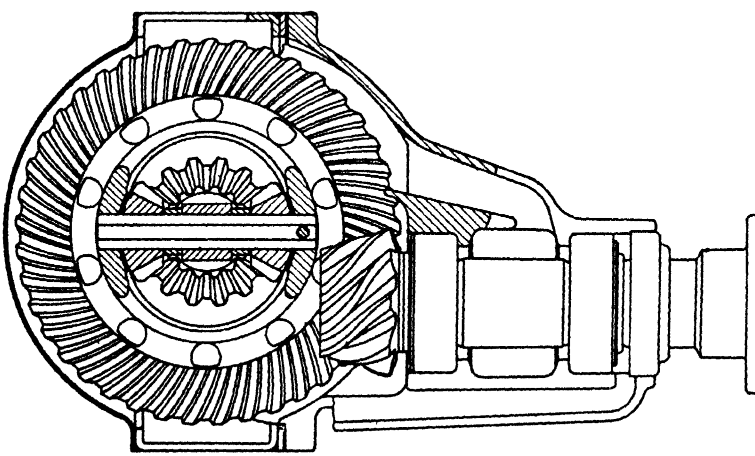


FIG. 16.—HYPOID-GEAR ASSEMBLY WITH OVERHUNG PINION MOUNTING.

far from the center plane of the differential (symmetrical layout), the center distance between them should be at least equal to the gear diameter, as otherwise it is impossible to provide adequate ribbing for the differential flange to which the ring gear is bolted. In asymmetrical design the distance between differential bearings can be made somewhat less, but there the distance from the center of the differential to the center of the bearing back of the ring gear should be at least equal to one-half the gear diameter. To ensure satisfactory gear operation, the ring gear must not lift more than 0.003 in. under maximum low-gear torque, and must not move more than 0.010 in. away from the pinion at the point of mesh. Some designers, in order to limit the movement of the gear away from the pinion, place an adjustable thrust button, or thrust shoe, back of the ring gear at the point of mesh. This thrust button is adjusted to have about 0.010 in. clearance, and it supports the ring gear only after this clearance has been taken up. A heavier construction evidently would be required to provide equal protection for the gears without the thrust button, and the only valid objection to the latter is that it may possibly be maladjusted.

It has been the general custom to provide ring gears with an internal web by which they are bolted to the flange on the

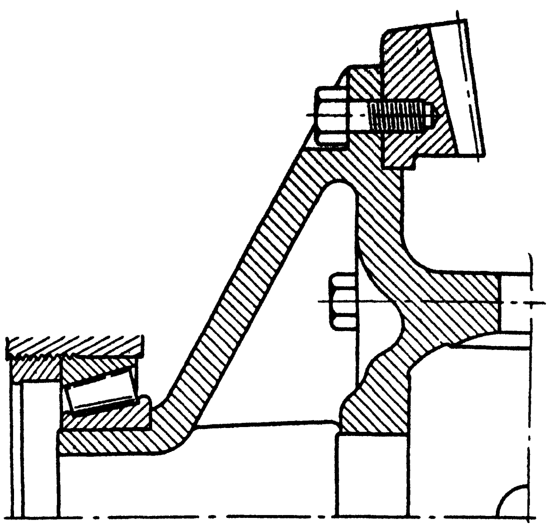


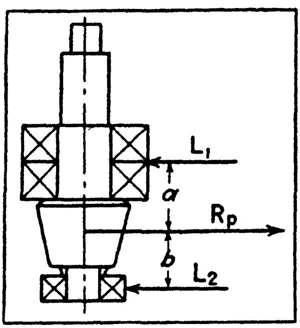
FIG. 17.—SPIRAL BEVEL GEAR ASSEMBLY COMPRISING A "WEBLESS" RING GEAR.

differential housing. The Gleason Works recommends that the gears be made without a web, as shown in Fig. 17, to make it possible to provide adequate ribbing for the differential flange. This construction is now being used by several leading manufacturers. There should be at least six ribs on the housing, either internal (as shown in Fig. 17) or external, and at least 12 bolts holding the ring gear.

In passenger-car axles the overhanging mounting is now commonly used. It goes together well with hypoid gears, where the larger pinion permits the use of a large pinion shaft and large bearings.

Determination of Bearing Sizes—After the radial and thrust loads due to tooth pressure have been calculated by means of the equations given in the foregoing, it becomes necessary to determine the division of the radial loads on pinion and gear between the two bearings supporting each. If the pinion is straddle-mounted, the sum of the loads on the two bearings is equal to the entire radial load, and the load on each bearing is inversely proportional to the distance of its center from the center of load distribution on the pinion. Using the notation given in Fig. 18 the load on the bearing back of the pinion is

$$L_1 = \frac{b}{a + b} R_p$$



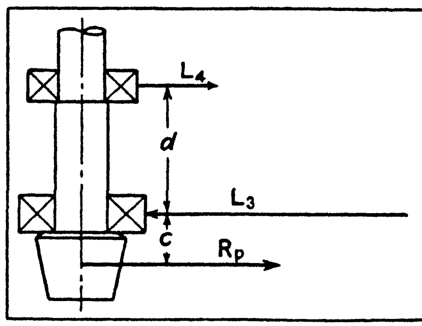


FIG. 18 (*left*).—DIVISION OF BEARING LOAD WITH STRADDLE MOUNTING.
FIG. 19 (*right*).—DIVISION OF BEARING LOAD WITH OVERHUNG MOUNTING.

and that on the bearing at the small end of the pinion,

$$L_2 = \frac{a}{a + b} R_p.$$

When the pinion overhangs its bearings the load on the bearing directly back of it is greater than the radial load due to the tooth pressure. It is equal to the sum of that load and the load carried by the remote bearing, which latter two are inversely proportional to the distances of their centers from the center of the bearing immediately back of the pinion. Using the notation of Fig. 19 the load on the bearing close to the pinion is

$$L_3 = \frac{c + d}{d} R_p$$

and that on the forward bearing,

$$L_4 = \frac{c}{d} R_p.$$

The radial load on the gear divides between the two differential bearings in the same way as the radial load on the pinion divides between the two pinion bearings in the case of straddle mountings. After the bearing loads have been thus determined, it is well to consult with the bearing manufacturer regarding the choice of bearing sizes.

Hypoid Assemblies—Additional assembly views of hypoid rear-axle gears are shown in Figs. 20 to 22. The same as spiral bevel gears, hypoid gears are mounted in gear carriers pro-

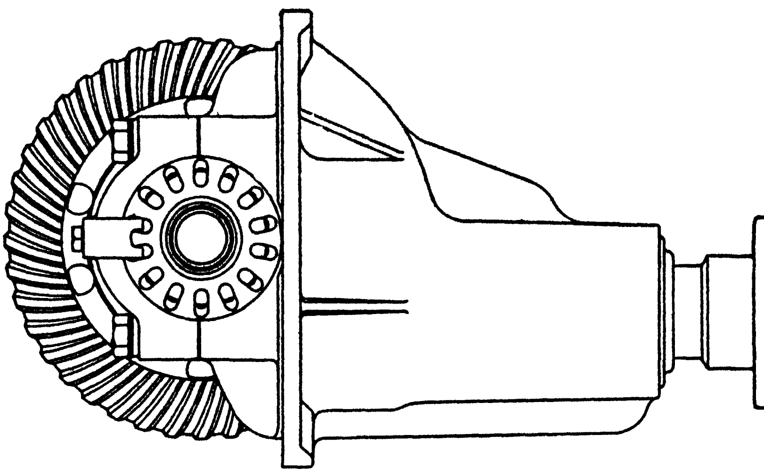


FIG. 20.—DIFFERENTIAL CARRIER FOR HYPOID GEARS.

vided with flanges by which they are bolted to the axle housing. Hypoids give rise to some special design problems, because the pinion axis is offset from the center of the flange (which usually is in line with the center of the differential) both horizontally and vertically, the gear carrier assuming a rather asymmetric form. One solution is to tilt the axle banjo about 45°, which brings the bearing boss close to the banjo. In other designs, to ensure the necessary rigidity, the bearing boss for the pinion shaft must be well ribbed to the carrier flange. Two methods of ribbing are illustrated in Figs. 20 and 22. It is common practice in connection with overhung pinion mountings to provide for oil circulation through the space between the two pinion bearings. As shown in Fig. 16,

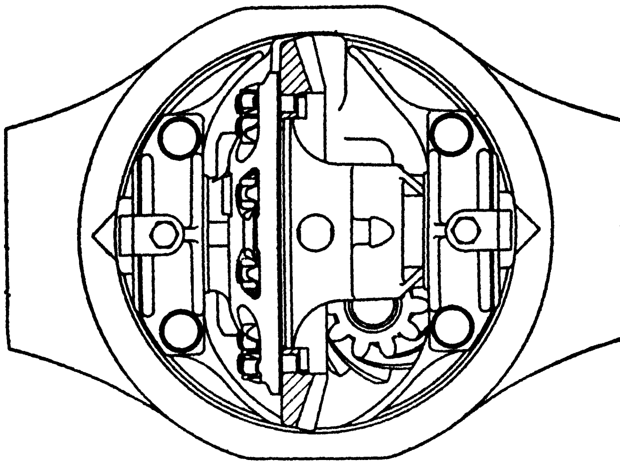


FIG. 21.—REAR VIEW OF HYPOID-GEAR ASSEMBLY.

oil thrown off by the final-drive gear can enter this space through a channel in the upper part of the gear carrier, and drain back to the center housing through another channel below the shaft. Figs. 20 and 21 show the caps of the differential bearing supports, which are held in place by two cap screws each, and the adjusting nuts for the roller-type differential bearings, as well as the locking means for these nuts. In the Dodge axle the ring gear is secured to the flange by means of cap screws which have a part of their head cut away.

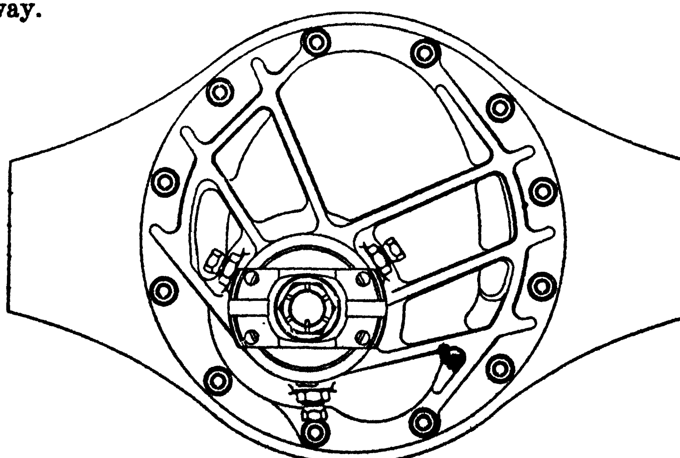


FIG. 22.—FRONT VIEW OF HYPOID-GEAR CARRIER.

CHAPTER X

Double-Reduction and Two-Speed Axles

During the early years of the industry, when solid rubber tires and poor roads limited speeds to between 10 and 15 mph, practically all commercial vehicles except those equipped with worm-gear drive required a double reduction between the engine crankshaft and the drive wheels. Since then the need for such drives has greatly diminished, because of the increase in vehicle speeds and the introduction of spiral bevel and hypoid gears, which permit of a greater speed reduction in a single step than a straight bevel gear drive. At present double-reduction drives are used only on the heavier types of commercial vehicle, and particularly on the "off-the-road" type. On such vehicles a single-reduction geared drive usually would call for a center housing of such large diameter as to leave insufficient road clearance under the axle. The terms "double-reduction drive" and "double-reduction axle" are most generally applied to a construction in which the rotative speed of the engine crankshaft is reduced by one pair of bevel, and one pair of either spur, helical or herringbone gears, both enclosed in a housing at the center of the axle. However, the terms are applicable to all designs in which there is a double reduction in rotative speed, regardless of the type of mechanism by which it is effected and where this mechanism is located on the axle.

Owing to the fact that the propeller shaft extends at right angles to the rear axle, one reduction is always by bevel gears. The other pair may consist of spur gears, herringbone gears, or a spur pinion and an internal gear; and the second pair of gears may be located at the center of the axle, or there may be two pairs of second-reduction gears which may be located at the center or the ends of the axle, or in any intermediate position.

As a rule, the first reduction is by bevel gears, because bevel gears create end thrust, and if they are used for the first reduction the end thrust will be smaller and therefore more readily taken care of. However, the opposite arrangement, in which the second reduction is by bevel gears, also has been used, probably for manufacturing reasons. A train of double-reduction gears comprising a first-reduction pair of spiral bevel and a second-reduction pair of herringbone gears is shown in Fig. 1.

There are several possible arrangements of the bevel-spur and similar gear trains of double-reduction drives. In the first place, all of the shafts (driving-pinion, intermediate, and differential) may be located in the same plane, which is

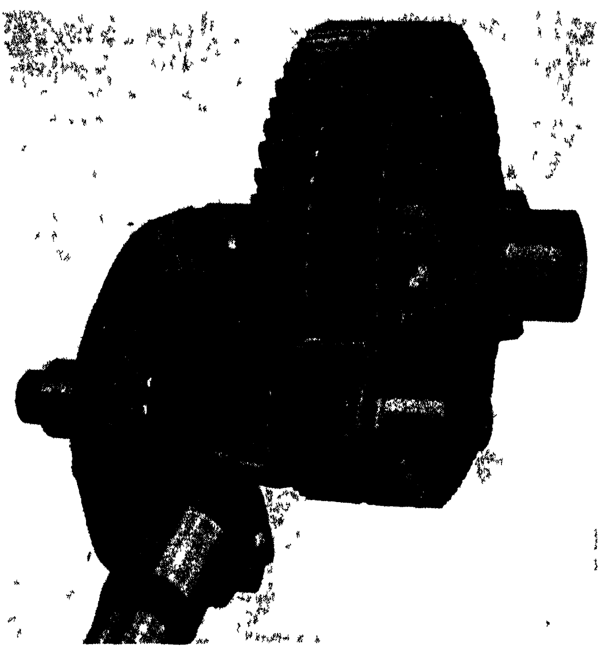


FIG 1—ASSEMBLY OF SPIRAL BEVEL AND HERRINGBONE GEARS.

preferably slightly inclined to the horizontal to give a nearly straight-line drive. This is exemplified in Fig. 2, which shows a bus axle of an early European design. The housing for the whole assembly is split in a plane through the shaft axes, and axle tubes are flange-bolted to it. This solves the assembling problem, because all internal parts can be assembled on the lower half of the housing and the upper half then applied. The drawing also illustrates the former European practice of providing separate ball bearings for radial and

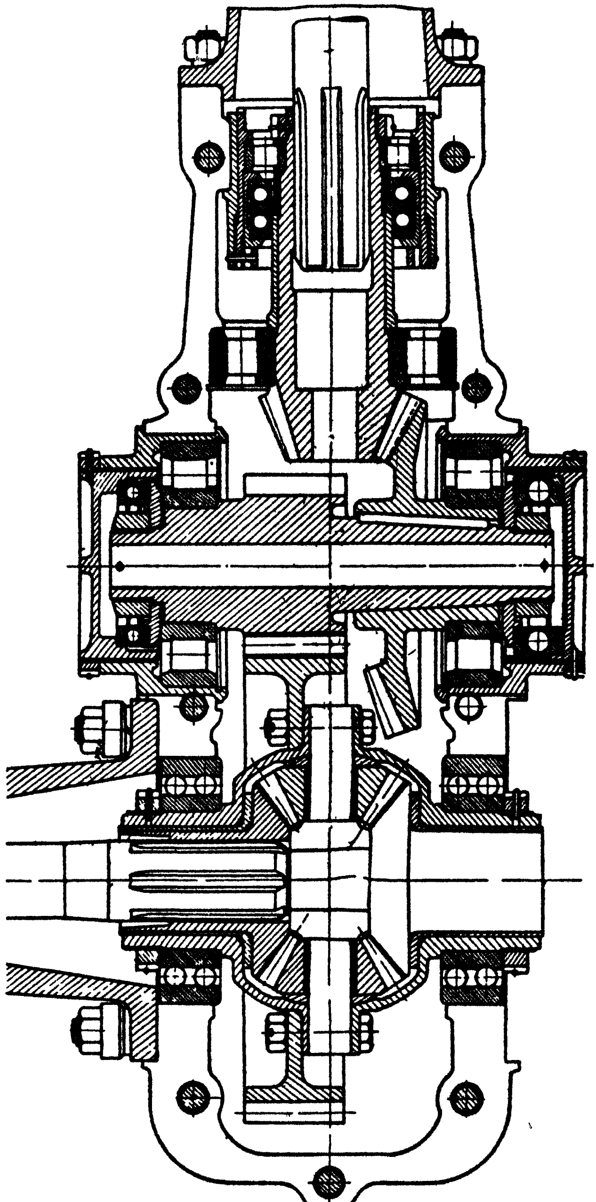


Fig. 2.—Double-Reduction Drive with All Shafts in the Same Plane.

thrust loads. With the first reduction gear in front of the axle proper, the housing naturally extends forward a considerable distance, creating a moment around the axle axis, which might be considered a disadvantage. But since on forward drive this moment is in opposition to the torque reaction on the axle housing, it is rather an advantage.

Another arrangement consists in having the intermediate shaft directly—or nearly directly—above the differential, as in the Timken double-reduction axle illustrated in Figs. 3 and 4. In this design the center housing of the axle is bowl-

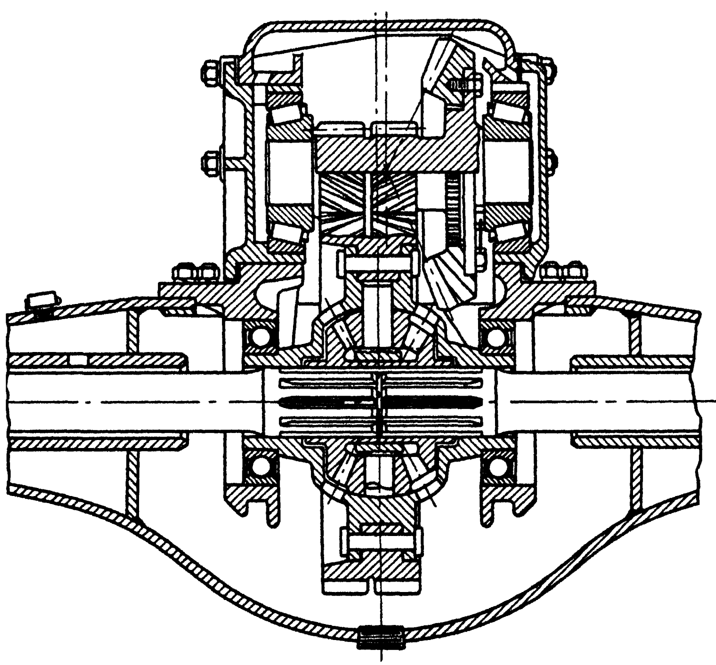


FIG. 3.—TIMKEN DOUBLE-REDUCTION DRIVE, VERTICAL SECTION THROUGH AXLE CENTER.

shaped, and the differential carrier is bolted over an opening in the top. The bowl-shaped housing naturally is considerably more rigid than an annulus with a thin-walled cover plate bolted to it. An advantage of this design from the production standpoint is that the differential carrier is interchangeable with one having a worm-gear drive. The spiral bevel pinion is mounted in a special housing piloted in a bore in the differential carrier, adjustment of the bevel gears being effected by means of shims under the flange on this housing. The herringbone pinion is formed integral with its shaft, while the spiral bevel gear is secured to a flange thereon by means of cap screws. Cover plates over the openings in the housing for the intermediate shaft have circular flanges formed on their inner side, which serve as seats or mountings for the outer races of the roller bearings. Shims under the cover plates serve a dual purpose. Adding or removing shims changes the bearing clearance or degree of preloading, while

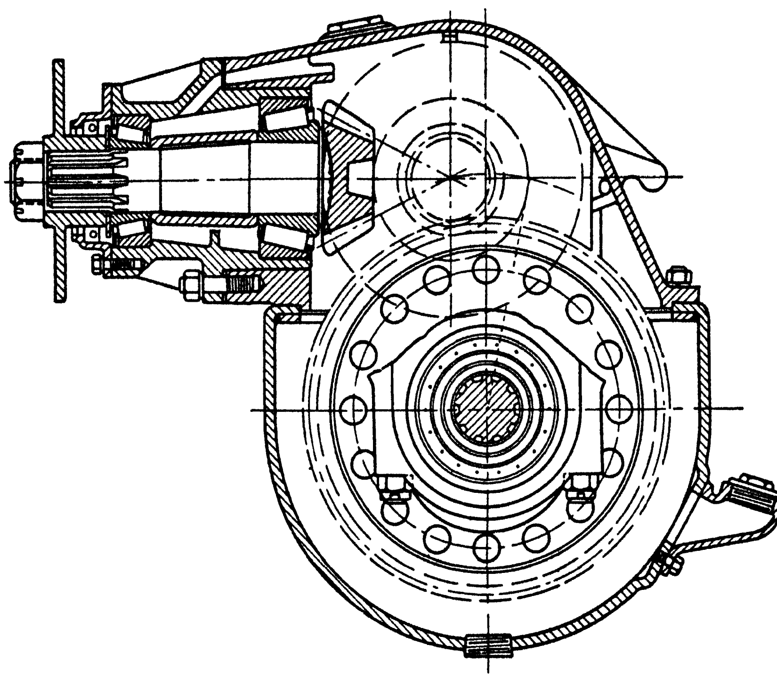


FIG. 4.—TIMKEN DOUBLE-REDUCTION DRIVE, VERTICAL SECTION THROUGH PINION AXIS.

transfering a shim from one side to the other changes the mesh of the gears.

Most frequently the intermediate shaft is located forward of and at a somewhat higher level than the differential. A side view of a center housing enclosing gears thus arranged is shown in Fig. 5. There the joint between the differential carrier and the axle housing is in a 45° plane, but when the intermediate shaft is not very much higher than the differential shafts, it is possible to have the joint in the vertical plane. A double-reduction axle in which the first reduction is by spur and the second by bevel gears is illustrated in Fig. 6. This design is no longer being used, but it gave good service for many years. The axle comprises a cast center housing into which the differential is inserted from the rear, the differential bearings being mounted directly in it. A problem that arises in connection with all of these designs is that of ease of assembling. In this particular case the intermediate shaft, with the bevel pinion and spur gear secured to it, evidently is inserted through the opening on top of the housing, the forward end being passed through the opening for the front bearing. After being maneuvered into position, with the ball bearing at its rear end in its seat, the spacer is introduced through the front-bearing opening, then shims and the

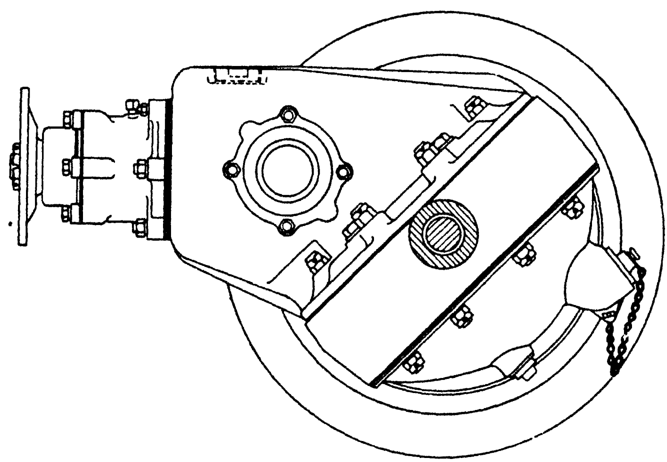


Fig. 5.—Double-Reduction-Axle Center Housing with Banjo at 45 Deg.

forward bearing are inserted, and finally a nut is screwed onto the forward end of the shaft, which holds all of the parts carried by it in their correct lengthwise positions. The outer race of the front bearing is secured between an inward flange in the bearing boss and a threaded cap screwing into the latter.

Reduction Ratios—In a double-reduction drive of the type discussed in the foregoing, the ratio of the first reduction is always made smaller than that of the second. The reason for this is that the gear of the second-reduction pair, which is mounted on a flange on the differential housing, must necessarily be of considerable diameter, hence it is possible to obtain a reduction ratio of at least 4:1 without resorting to the use of a pinion with an inordinately small number of teeth. Besides, holding down the ratio of the first-reduction pair keeps down the diameter of the gear and makes for a more compact layout. A first-reduction ratio of about 2 is commonly used.

The gears of double-reduction axles can be calculated in the same way as those of single-reduction spiral bevel and hypoid axles. As far as the first-reduction pinion is concerned, it transmits the same torque and turns at the same speed as that of a single-reduction axle in a vehicle having the same maximum engine torque, and the maximum permissible tooth

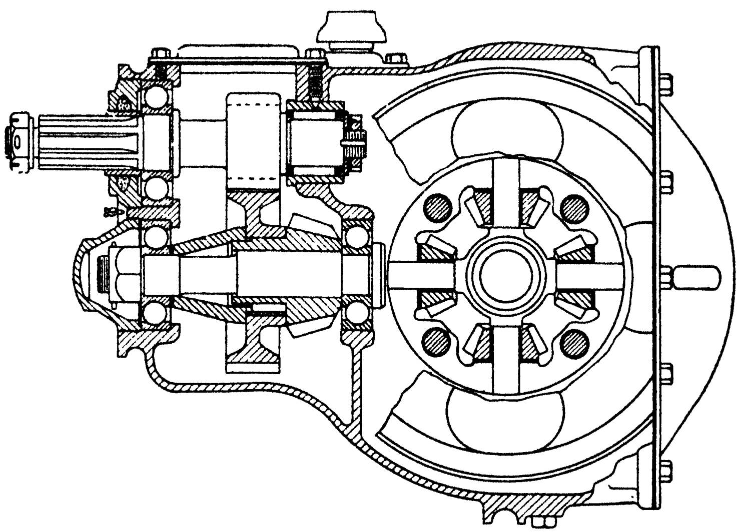


FIG. 6.—DOUBLE-REDUCTION GEAR WITH FIRST REDUCTION BY SPUR GEARS.

stress in the pinion therefore can be taken directly from the Almen-Boegehold *S-N* chart, Fig. 4, Chapter IX. The permissible stresses for the remaining gears also can be obtained from that chart, the fatigue lives of these gears being reduced in proportion to their speeds of rotation as compared with that of the driving pinion. In the strength formula for spur, helical and herringbone gears, the so-called taper factor is omitted, and the required face width is

$$W = \frac{6hF}{t^2S} \text{ in.},$$

the notation being the same as that in Chapter IX.

Second-Reduction Beyond Differential—In all of the double-reduction drives thus far discussed, both reductions in speed take place ahead of the differential gear. There is a certain advantage in effecting one reduction beyond the differential gear, as the latter then does not need to transmit so much torque and can be made smaller. However, the second-reduction gear in that case must be in duplicate, which tends to nullify the advantage. This second reduction gear can be located either adjacent to the differential, toward the ends of the axle on the inner side of the wheels, or at the extreme ends of the axle, on the outer side of the wheels. When it is located near the differential, the gear must be of the planetary type, as in Fig. 7, which shows an axle center housing of the British Scammell eight-wheel truck. This center housing has three compartments, the central one enclosing a set of spiral bevel gears with a 2:1 ratio, and the differential gear, and each of the outer compartments containing a planetary assembly of the internal-gear type. A short shaft extending laterally from the side of the differential compartment carries a sun pinion which meshes with three planetary pinions, the latter in turn meshing with an internal ring gear. In this reduction mechanism power is applied to the sun pinion and leaves through the planet carrier, which is splined to the axle shaft. In a planetary gear of this type, if we designate the number of teeth in the sun pinion by a and that in the ring gear by d, the reduction ratio is $(d + a)/a$, and the number of teeth in each planetary pinion, $(d - a)/2$. In this particular design there are 52 teeth in the ring gear, 14 in the sun pinion, and 19 in each planetary pinion; consequently, the reduction ratio is $(52 + 14)/14 = 4.715$, which together with a reduction ratio of 2 by the bevel gears gives an axle ratio of 9.43. It will be seen that the short differential shaft "floats," being supported by the differential side gear and the sun pinion of the planetary assembly, and that the sun pinion is supported solely by the three planet pinions.

A second reduction by spur gears on the inner side of the road wheels is used in the German MAN bus axle illustrated in Fig. 8. In buses it is desirable to have the floor of the body —and consequently also the chassis frame—as low as possible, as this increases the stability of the vehicle and facilitates passenger ingress and egress. Therefore, a drop axle is often used, in which the axis of the differential is at a lower level than the wheel axis. The housing of the axle represented by Fig. 8 is of cast construction, in three parts. The end portions, which comprise the wheel spindles, are piloted on the center portion. Housings for the second-reduction gears are formed between the central and end portions of the axle housing. One disadvantage of this design seems to be that the gear housing, being located inside the brake drums, cramps the brake mechanism.

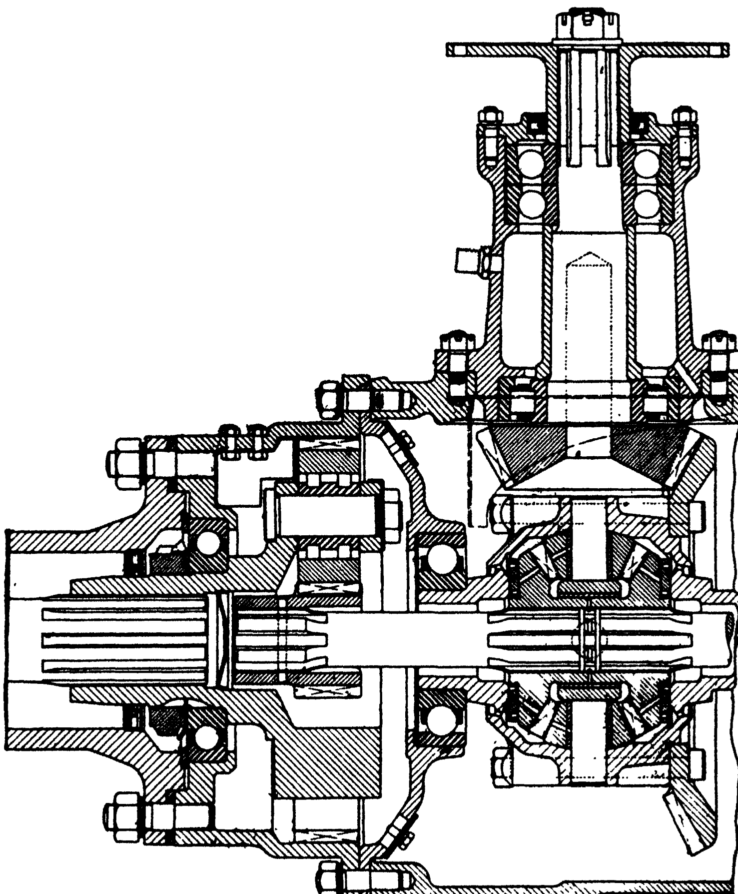


FIG. 7.—DOUBLE-REDUCTION GEAR WITH SECOND REDUCTION BY PLANETARY GEARS.

Internal Gears—At one time internal gears were extensively used for the final drives of light trucks and delivery wagons, in combination with a pair of bevel gears. Vehicles of this type now generally have single-reduction drives by spiral bevel gears, and internal-gear drives have largely dis-

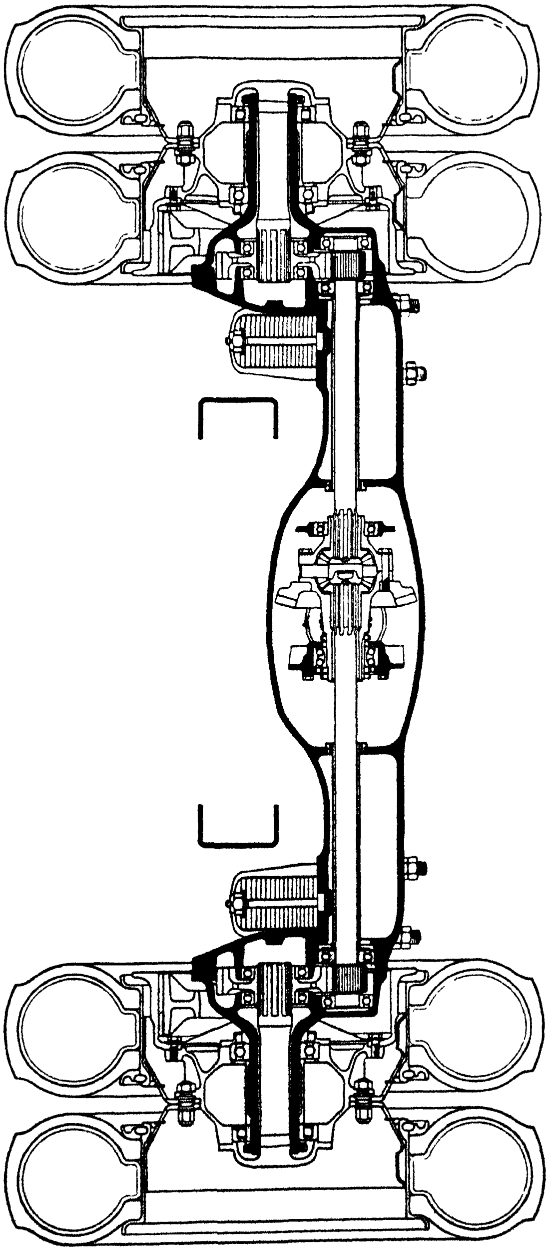


FIG. 8.—DROP-TYPE BUS AXLE WITH SECOND REDUCTION BY SPUR GEARS AT WHEELS

appeared. It has always been claimed for internal gears that they are more efficient than other types of toothed gearing, this claim being based on the fact that there is less sliding motion between the teeth of a spur pinion and an internal gear than between those of a pair of spur or bevel gears. Internal gears are always mounted on the road wheels, either on the inner or the outer side. When the internal gear is on the inner side of the road wheel, it is driven by a pinion eccentric to the axle spindle, at the end of one of the differential shafts. If the internal gear is on the outer side of the road wheel it is in mesh with a number of equally spaced idler gears which in turn mesh with the driving pinion, the latter in this case being mounted concentric with the axle spindle, on the end of the differential shaft extending through the spindle.

Where internal gears can be rigidly mounted and operated in a bath of oil, they are unusually quiet in operation, but it is impossible to operate them in a bath of oil on the inner side of the road wheels. The gear itself forms a sort of drum which can be closed by a cover or backing plate, but since the pinion shaft must extend through this plate, the later must be non-revolving, and there must be a sliding joint between it and the revolving gear. Such a joint can be so made that it is fairly dustproof, but it will not hold oil. In fact, as the internal gear is located within the brake drum, it must be operated without lubricant. This is undoubtedly one of the reasons why axles with internal gears on the inner side of the road wheels are no longer being used to any extent.

A final drive through internal gears located on the outer side of the road wheels is used in the heavy truck axle illustrated in Fig. 9. In this axle there is a bearing on the axle shaft outside the central pinion, which is supported by a spider from the studs on which the idler pinions revolve. In one design of axle of this general type the axle shaft was at first supported in a bearing mounted in the end of the axle tube, so that it overhung its bearing. This, however, gave a great deal of trouble, and much better results were obtained by entirely eliminating the bearing and letting the pinion seek its position of equilibrium between the three idler gears in mesh with it.

Another means whereby a second speed reduction may be effected on the outer side of the road wheels consists of a bevel-type differential gear. The inner side gear is keyed or splined to the axle housing, the outer one is secured to the end of the axle shaft, and the pinion carrier is secured to or made integral with the wheel hub. A reduction of 2:1 is obtained in this way. A British manufacturer of commercial vehicles (Foden) has used a reduction gear of this type. End thrust on the side gears is taken up on deep-groove radial ball bearings mounted, respectively, in the extension of the wheel hub which serves as pinion carrier, and in a cap or cover secured thereto by cap screws.

A double reduction in speed is obtained also with the so-called angular drive used with buses in which the power-plant is mounted transversely at the rear. Both reductions

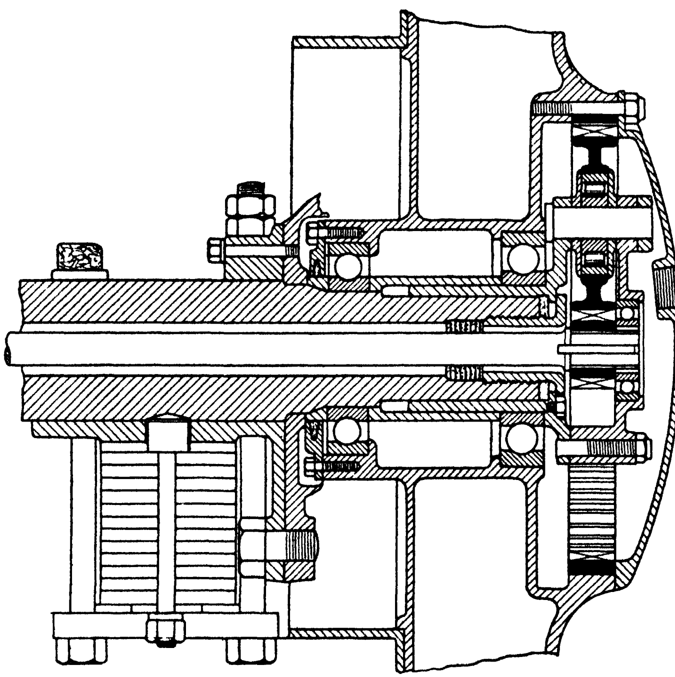


FIG. 9.—DOUBLE-REDUCTION AXLE WITH SECOND REDUCTION BY INTERNAL GEAR AT OUTER SIDE OF ROAD WHEEL.

are by spiral bevel gears, one set being enclosed in a compartment of the transmission housing, the other in the axle housing.

Two-Speed Axles—During the early part of the second decade of the current century there was considerable interest in two-speed axles for passenger cars, and the 1914 model of the Cadillac, among others, was equipped with such an axle. These axles had two different-sized bevel gears on the differential housing, two bevel pinions in mesh therewith on the driving shaft, and a shifting mechanism by means of which one member of either pair could be made rigid with its shaft or disengaged therefrom at the will of the operator. The object aimed at with these two-speed axles is now accomplished more satisfactorily—from the driver's standpoint—with the

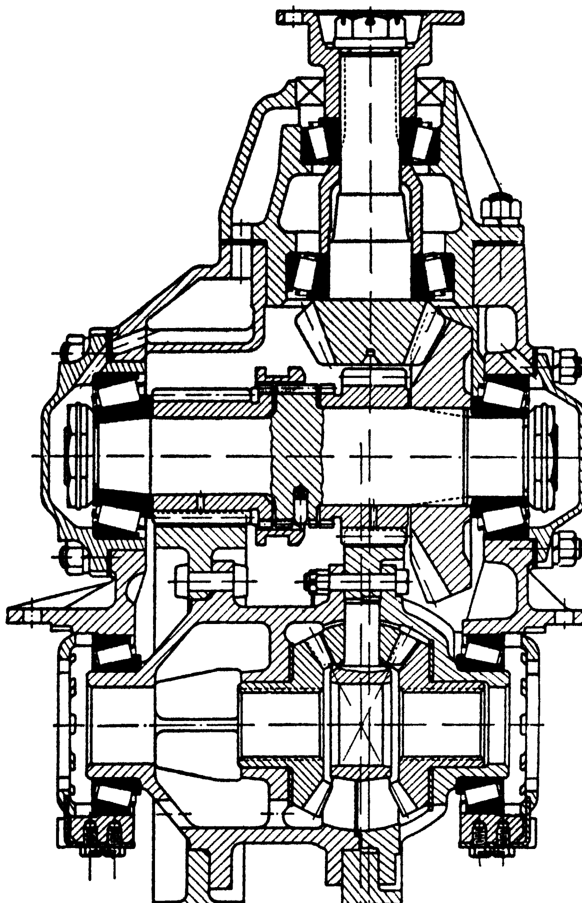


FIG. 10.—GEAR ASSEMBLY OF TIMKEN TWO-SPEED AXLE.

automatic overdrive, and two-speed passenger-car axles have gone out of use.

At present two-speed axles are incorporated in some heavy trucks, to obtain the same range of axle torque or tractive force which in other designs is obtained with multi-speed transmissions. However, the passenger-car type of design, with two pairs of bevel gears, is not used for this purpose, apparently because the problem of providing adequately for the end thrust on two bevel pinions involves considerable difficulty. In a two-speed axle either both speeds may be obtained by double reductions or one speed by a single and the other by a double reduction. A two-speed axle manufactured by the Timken-Detroit Axle Company, illustrated in Fig. 10, has a first-reduction pair of spiral bevel gears and two second-reduction pairs of helical gears. The pinions of both sets of helical gears are loose on the intermediate shaft, but can be placed in driving relation therewith by means of an internally splined sliding collar, the teeth of which engage both with spur teeth on the central portion of the intermediate shaft and with corresponding teeth at the inner ends of the pinions. In designing two-speed axles of this type, it is the usual practice to make the ratio between the two gear ratios or between the two speeds equal to the square root of the ratio between successive transmission speeds, so that two ranges of road speeds are obtained, the individual speeds forming a geometrical series. Alternately, the ratio between axle gear ratios may be so proportioned as to give with one axle ratio and a particular transmission ratio a vehicle speed which is midway between the road speeds obtained with the other axle ratio and the next higher and next lower transmission speeds, respectively. This latter plan seems to have been followed in the design of the Timken axle, which has a high gear about 1.30 as fast as the low gear. In four-speed truck transmissions the ratio between successive gear speeds is usually about 1.88. Suppose that a truck with both the transmission and the axle in high gear has a road speed of 50 mph. Then, if the next-to-highest gear in the transmission is 1.88, the maximum road speed in that gear with the axle in high will be $50/1.88 = 26.6$ mph. If, on the other hand, the axle were shifted into low and the transmission remained in high, the road speed would be 38.5 mph, which is about midway between 26.6 and 50 mph.

Two-speed axles can be evolved also by combining a two-speed planetary assembly with the axle driving mechanism. One such two-speed axle, manufactured by Eaton Manufacturing Co., is illustrated in Fig. 11. The straddle-mounted bevel pinion meshes with a ring gear which, instead of being rigidly secured to the housing of the differential gear, is formed integral with the ring gear of the planetary assembly, the housing of which is supported on what are usually referred to as the differential bearings. The housing of the differential forms the planet carrier. When the planetary gear is in action, its sun pinion is held from rotation by means of clutch teeth on its shaft engaging with internal teeth on a clutch member rigidly secured to the axle housing. This clutch member, by the way, has the additional function of an adjusting nut for the roller bearing supporting both the planetary and

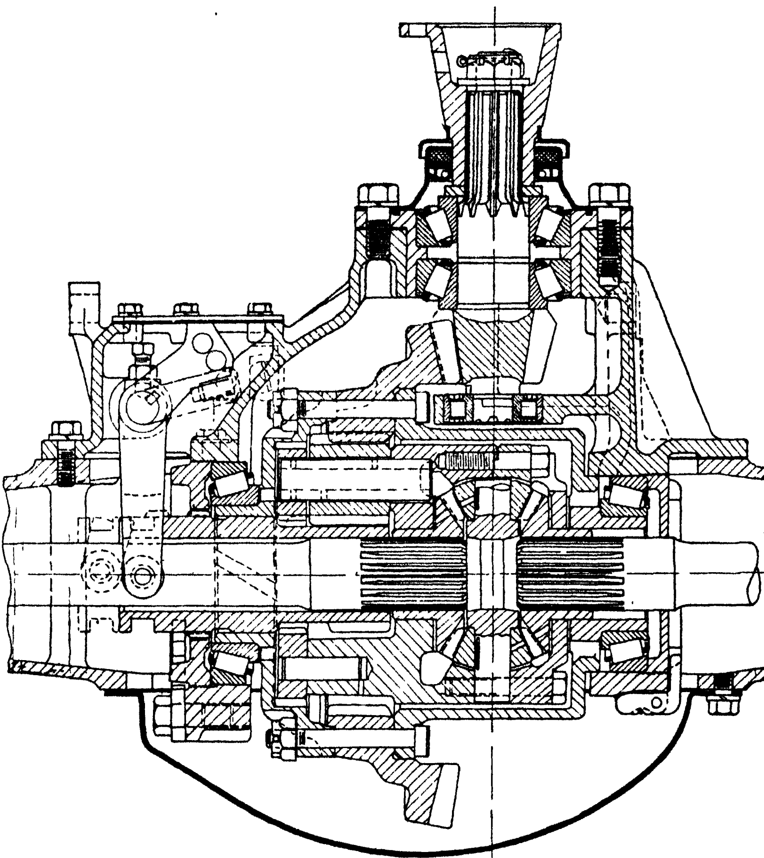


FIG. 11.—GEAR ASSEMBLY OF EATON TWO-SPEED AXLE.

the differential assembly. When the sliding member (the sun pinion) is in the position in which it is shown in the drawing, power enters the planetary assembly through the ring gear and is delivered by it through the planet carrier to the differential.

With an assembly of this type the reduction ratio is

$(d + a)/d$, where d is the number of teeth in the ring gear and a the number in the sun pinion. In this particular case the ring gear has 46 and the sun pinion 18 teeth, hence the reduction ratio is $(46 + 18)/46 = 1.391$. When the axle is in high speed, the planetary assembly is locked, the sun pinion then being shifted to the left so that its teeth engage internal teeth on a part of the planet carrier. This same shifting motion disengages the clutch teeth on the sliding member from those of the stationary clutch member, as indicated in dotted lines. The spiral bevel gears of this axle have 7 and 36 teeth, and give a high ratio of 5.142 and a low ratio of 7.15.

In a later model of this axle a number of refinements were introduced. The shifter fork now is splined to its shaft, and provision can be made for shifting by means of a vacuum power unit. The pitch diameter of the bevel gear was increased 5 per cent, which increased the face width of the gear about 25 per cent. A pressed-steel drum was provided back of the bevel gear, which picks up oil, and this oil is scraped off by the edge of a sliding tube and delivered through separate channels to the pinion bearings and the right-hand bearing of the differential. The two-speed gear remained unchanged in principle.

CHAPTER XI

Front Drive, All-Wheel Drive and Six-Wheelers

Front Drive—The great majority of all motor-propelled road vehicles always have been driven by their rear wheels. It is possible, however, to drive a vehicle by its front wheels, and the method offers certain advantages. Shortly before 1930 a wave of interest in front drive was stirred up in the United States, and two front-driven passenger cars were developed here, the Ruxton and the Cord. The first of these never reached the production stage, while the second was in production on a moderate scale for a few years, and then disappeared as the manufacturer (who also produced conventional cars) went out of business. From here the interest spread to Europe, and especially to the Continent, where a somewhat greater degree of success was achieved. Of the total German production of passenger cars in 1934, approximately 20 per cent had front drive, and at the Paris automobile show held toward the end of that year, eight out of a total of about 60 models exhibited were front-driven.

At the time when interest in front-wheel drive ran high, various priority claims were put forth, but a search then instituted revealed that the idea of driving a road vehicle by its front wheels is a very old one. One of the most celebrated patent cases ever fought in the courts of this country centered around the Selden patent, for which application was filed in 1877, and which was claimed to be a master patent broadly covering the gasoline automobile. The drawings accompanying the specification of this patent showed a front-driven vehicle. The first motor-propelled vehicle of which there is an authentic record of its having been actually built, was a steam military tractor constructed by Joseph Cugnot in France about 1770. It had only a single front wheel, and that was a combined steering and driving wheel.

Advantages—As compared with a conventional car, the chief advantage of the front-driven one is that it has no propeller shaft extending between the front-mounted powerplant and the rear axle; consequently, the floor of the body can be placed quite low, and the center of gravity of the car thus lowered. Of course, the same advantage is inherent in rear-engined cars with drive to the rear wheels, and front-driven and rear-engined cars may be considered different solutions of the same basic problem. Front-driven cars also are said to have less tendency to skid when rounding curves at speed. They improve the riding quality of the rear seat, because when a rear driving axle is replaced ~~by a~~ "dead" ~~axle,~~ the unsprung weight at the rear is materially reduced, ~~and riding~~ comfort is largely a function of the ratio of sprung ~~to~~ unsprung mass. It also stands to reason that there is less noise in the rear part of the car when all the driving mechanism is located in front; but since the general noise level is not lowered, the front compartment is somewhat more noisy.

Disadvantages—The chief disadvantage of front drive is that with it the maximum tractive force available is less than with rear drive. Under operating conditions making it difficult to get sufficient traction, there will always be considerably less weight on the front than on the rear wheels, and the maximum traction which a wheel can develop is directly proportional to the force pressing it against the road surface It is not difficult to so design a front-driven passenger car that without passenger load and while at rest on level ground there will be considerably more weight on the front than on the rear wheels; but under conditions that call for maximum tractive force, the weight distribution is materially changed. In the first place, most of the passenger weight comes on the rear wheels. Secondly, the torque on the driving wheels produces a reaction on the chassis which adds to the effective load on the rear wheels and subtracts equally from that on the front wheels. Finally, as indicated in Fig. 1, when a car is traveling on an up-grade, the weight on its front wheels is reduced by the inclination, and the weight on the rear wheels is increased. The car shown by the drawing has its weight equally divided between the front and rear wheels when on level ground, as indicated by the fact that the center of gravity is midway between the front and rear axles. When on an incline, the weight on the front wheels is proportional to the distance from a perpendicular through the center of gravity to the center point of ground contact of the rear wheels, and that on the rear wheels, proportional to the distance from the perpendicular through the center of gravity to the center point of ground contact of the front wheels. On a grade of 18 per cent, as shown in Fig. 1, the weight distribution, as a result of the inclination, will change from 50:50 to about 43.5:56.5. In a car of average proportions and "ability," the maximum torque will take about 4 per cent of the total car weight from the front wheels and add it to the rear wheels.

Loss of Tractive Force—It has been shown analytically that with a car of 120-in. wheelbase, with the center of gravity midway between axles and 28 in. above the road surface, if the traction resistance is 20 lb per 1000 and the coefficient of road adherence 0.6, it is possible to obtain 30 per cent more tractive force with rear than with front drive, and 126 per cent more with four-wheel than with front drive, the latter under the condition that the driving torque is so divided between front and rear wheels that the road adhesion of both

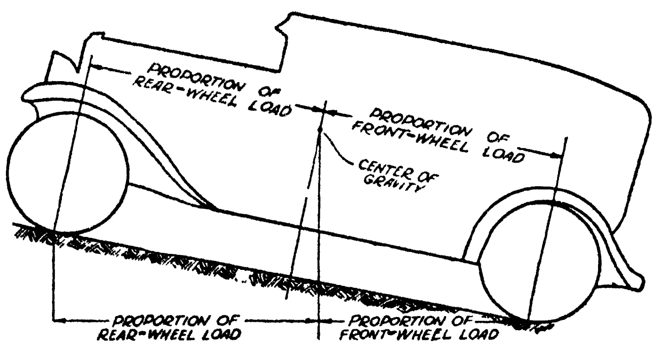


Fig. 1.—Showing How Weight Is Transferred from Front to Rear Wheels by an Up-Grade.

is completely utilized. The maximum grades which can be ascended by vehicles with the different drives are 24.6 per cent for the front-drive, 32.5 for the rear-drive, and 58 per cent for the four-wheel drive. As these figures indicate, no trouble is likely to be experienced from loss of traction with a front-drive vehicle as long as the coefficient of road adhesion is anywhere near 0.6; it is when the roads are muddy or icy that this weakness of the front drive shows up.

Low Floor Height—Low floors are desirable because they lower the center of gravity, and thus increase the stability of the car, and also because they permit of reducing the over-all height of the vehicle and to thus improve its appearance. In conventional passenger cars, in order to make up for what may be considered excessive floor height, the depth of the seat cushions is sometimes reduced, which results in less comfortable seats. With the low floors possible with front drive, the depth of the cushions need not be thus limited. A disadvantage of very low floors and resulting low seats is that the visibility from the driver's seat is impaired. The minimum permissible floor height is limited by considerations of ground clearance required. As shown in Fig. 2, the minimum ground clearance may well be somewhat lower in a front-driven than in a rear-driven vehicle, because in a car of long wheelbase, trouble from insufficient ground clearance between axles is most likely to be encountered when passing the crest of a hill. The ground clearance is then reduced most at a point about midway between axles, and in a rear-drive car the lowest point of the chassis, the bottom of the flywheel housing, is much nearer the mid-point between axles than in a front-drive car.

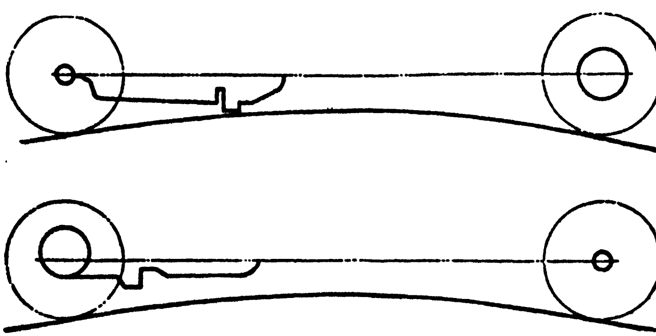


FIG. 2.—SHOWING THAT IN A FRONT-DRIVEN CAR THE FLYWHEEL HOUSING IS LESS LIKELY TO SCRAPE THE CREST OF A HILL.

Power-Plant Arrangement—In order to increase the maximum tractive force available in a front-drive car, the center of gravity must be brought as far forward as possible. In such cars the engine, clutch, transmission and final-drive housing usually form a single block, which is mounted on the frame with the engine toward the rear—instead of toward the front as in conventional cars. Fig. 3 shows such an assembly. It can be readily seen that if all of the units are of the same general design as in a rear-driven car, the hood will have to be a great deal longer, for in addition to covering the engine, it must extend over the clutch and transmission housings, and also at least partly over the final-drive housing.

To reduce the length of hood required, and move the center of gravity forward, a type of engine should be chosen which is short in relation to its piston displacement, such as an eight-cylinder V for large displacements, and a four-cylin-

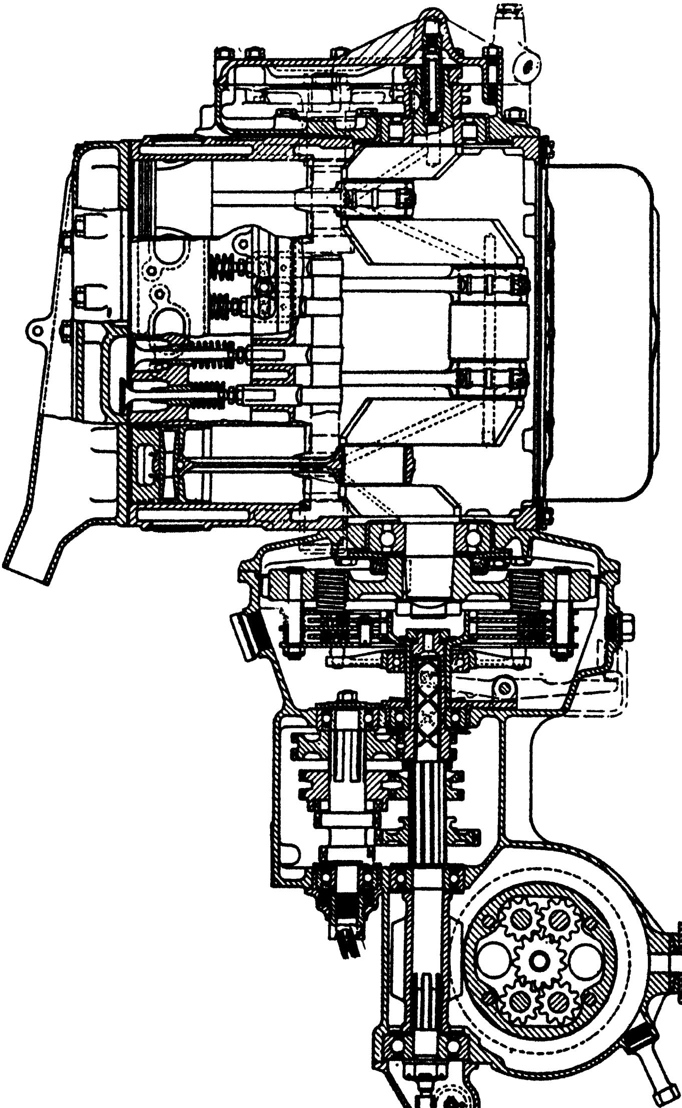


Fig. 3.—Powerplant of the B.S.A. (British Small Arms) Front-Driven Small Car.

der opposed for smaller ones. Then, the transmission, which normally is located between the engine and the final-drive gear, can be placed beyond the final-drive gear, so that it is forward of the front axle. This, of course, requires a special layout, for the power must be transmitted past the final-drive gear to the transmission, and then back again to the final-drive gear. Direct drive is possible with such a layout if worm drive is employed, and probably also with hypoid-gear drive, by moving the ring gear over to one side of the differential so that the housing of the latter will not interfere with the clutch shaft beyond the hypoid pinion.

Where worm drive is used, the worm is bored out and the clutch shaft extends through it. The worm can then be driven either directly, by being secured to the clutch shaft by means of a positive clutch, or indirectly through any of the gear trains of the transmission.

Another possible arrangement dispenses with the direct drive, and transmits the power through one pair of gears in the transmission for each forward speed. This plan was followed by Alvis in England and Citroen in France. Since the low-gear reduction must be obtained in a single step, the center distance between transmission shafts naturally must be somewhat greater than in a conventional transmission designed for the same torque load.

Front-Drive Axles—Both of the American front-drive cars of the early thirties had front axles embodying the same principle as the De Dion rear axle; that is, they comprised a curved carrying member, a center housing supported from the car frame, and jointed axle shafts extending through tubular wheel spindles. In the Ruxton (Fig. 4) the carrying member was of I-section, and located below the center housing, while in the Cord it was tubular and curved around the front of the housing. The I-beam carrying member had the steering head entirely below the steering spindle, while the tubular axle center had backwardly inclined, forked steering heads.

Instead of being mounted on the spindles of steering knuckles pivoted to the ends of a rigid carrying member, the front wheels may be independently suspended. In that case the axle shafts extending from the center housing to the steering heads are enclosed in tubes which are articulated to the center housing and swing in a transverse vertical plane under spring action.

Universal Joints for Front Drives—As the center housing is spring-suspended, each axle shaft must be provided with a universal joint at each end. That at the differential end operates at relatively small angles, while the one at the wheel end occasionally must work at angles up to 35°, and even more. When it is considered that with a conventional universal joint operating at an angle of 35°, if the driving shaft rotates at constant speed, the speed of the driven shaft will fluctuate between 82 per cent and 122 per cent of that speed, it would seem absolutely necessary to use a constant-velocity-type universal joint at the outer end. However, an American make of four-wheel-drive truck has successfully used a conventional universal in this position for several decades, and two makes of British front-driven passenger cars also use such joints there. The reason this can be done probably is that—unlike the propeller shaft of a conventional car—the

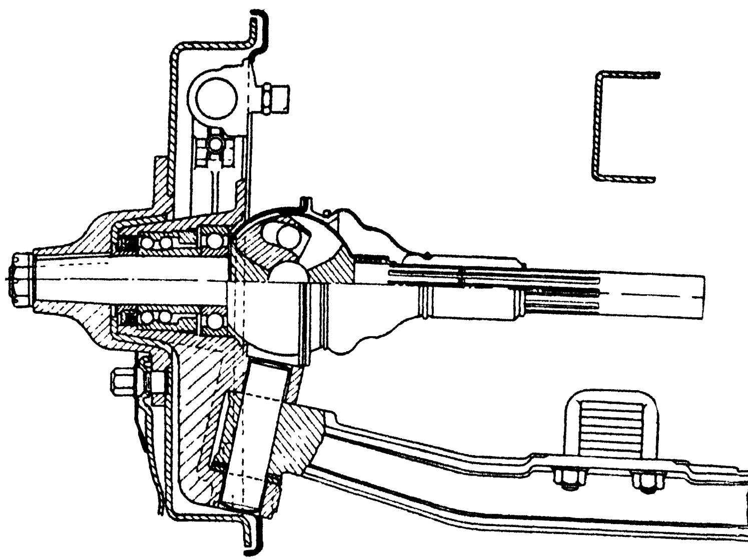


FIG. 4.—FRONT-AXLE END OF RUXTON FRONT-DRIVEN CAR.

axle shaft of a front-driven vehicle is not positively connected to masses of great moment of inertia at both ends. The only mass positively connected to it at its inner end is the differential side gear, and that is small. Owing to the inertia of the car as a whole, and the adhesion between tire and road, on curves the road wheel tends to keep rotating at constant speed, and constant speed of the road wheel implies fluctuating speed of the axle shaft. Now, at least one-half of the time while one axle shaft is being accelerated by the angularity of the shaft at its outer end, the other shaft is being decelerated, and equalization can take place through the differential gear, so that the ring gear (and the engine) can keep revolving at substantially constant speed. But even when no such compensation occurs, one road wheel is connected to the other

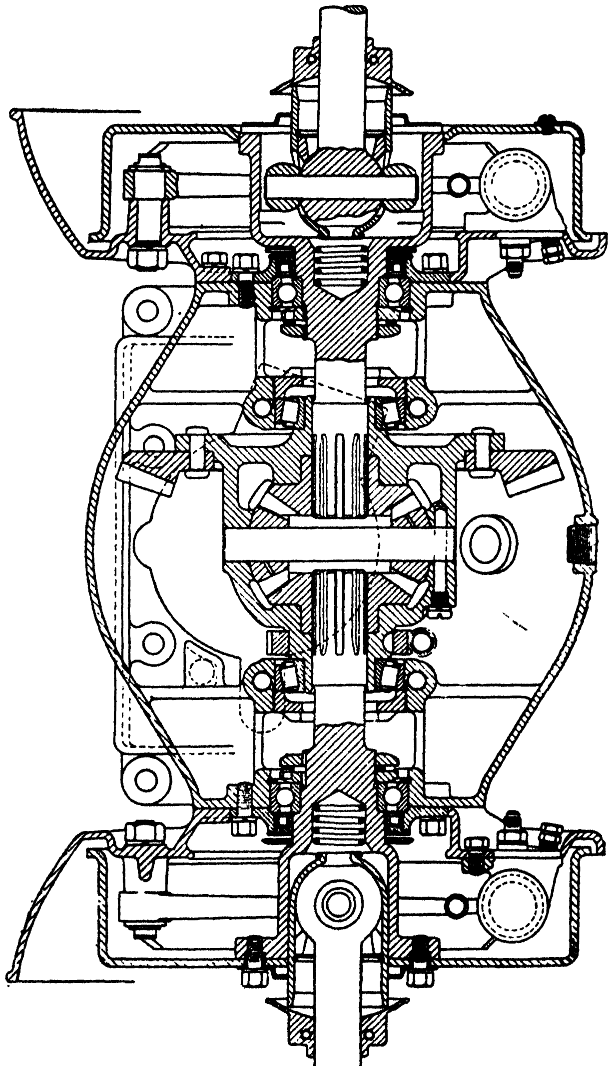


FIG. 5.—SECTION THROUGH CENTER HOUSING OF CORD FRONT DRIVING AXLE, SHOWING BRAKES MOUNTED ON THE HOUSING.

through the two axle shafts, and to the engine through one axle shaft and the propeller shaft, and the torsional flexibility of these shafts, together with any torsional flexibility of the joints (which sometimes are of the flexible disc type), and any circumferential flexibility of the tires will absorb any small irregularities in the angular velocity. However, it is undoubtedly preferable to use universal joints of the constant-velocity type at the steering heads, and practically all of the later designs of front-drive vehicles have them. These joints, moreover, must be of a design permitting of satisfactory operation up to the maximum angle of steering deflection. The joint must be so mounted that its center lies in the knuckle-pin axis, and it must be enclosed in a grease- or oil-retaining housing in two parts capable of sliding over each other around the knuckle-pin axis. The housing usually is a part of the regular universal-joint assembly. There is, of course, much less need for a joint of the constant-velocity type at the inner end of the axle shaft, but this type usually is used there too, probably because it simplifies manufacturing and servicing problems to have only one type.

An incidental advantage of front-wheel drive is that the front wheels may be utilized for braking without mounting brake drums on them, drums so located being undesirable because they increase unsprung weight and promote forms of front-end periodic vibration known as wheel shimmy and wheel wobble. Front-wheel braking can be provided for by placing a brake drum on the propeller shaft or on each of the differential shafts where they emerge from the differential housing. (Fig. 5.)

All-Wheel Drive—All-wheel drive includes four-wheel drive, six-wheel drive, and eight-wheel drive, but four-wheeled vehicles with all wheels driven have been produced in much greater numbers than six- and eight-wheeled with the same feature. All-wheel drive is especially adapted to military trucks and tractors, which frequently must be operated off the beaten paths, and therefore are sometimes referred to as cross-country vehicles. A certain number of four-wheel-drive trucks were developed for World War I, but their design was rather primitive. These vehicles were greatly improved between wars, and they were used in much greater numbers in World War II. The familiar Jeep or General Purpose vehicle has four-wheel drive, and all of the heavier vehicles of the Ordnance Department also were driven on all of their road wheels.

All-wheel drive is adapted also to trucks for use in mining work and for haulage in undeveloped countries. Since the advantage of the all-wheel drive is primarily due to the fact that all of its weight is available for traction purposes, it lends itself particularly for use on vehicles designed to haul one or more trailers behind them. Aside from its greater limiting traction, the all-wheel-driven truck has the advantage that the total load can be divided substantially equally between its wheels, which obviates abnormal loads on any wheel and makes unnecessary a large overhang over either axle. For general haulage on paved roads the use of all-wheel-drive vehicles does not seem to be warranted, for their cost of construction is necessarily higher, their efficiency is slightly lower, and most of their advantages do not come into play there.

All-wheel drive involves two major problems, that of dividing the torque of the engine between the different axles, preferably in the proportion of the weights on them when climbing a steep grade under full load, and that of a combined steering and driving front axle. The latter problem is akin to that of a front axle for front-driven vehicles, but differs therefrom because the housing enclosing the final-drive gear and differential usually is not spring-suspended, but forms an integral part of a rigid axle structure. In an all-wheel-drive vehicle the engine is mounted on the frame in the usual way, with its driving end toward the rear, and the front axle is driven through a propeller shaft, the same as the rear axle. Except for the fact that the wheel spindles, instead of being integral with the axle housing, are pivotally connected thereto, the front axle of a vehicle with all-wheel drive is very similar to a full-floating rear axle.

Drop Gear—In all-wheel-drive trucks the engine generally is mounted in a comparatively high position, so that the propeller shaft for the front axle can pass under it, or under its supporting arms, without interference. This calls for the use of a "drop gear" (also called "transfer gear" or "transfer case") by which the power is transmitted to a lower level. In order to be able to fully utilize the adhesion of the tires in an all-wheel-drive vehicle, the engine must be capable of impressing sufficient torque on the drive wheels to slip, or at least to bring them to a point where slip begins. With a coefficient of adhesion of 0.6, this requires $600r$ lb-ft of torque per 1000 lb of gross vehicle weight. Heavy trucks usually are equipped with engines having a maximum torque of about 13 lb-ft per 1000 lb gvw. However, owing to losses occasioned by the driving of accessories and by friction in the drive, only about 10 lb-ft of the 13 actually developed by the engine reaches the tire treads. The effective radii of truck wheels range between about 1.5 and 1.8 ft, hence the total reduction ratio between engine and road wheels should range between $(600 \times 1.5)/10 = 90$ and $(600 \times 1.8)/10 = 108$. Still higher ratios may be desirable, for the reason that the engine torque gradually decreases in service, and in one case at least a four-wheel-drive truck had a maximum reduction ratio of 130 to 1. Such ratios are much higher than the combined ratio of the usual final drive and the low-speed gear of a four- or five-speed transmission, and it is therefore customary to incorporate a two-speed auxiliary transmission in the transfer case. Toothed chains and sprockets sometimes have been used in transfer cases, but usually either spur gears or spur and herringbone gears are employed. As the "drop" must be of the order of 12 in., an intermediate shaft is used, and the power is transmitted through a three-gear train for high and a four-gear train for low speed, in order to keep down gear diameters and pitch-line velocities. The speed reduction for low gear may be effected either by the first pair of the four-gear train alone, or partly by each of the two pairs.

Three different arrangements of the auxiliary reduction gear in the drop case are possible. In the first place, as shown in Fig. 6, both gears on the upper shaft can be combined in a single unit and arranged to be slid into and out of mesh with their respective gears on the intermediate shaft. This avoids idling of one of the gears on its shaft when the other one is transmitting the power, but is open to the objection that the gears on the intermediate shaft must be sufficiently far apart to accommodate the spool gear between them, with clearance on both sides. To reduce the spacing required to a minimum, it is now customary to chamfer the "clashing" ends of the gears to the depth of the teeth. Alternately, both gears on the upper shaft may be free thereon and provided with clutch jaws so they can be clutched to the shaft by means of a clutch member sliding on a splined section thereof. This makes the most compact design, but each of the gears on the upper shaft idles when the other is driving. It has been learned by experience that idling gears in the upper part of a housing are apt to give trouble when fitted with plain bushings, and they are now generally mounted on antifriction bearings. If the low-speed pinion is arranged to slide on splines on the shaft, into and out of mesh with its gear, and is provided with clutch teeth or clutch jaws by means of which the other gear on the upper shaft can be locked to it, such bearings are needed for one of the gears only.

Drop-Gear Ratios—In the discussion of two-speed axles it was stated that the ratio of the two axle speeds usually is made equal to the square root of the ratio between successive transmission speeds, so that the different road speeds obtainable by means of the transmission and axle change gears form a geometric series; or, alternately, the ratio is made such that when changing the axle gears, the speed obtained is about midway between those next to it in the other axle gear. This calls for ratios between the high and low axle gear of between 1.30 and 1.40. Such a low ratio, however, would not be satisfactory in the case of the transfer gear, because the low gear in the transmission and the regular axle gear together usually give a total reduction ratio of less than 50, and to obtain the required maximum ratio of 100 or more, the reduction ratio

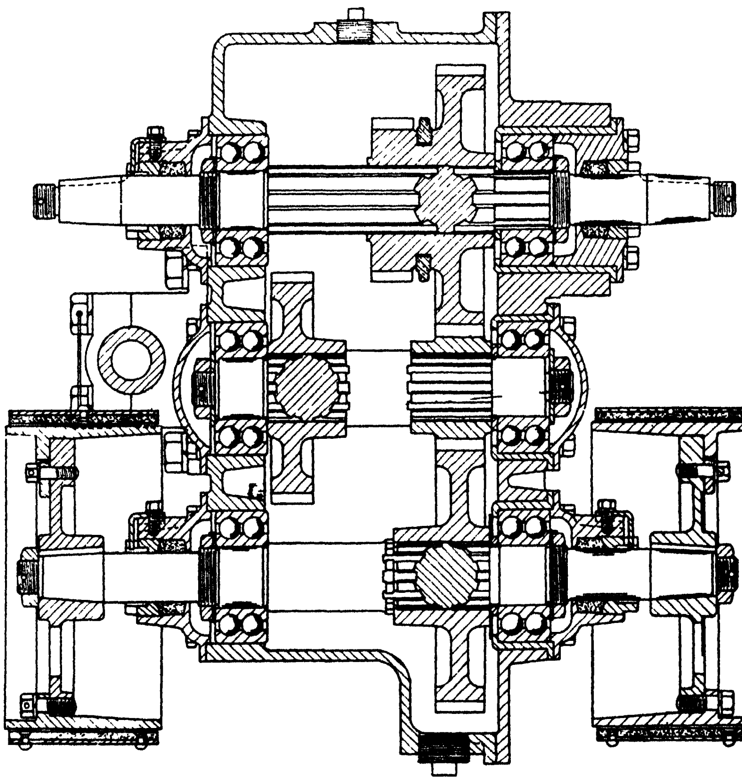


Fig. 6.—Drop Gear of Four-Wheel-Drive Truck.

in the transfer gear must be 2 or higher. As it is undesirable to have two over-all gear ratios which are nearly alike, it is advisable to make the ratio in the transfer case equal to $r^{1.5}$, where r is the average ratio between successive transmission speeds. For instance, if the ratio between any two successive transmission speeds is, say, 1.8, then the ratio between the two transfer-case speeds should be $1.80^{1.5} = 2.40$. The various nominal vehicle speeds then will form a geometric series in which two terms—those next to the end terms—are missing. In the above case the ratio between the highest and next-to-highest, and also the ratio between the lowest and next-to-lowest speed would be 1.8, while the ratio between any other two successive speeds would be 1.34.

Face Width Required—It is difficult to conceive of conditions of operation where the low gear in the transfer box would be used for a considerable portion of the total mileage. It is essentially an emergency gear, used only rarely, and the theoretical full-load stress in the gears of the low-speed train therefore can be made quite high, especially also because in a transfer box the shafts are short and rigid, and usually supported by large, rigidly mounted bearings. The necessary width of face for any of the gears of the low-speed train can be made

$$w = \frac{Fp}{45{,}000} \text{ in.},$$

where F is the maximum pitch-line load on the gear with the transmission in low gear, and p the diametral pitch. The maximum stress in the teeth of the high-speed train then will be less in the proportion of the low-speed ratio. There is appreciable end thrust only on the lower, driven shaft, which is due to changes in the length of the propeller shafts under spring action, and the lower shaft usually is mounted in taper roller bearings. The other two shafts can be mounted in ball bearings of the annular type.

A modern transfer case combining a number of auxiliary devices is shown in Fig. 7. In line with the upper shaft at the rear there is a short power-take-off shaft supported in two roller bearings (the tapered end of the shaft with key and nut is not shown in the illustration). This shaft can be placed in driving connection with the upper main shaft by a positive clutch. There is also a separate short shaft at the forward end of the lower shaft, which can be connected to the latter by means of a positive clutch. The short shaft is piloted in the main shaft and normally, when the vehicle is driven by both its front and rear wheels, is clutched to it. But should it be desired to drive through the rear wheels alone, the forward propeller shaft can be readily disconnected from the drop gear by means of this clutch.

Wheel End of Driving-Steering Axles—Most front axles for trucks with all-wheel drive have rigid axle centers enclosing the final drive gears and axle shafts, and supporting the chassis frame through laminated springs. It is, of course, also

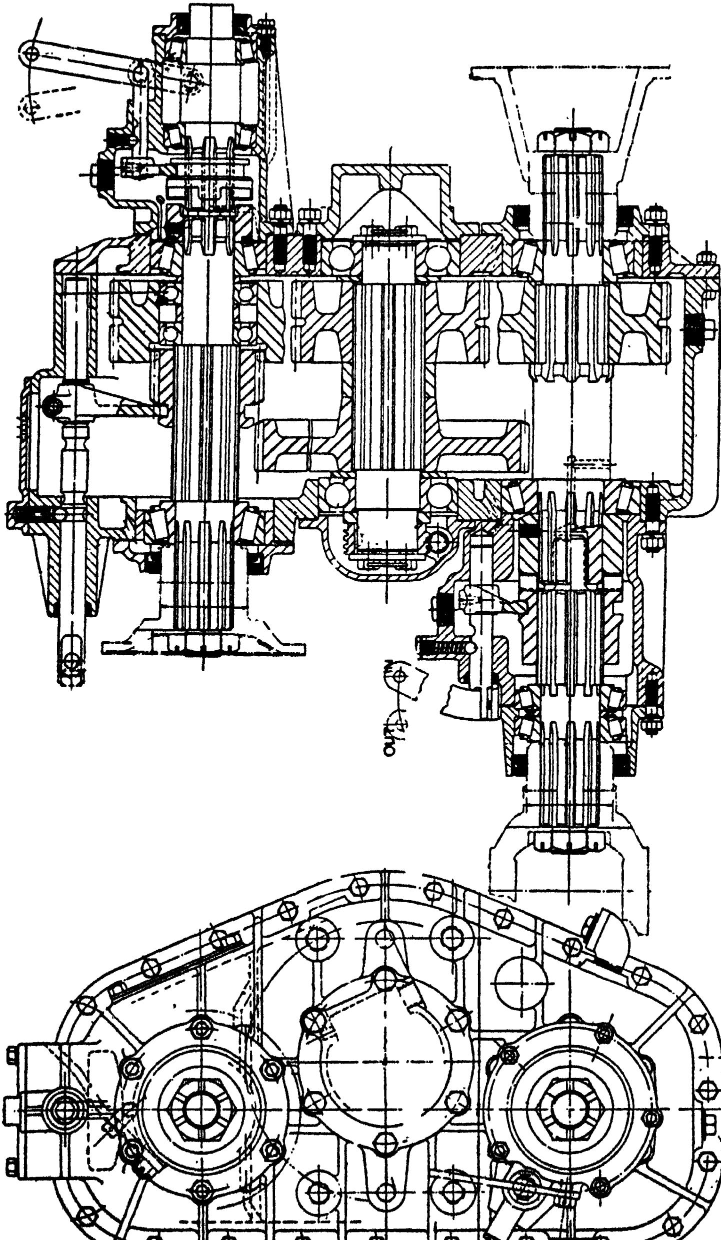


FIG. 7.—WISCONSIN TRANSFER CASE WITH POWER TAKE-OFF AND FRONT-DRIVE CUT-OUT CLUTCH.

possible to use independent suspension for the front end of all-wheel-drive trucks, in which case the final-drive housing must be mounted on the frame. Four-wheel trucks with all wheels driven and with independent suspension on coil springs all around have been built by the Daimler Motor Company in England, and eight-wheel trucks with the same features by the Daimler Motoren Gesellschaft in Germany. Eight-wheel-drive trucks are a comparative rarity, however. In rigid front axles, the axle shafts are supported at their inner end by the side gears of the differential, and at the outer end by the universal joint within the steering head. The universal joint in turn is supported by the stub shaft in the wheel spindle,

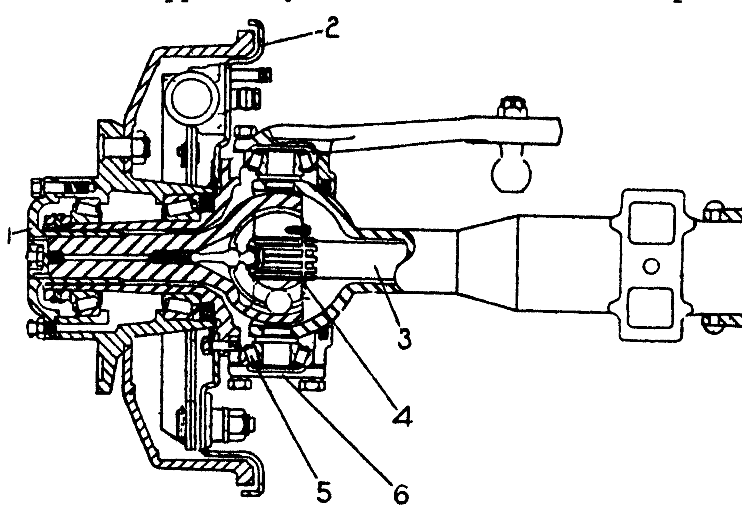


Fig. 8.—Wheel End of Timken Front Driving Axle.

which has a bearing therein at its inner end and is connected to the wheel hub through a driving flange at its outer end.

Wheel ends of two designs of front driving axles for military vehicles are shown in Figs. 8 and 9 respectively, the first being that of a Timken axle built for Ordnance Department vehicles during the thirties, the second that of the front axle for the G.P. vehicle of World War II. Both are equipped with constant-velocity universal joints. The chief difference between the two designs seems to be that while in the former the knuckle pivot axis is vertical, in the latter it is inclined. The wheel driving flanges in both designs are splined to the stub shafts and secured to the wheel hubs by cap screws.

Front-Wheel Drive Through Bevel Gears—The method of driving the road wheel through a stub shaft and universal joint, described in the foregoing, is the one most commonly used. Front wheels, however, can be driven also through trains of bevel gears, the intermediate members of which are mounted concentric with the knuckle pins. Each axle shaft then has a bearing in the end of the axle tube or housing, and carries a bevel pinion immediately beyond the bearing. A similar bevel gear is secured to the wheel hub. These two bevel gears—the driving and driven members of the train—either mesh with the same intermediate bevel pinion, as in the front-wheel drive of the British War Office illustrated in Fig. 10, or they mesh with separate intermediate pinions, as in the Mack design illustrated in Fig. 11. With bevel-gear

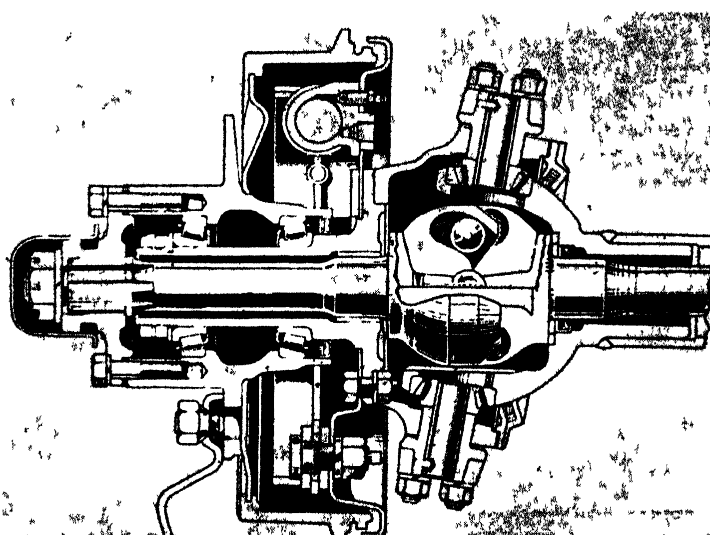


FIG 9—WHEEL END OF DRIVING FRONT AXLE OF WILLYS-BUILT "JEEP"

drive directly at the wheels, a double reduction in speed can be effected there, and if that is done, the first-reduction gears at the center of the axle, the differential, and the axle shaft can be made proportionately lighter. This possibility is taken advantage of in the Mack design, but not in that of the British War Office. The latter, by the way, dates back to the twenties and does not seem to have proven entirely satisfactory, for all of the British all-wheel-drive military vehicles of which descriptions were published during World War II had front-wheel drive through universal joints in the steering knuckles, together with independent suspension on coil springs.

With bevel-gear drive in the steering knuckles, all of the gearing must be enclosed in a housing which will retain the lubricant, and this housing must be made in two or more parts which must slide over each other as the front wheels are

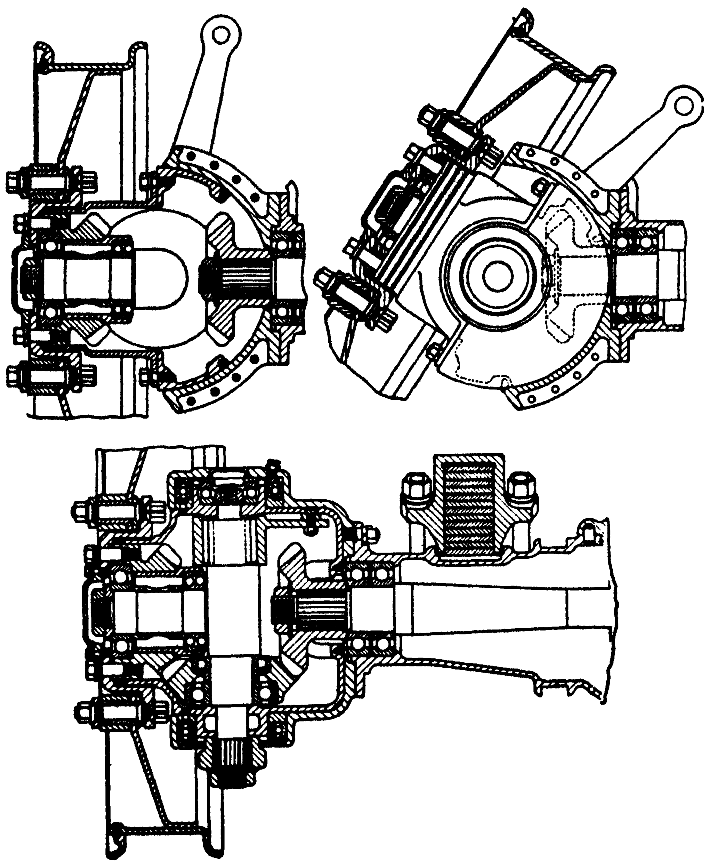


FIG. 10.—FRONT-WHEEL DRIVE THROUGH BEVEL GEARS (EARLY BRITISH WAR OFFICE DESIGN).

turned in steering. One of the gears is secured to the road wheel, and there is therefore another sliding joint between the housing and the hub of the wheel. It hardly needs to be emphasized that the design of such a housing calls for considerable technical skill, for the housing must be inexpensive to produce and easy to install, and it must retain its grease-retaining quality in service. With bevel-gear drive at the front wheels there are no periodic variations in the transmission of motion, regardless of the angular position of the road wheels. Front-wheel drive through bevel gears, however, involves considerable complication, as in addition to the three or four gears per wheel, a considerable number of antifriction bearings are required.

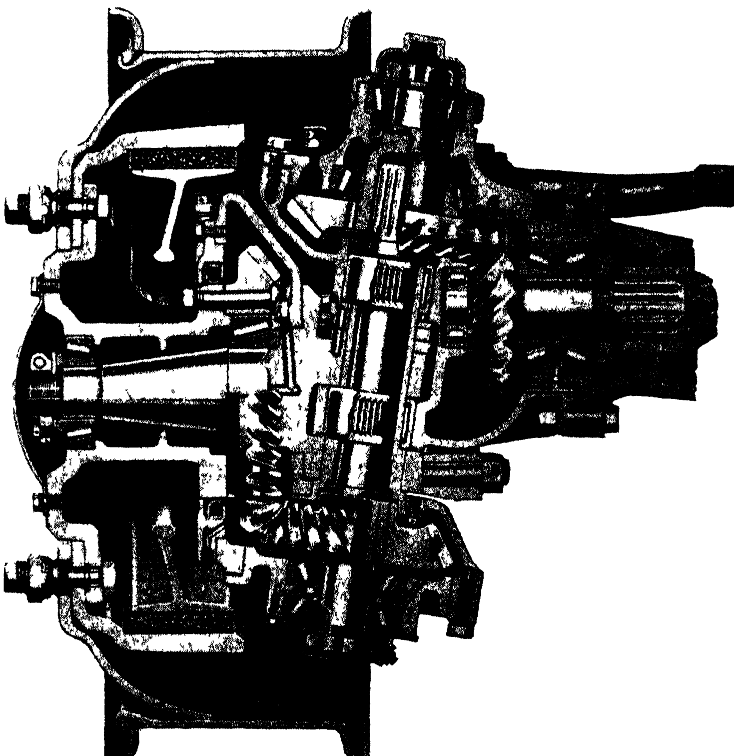


FIG. 11.—MACK FRONT-WHEEL DRIVE THROUGH BEVEL GEARS.

Six-Wheelers—Six-wheeled vehicles were introduced in 1918 when the development of so-called "giant" pneumatic tires first made long-distance transportation of freight over the highways practical. The pioneer work was done by the Goodyear Tire & Rubber Co., which used the trucks to transport materials and rubber goods between Akron, O., and Boston, Mass. These early "giant" pneumatics had only a moderate load capacity, and by using six wheels instead of four, the carrying capacity of the truck could be materially increased. Some time later it was shown experimentally that the maximum road impact produced by a six-wheeler is much less than that produced by a four-wheeler of equal gvw. The stress distribution in the subsoil below the driving wheels of the two types of vehicle is substantially as shown in Fig. 12. From this it was concluded that six-wheelers were less severe on roads, and state legislatures sought to encourage their use by granting them higher maximum weight limits. At present by far the greater part of the freight carried long distances over the highways is moved in six-wheeled units, either tractor-semi-trailer combinations or six-wheeled trucks. In England these two types are known as articulated and rigid six-wheelers.

In Great Britain the use of six-wheeled vehicles was encouraged by a decision of the War Office, some time after the conclusion of World War I, to pay a subsidy to purchasers of six-wheeled trucks meeting certain specifications,

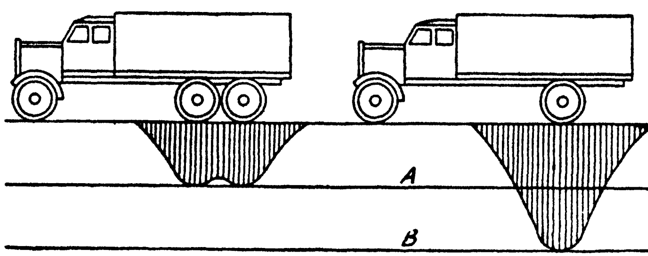


FIG. 12.—STRESS DISTRIBUTION IN ROAD BED UNDER REAR WHEELS OF FOUR-WHEELED AND SIX-WHEELED VEHICLES.

which rendered them suitable for military purposes. A design ensuring great flexibility of the chassis, enabling it to negotiate rough terrain without injury, was worked out and patented by the War Office technical staff, and its use was offered free to manufacturers wishing to manufacture "subsidy lorries." This design, with certain detail improvements, has now become the practical standard for six-wheeled vehicles in that country.

For motor-bus application the six-wheeler has the further advantage that it provides easier riding qualities. Each driving wheel of a six-wheeler naturally weighs much less than one driving wheel of a four-wheeler of equal total weight; hence, when one wheel meets an obstruction and is thrown upward, the reaction on the chassis and body is less in the case of a six-wheeler. Moreover, the system of levers represented by the springs and axles by which the upward movement of the wheel is transmitted to the chassis frame has a greater reduction ratio in the case of the six-wheeler. Of course, the chassis of the six-wheeler receives a larger number of shocks in a given time, but this does not nullify the advantage gained through the reduction of the magnitude of the shocks, for if we could continue to increase the frequency and decrease the magnitude of the shocks indefinitely, we would reach an ideal condition. It is also claimed for six-wheelers that they are less given to skidding than four-wheelers, and this claim seems logical, for there is naturally less chance of four wheels losing their grip on the road at the same time.

Effect of Unequal Wheel Diameters—In a six-wheeled vehicle the two rear axles usually are driven axles. They can be either positively geared together or connected through a differential gear. If the effective radii of the two pairs of driving wheels were exactly equal, there would be no need for a differential; both wheels then would advance along the road at the same speed, and equal road speeds would correspond to equal speeds of rotation. But in practice the wheel radii never are exactly equal, differences being due chiefly to different degrees of tread wear and inflation. Equal rotative speeds will be maintained by the positive gearing, and as the two pairs of wheels necessarily must advance over the road at the same speed, the difference in their diameters will be compensated for by slippage of one pair. When driving at some speed over rough roads, the different wheels successively are thrown upward by road inequalities, and as a result lose their adhesion either partly or completely, and slippage always occurs at the wheel which momentarily has the least adhesion. Only one wheel needs to slip, because of the differential gear between wheels on opposite sides.

The Case for a Third Differential—On rough roads, where one or two wheels are in only light contact with the pavement most of the time, slippage can occur without much power loss and without excessive tire wear, and the need for a differential between the driving axles therefore is greatest in the case of low-speed vehicles operating on comparatively smooth pavements. This may explain why a differential between driving axles was first adopted for six-wheeled buses in service in the city of London. While a differential gear in this location will eliminate tire slip due to unequal wheel diameters, it has the disadvantage that if any one of the four driving wheels gets onto slippery surface, all traction is lost. This, of course, applies to the conventional differential gear which divides the torque applied to it equally between its two side gears or between the shafts connected to these gears. The self-locking differential has been tried out in this location, but apparently has not proved entirely satisfactory either. One reason for this may be that, instead of always dividing the driving torque equally between the two axles, it periodically throws the entire torque on one axle when the wheels of the other lose traction, as when one of these wheels is thrown upward by a bump. That, of course, may result in overstressing the axle shafts. Some use is being made in this location of the "biased" type of differential which, without entirely preventing spinning, enables the axle which has good traction to develop greater torque than the one tending to spin. Two such differentials were described in Chapter VII.

Lateral Tire Slip—In addition to circumferential tire slip due to unequal effective tire radii, there is in all six-wheeled vehicles a certain amount of lateral slippage or tire scuffing when curves are being negotiated. As explained in the chapter on Steering Gears, when two of the axles of a vehicle are held in rigid parallelism, the axes of all wheels produced cannot possibly pass through a single point or a vertical line, and when corners are being turned, some of the wheels necessarily must slip sideways. This slippage can be reduced somewhat by so arranging the design that the rear axles are not held rigidly parallel. For instance, if the axles are connected by springs supporting the chassis frame, on curves the centrifugal force on the spring-supported mass will transfer load from the inner to the outer spring. If the springs have considerable arch, this will shorten the inner and lengthen the outer spring, with the result that the axles will be thrown out of parallelism in such a way as to improve the steering conditions and reduce the scuffing action. It is also obvious that with the axles held in rigid parallelism, a six-wheeler requires a greater proportion of the total weight on the steering wheels than a four-wheeler for positive steering under adverse conditions. To reduce the scuffing action, the distance between rear axle centers should be made as small as practical; it is usually made from 6 to 10 in. greater than the tire diameter.

Driving Methods—The design of a six-wheeled chassis involves three major problems, namely, the method of transmitting the driving torque from the powerplant to the two axles, the method of transmitting the driving thrust from the axles to the frame and taking care of the driving and braking torques, and the method of supporting the chassis frame on the two driving axles. The last two problems, of course, are interrelated, as the chassis springs usually function also as drive and torque members.

The first six-wheeled vehicles developed in this country had worm drive, and this drive is still largely used for the purpose in Great Britain. A side view and a half-plan of the Safeway bus chassis, which was based on the development work initiated by Goodyear, are shown in Fig. 13. The worms were under-mounted and connected by an intermediate shaft with two universal joints. Torque and brake reactions were taken up on two telescoping tubes, one connected to each axle through a vertical pivot joint. The driving thrust was taken on

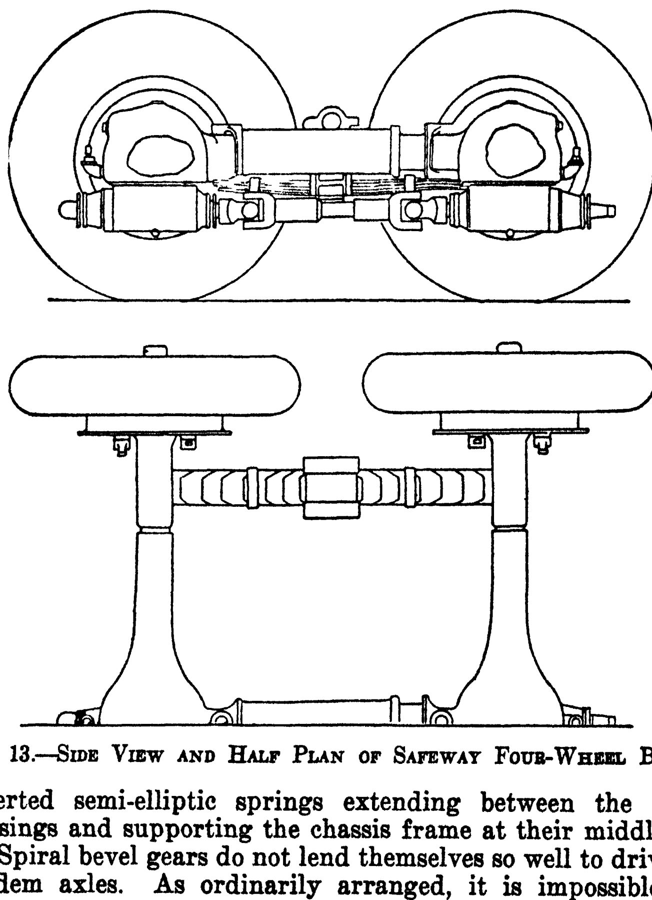


FIG. 13.—SIDE VIEW AND HALF PLAN OF SAFEWAY FOUR-WHEEL BOGIE.

inverted semi-elliptic springs extending between the axle housings and supporting the chassis frame at their middle.

Spiral bevel gears do not lend themselves so well to driving tandem axles. As ordinarily arranged, it is impossible to connect the pinion of the forward axle to that of the rear axle through an intermediate shaft, because of interference with

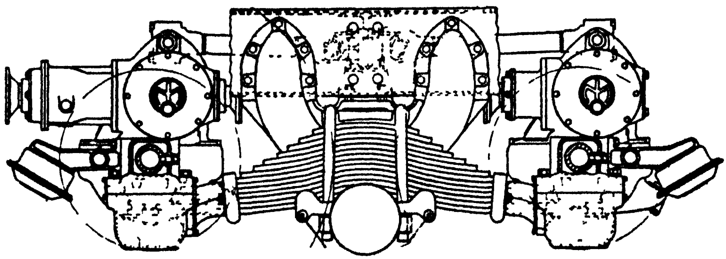


FIG 14.—SIDE VIEW OF MACK FOUR-WHEEL BOGIE

the differential of the forward axle, or its axle shafts. There are, however, at least three methods by which spiral bevel gears can be used for a tandem drive. With a double-reduction drive having a first reduction by spiral bevels, if the shaft of the bevel gear is located well above the differential shaft, and the gear sufficiently to one side of the center plane of the differential, a shaft can be carried back from the spiral bevel pinion of the forward axle without interference. In the conventional bevel-spur double-reduction drive the cone center of the bevel gear is within the spur pinion, but in a bevel-spur double-reduction drive for tandem axles the bevel gear is turned the other way around, so that the bevel-pinion axis is considerably offset from the center plane of the differential. This arrangement is used in the drive of the Mack six-wheel truck, the driving bogie of which is illustrated in Figs. 14 and 15. There the chassis frame is supported on inverted semi-elliptic springs clipped to spring seats oscillating on a trunnion held in spring hangers depending from the frame side

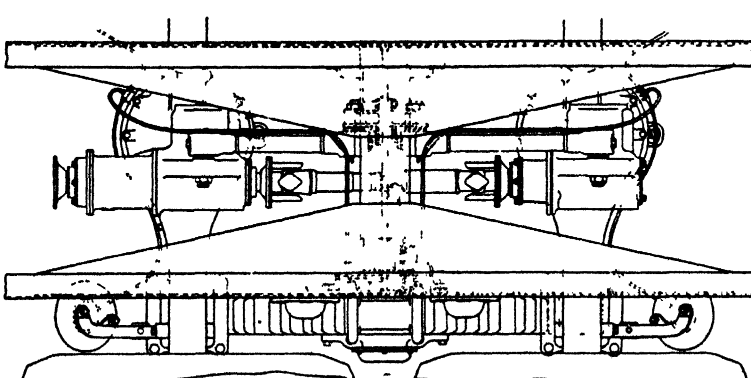


FIG. 15.—PLAN OF MACK FOUR-WHEEL BOGIE.

rails. These springs connect to the axle housings through rubber shock insulators. Driving and braking torque is taken or the frame through short torque rods extending from a bracket on top of each axle housing to ball joints on a heavy frame cross member.

Another method consists in dividing the torque immediately behind the transmission, by means of a spur or herringbone pinion meshing with gears on opposite sides of it, and connecting each of these driven gears to one of the axles through a divided propeller shaft. In one six-wheeled vehicle having this general layout, the final drive to each axle is of the torque-tube type, the rear section of the propeller shaft being surrounded by a torque tube with a spherical forward end supported in a ball socket on the frame. The differential housings on the two axles are equally offset from the longitudinal center line of the vehicle, to opposite sides. While this is a straight-forward design embodying no untried features, duplication of the long propeller shaft and the extra reduction or power-dividing gears running at high speed are not very desirable.

While with a coventional spiral bevel-gear drive to the forward one of tandem axles, the pinion of the rear axle cannot be directly connected to that of the forward axle through a shaft, it can be placed in driving relation with it by providing an extra pinion in the gear housing of the forward axle, meshing with the bevel gear therein back of the differential, and connecting this pinion to that of the rear axle through an intermediate jointed shaft. Theoretically this plan is applicable to single-reduction axles, but the only application of which the author knows is to a double-reduction axle in which the final reduction is effected by spur gears on the inner side of the wheels. Both axles have forged carrying members of I-section forming a banjo at the center, to which the housing and cover for the first-reduction gears are bolted.

British War Office Design—Reference has been made already to the six-wheel-drive construction of the British War Office, and this is illustrated in Fig. 16. One feature is that the torque on each axle is taken on a short torque rod extending from a bracket on the axle housing to a frame cross member located between the axles. There are two pairs of half-elliptic springs in this six-wheeler, which virtually form parallel links. These springs are pivotally mounted on frame brackets, and the whole structure is very flexible, enabling the vehicle to accommodate itself to very uneven road surfaces. The difference in the levels of the two axles may attain 9 in., and the angles between the two worm shafts and the intermediate shaft, 26°, hence universal joints capable of operating at large angles must be used.

Mention may be made here also of a quite unusual six-wheel construction brought out in Great Britain during the twenties by Scammell Lorries, Ltd. In this there was only a single rear axle, the housing of which at each end supported

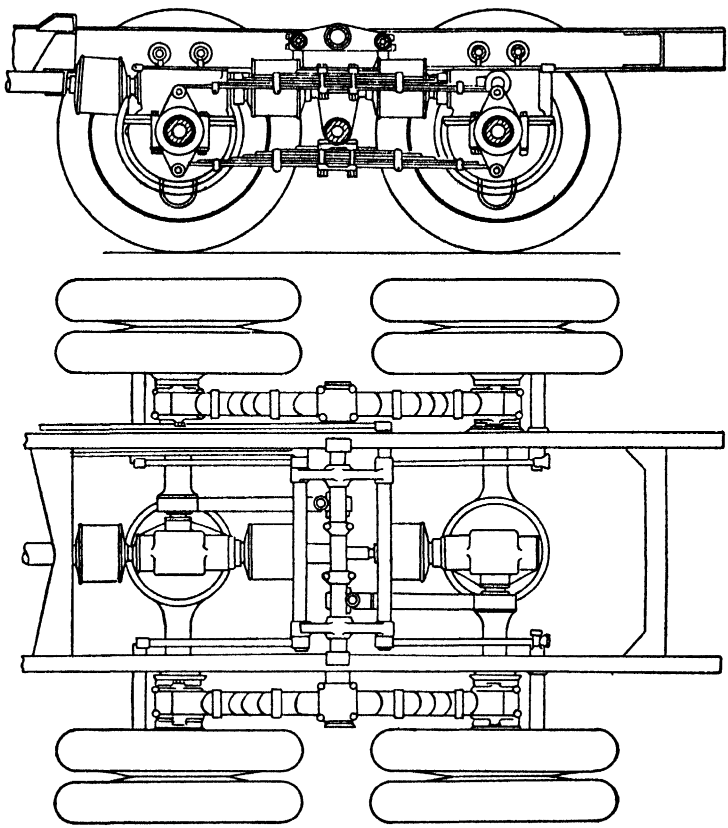


FIG. 16.—SIDE VIEW AND PLAN OF BRITISH WAR OFFICE FOUR-WHEEL BOGIE.

a well-braced, swiveling gear housing extending fore and aft. Stub axles for the two driving wheels on each side were mounted on ball bearings in the housing, and were driven through gear trains from the axle shaft. An axle of this type naturally is very flexible, and would appear to be well suited for cross-country operation. One other advantage claimed for it was that the wheels on each side always remain in the same plane, which improves the operating conditions if a chain track or non-skid band were to be applied to the wheels.

Weight Transfer in Six-Wheelers—There are two distinct methods of taking care of the torque reaction in six-wheelers. Torque arms on the two axle housings either may have their ends supported on the frame, or the torque arm on one may have its point of support on the other axle housing. As axle torque always tends to transfer weight from front to rear when the vehicle is being driven, and from rear to front when it is being braked, the question arises whether and how this weight transfer is affected by different arrangements of the torque members.

As regards the amount of weight transferred from the rear to the front wheels by the application of the brakes, the method of supporting the torque member makes no difference whatever. As shown in Fig. 17, when a vehicle is being braked it is acted upon by two forces, the inertia force acting at its center of gravity urging it forward, and the frictional force

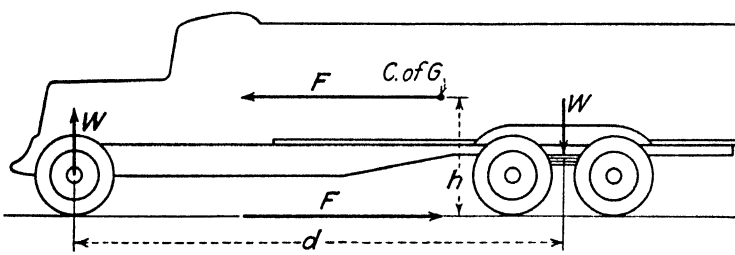


Fig. 17.—Diagram of Weight Transfer Under Brake Action in a Six-Wheeler.

between the tire treads and the pavement holding it back. These two forces are equal, and they form a couple which for the condition of equilibrium is held in balance by another, opposite couple. The forces producing the balancing couple act through the chassis springs, and each of them is equal to the weight removed from the rear springs and added to that on the front springs. The arm of this couple is equal to the distance between front and rear spring centers. Referring to Fig. 17, if we designate the retarding force at the road surface by F, the height of the center of gravity by h, the change in road reaction due to weight transferred from rear to front by W, and the distance between spring centers by d, then $Fh = Wd$, and the weight transferred, $W = Fh/d$.

But while the amount of weight transferred from rear to front is always the same, regardless of the method of taking up the brake-torque reaction, the loss in weight on both rear axles need not be the same. Where both torque arms are supported by the frame, the reduction of the weight on both rear axles will undoubtedly be substantially the same. On the other hand, if the two axle housings are braced against each other, either by telescoping torque tubes or by parallel-link type of torque members, a very considerable amount of weight is transferred from the rear to the forward driving axle when the brakes are applied. If we designate the load on the two driving axles by L, the maximum retarding force which their brakes can produce is approximately $0.6L$, and if the effective wheel radius is designated by r, the braking torque on the four driving wheels is $0.6Lr$. If the distance between driving-axle centers is e, then the weight transferred by the braking torque from the rear to the forward driving axle is $0.6L(r/e)$. In practice the value of r/e is usually about 0.4, hence the amount of weight transferred from the rear to the forward driving axle when the brakes are applied practically to the locking point is $0.24L$ lb. Here L represents the weight normally on the driving axles less that which is transferred from the driving axles to the steering axle by brake application.

As regards weight transfer due to driving torque, if the torque members are supported on the chassis frame, weight is transferred from the front to the rear wheels the same as in a four-wheeled vehicle. If, on the other hand, the torque reaction is taken up within the driving bogie, there would be absolutely no weight transfer from the front to the rear wheels if the only resistance to motion were the traction resistance, which acts in direct opposition to the propelling force, so that there is no couple. However, there is always some air resistance, which acts through the center of pressure, and together with the propelling force overcoming it forms a couple. Such a couple naturally creates a balancing couple, which means a transfer of weight, but under the driving conditions when the axle torque is really high (low-gear operation), the speed is always quite low, and the air resistance therefore negligible. Hence, for practical purposes we may say that with the torque taken up within the driving bogie, there is no appreciable weight transfer from the front to the rear wheels due to the driving torque. There is, however, a transfer of weight from the forward to the rear driving wheels, which for a given axle torque is the same as the weight transfer in the opposite direction under brake application.

Spring Suspension of Rear Ends—There are many possible arrangements of the rear suspension for six-wheelers. Some of these are specially adapted for use on vehicles intended for operation on rough terrain, which require great flexibility of the chassis. In the following list of suspension systems embodying laminated springs, which have been either actually used or proposed for the purpose, the description applies to the suspension members at one side of the bogie, those on the other side being identical:

1. Semi-elliptic springs on both axles, articulated to frame in front and shackled at rear.
2. Semi-elliptic springs on both axles, articulated to a common bracket at their near ends and shackled to the frame at their far ends.
3. Semi-elliptic springs on both axles, shackled to an equalizer lever at their near ends and articulated to the frame at their far ends.
4. Semi-elliptic springs on both axles, front ends articulated to frame, rear ends connected to bellcranks fulcrumed on frame, whose other arms are connected by a link.
5. Inverted semi-elliptic spring extending between brackets on axle housings and swiveled on a frame bracket at its middle. Axle housings also connected by links.
6. Two superposed inverted semi-elliptic springs connecting brackets on top of and below axle housings, respectively, and swiveled on frame at their middle.
7. Reach bar between axle housings supporting a semi-elliptic spring at its middle.
8. Inverted semi-elliptic spring having a trunnion support on the frame at its middle and bearing against pressure plates on the axle housings with its ends, torque and thrust being taken on four links between a frame bracket and the axle housings.

CHAPTER XII

Worm-Gear Drive

Worm-gear drive is not much used for automotive purposes in the United States at present, but it is the standard drive for public service vehicles in Great Britain, where it is being used also for heavy motor trucks. In the past it has been used for passenger cars to a considerable extent, but in that field it has been losing ground to the spiral bevel and hypoid gear drives. Worm-gear drive has been used by the Ford Motor Company in its Fordson farm tractors for many years, and it predominates in the field of industrial trucks and tractors.

Up to about half a century ago the worm and gear were considered merely a means of transmitting motion, as distinct from a means of transmitting power. As would be expected, with teeth inaccurately cut, worm and wheel crudely mounted, and little or no lubricant on the tooth contact surfaces, the efficiency was low, and wear was rapid. Moreover, the gear was not reversible; that is, power could not be transmitted from the gear to the worm. Worm gearing was first developed for power-transmission purposes in connection with electric motors. These were the first high-speed motors to come into practical use, and high reduction ratios were needed for many applications, such as that to elevators and hoists. Worm drive was first applied to motor vehicles in England, by F. W. Lanchester and the Dennis Brothers.

Advantages of Worm Drive.—The chief advantages of worm drive are that it ensures absolute silence of operation under all conditions, and that it permits of a larger reduction ratio in a single step than is possible with any other known form of axle drive. Its silent operation was the chief reason for its original adoption for passenger-car drives, at a time when the only other types available were the roller chain and straight bevel gears, both of which were rather noisy under certain conditions. With the introduction of spiral bevel gears this advantage of the worm drive lost considerable importance, but the latter still retained the advantage of the possibility of a large reduction in a single step, and at present it is used chiefly in vehicles and tractors with a final-drive ratio of more than 6 to 1. The greater simplicity of a single-reduction gear as compared with a double-reduction one is an important advantage from the commercial point of view, as it permits of lower cost of manufacture. The efficiency of the worm drive is somewhat lower than that of a comparable single-reduction drive of the bevel-gear or chain type, but at least equal to that of a double-reduction mechanism, especially at light loads. This statement must be taken as applying to the respective types of transmission in a general way, and not

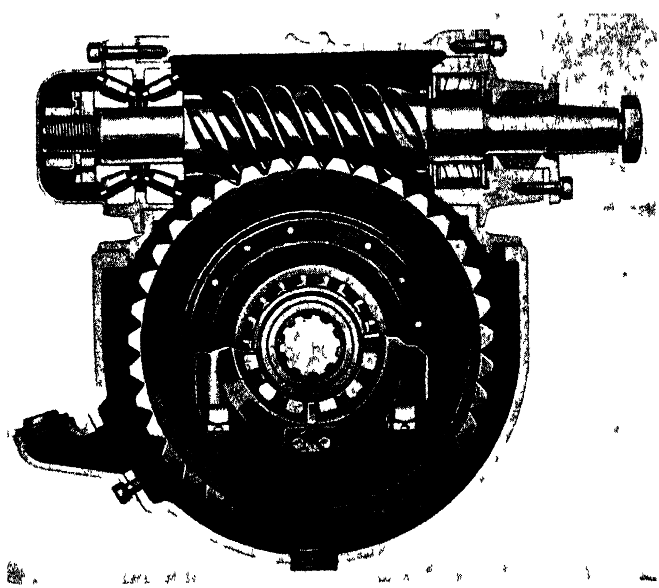


Fig 1.—Automotive-Type Worm and Wheel, Over-Mounted.

necessarily to any specific examples, for much depends upon details of design. With worm drive, for instance, there are three distinct sources of loss, viz., tooth-contact sliding friction, oil churning in the gear housing, and bearing friction, and each of these losses is independent of the others.

Finally, the worm gear has the advantage that it renders it possible to provide a wide range of gear ratios without changing the distance between the axes of worm and wheel, and without changes in adjacent parts. The worm drive gives a symmetrical rear axle, which is comparatively easy to assemble. A worm and wheel of the automotive type are shown in Fig. 1.

Materials for Worm and Wheel—The worm is invariably made of case-hardening steel. Where there is plenty of material below the teeth, carbon steel (No. 1015 or 1020) will do, but the nickel steels Nos. 2315 and 2320 are extensively used, as they give a stronger core. Case hardening is effected after the teeth are cut. This process will distort the teeth more or less, and they are therefore corrected by grinding, which can be readily done on special worm-grinding machines.

The wheel presents a decidedly more difficult material problem than does the worm. It might be thought that, since the teeth of worm and wheel are always separated by a film of oil, it would not make much difference of what material they were made, provided they had sufficient strength to withstand the tooth loads. This view, however, has proved to be erroneous, and the reason is believed to be that different metals have different capacities for maintaining the oil film. It is also possible that, especially with over-mounted worms, there is practically no oil on the teeth when the vehicle is first being started after a long rest, and that some metals will withstand momentary operation without an oil film better than others.

The blanks for worm gears are always made of bronze, and three different bronzes are in use for the purpose. One of these is S.A.E. standard phosphor bronze for gears (Specification No. 65), which is composed of 88-90 per cent copper, 10-12 per cent tin, 0.10 to 0.30 per cent phosphorus, and lead, zinc and other impurities not exceeding 0.50 per cent. Castings made of this alloy should show an ultimate strength of 35,000 psi, an elastic limit of 20,000 psi and an elongation of 10 per cent in 2 in. or proportionate gauge length. In order to increase the strength and hardness of the gears, the blanks are chill-cast, and when thus cast the above bronze will show a Brinell hardness of 75-90. A similar specification is largely used in England (87.7 copper, 12.0 tin, and 0.3 phosphorus).

The second material is aluminum bronze, composed of 89 per cent copper, 10 per cent aluminum and 1 per cent iron. This alloy is not chill-cast, but is subjected to a heat treatment which imparts the following physical properties: Ultimate strength, 80,000-90,000 psi; elongation, 4-10 per cent, and Brinell hardness, 170-200. This aluminum bronze can be die-cast, but, unfortunately, in most applications of worm gearing the production is not sufficiently large to make this possibility commercially available.

The third material is a nickel bronze of the following approximate composition: Copper, 88.50 per cent; tin, 10 per cent; nickel, 1 per cent; lead, 0.25 per cent and phosphorus, 0.25 per cent. This alloy is chill-cast, and in that condition shows an ultimate strength of more than 40,000 psi, an elongation of 4 per cent, and a Brinell hardness of 90-110.

Great efforts have been made to improve the physical properties of worm gear bronzes by improvement in foundry methods. The blanks are now usually east with chills on three sides, and in England the centrifugal casting method is used to increase the density of the metal. In some cases the blanks are cast with the teeth practically formed in the mold, leaving only a small amount of metal to be removed in the gear-cutting process. This not only speeds the gear-cutting operation, but retains more of the effect of the chills on the surfaces of the teeth.

Theory of Worm Gearing—The worm gear as applied in motor-vehicle drives is similar to a helical gear, the worm being always of the multiple-thread type, and some of the rules of helical gearing therefore also apply to worm gear. In a worm and worm wheel the gear reduction is equal to the quotient of the number of teeth in the worm wheel by the number of threads in the worm. The lead of the worm is the distance in the direction of the worm axis corresponding to one complete revolution of the worm thread. The lead angle is the angle made by the pitch line of the worm thread with a plane perpendicular to the worm axis (also the angle made by a worm wheel pitch line with the worm wheel axis). In connection with helical gears it is customary to speak of the spiral angle, which is the angle made by the pitch element of the gear tooth with the gear axis, and in case the two axes are at right angles to each other (as in a worm and wheel) the spiral angles for the two gears together make a right angle. The lead angle of the worm is equal to the spiral angle for the worm wheel, while the complement of the lead angle is the spiral angle for the worm.

Following are definitions of some terms used in connection with worm gearing:

Circular pitch (of wheel)	= (Pitch diameter × 3.1416)/No. of teeth.
Axial pitch (of worm)	= Lead/No. of threads.
Circular pitch of wheel	= Axial pitch of worm.
Normal circular pitch	= Circular pitch × cos of lead angle.
Normal circular pitch	= 3.1416/normal diametral pitch.

In calculating worms and worm wheels the following equations may be used:

Worm

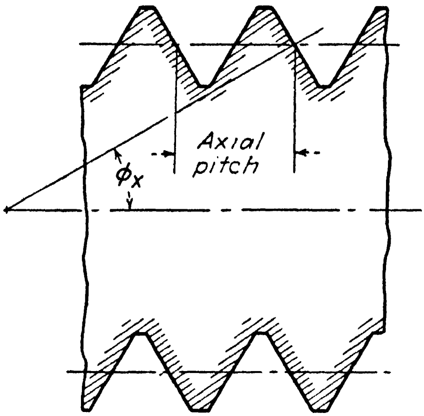
Pitch diameter = (No. of threads $\times$ axial pitch)/3.1416 $\times$ tan of lead angle).
Lead = Pitch diameter $\times$ 3.1416 $\times$ tan of lead angle.
Addendum = Normal circular pitch/3.1416.
Dedendum = Normal circular pitch/2.716.
Outside diameter = Pitch diameter + (2 $\times$ addendum).
Root diameter = Pitch diameter − (2 $\times$ dedendum).

Wheel

Pitch diameter = (No. of teeth $\times$ circular pitch)/3.1416.
Lead = (Pitch diameter $\times$ 3.1416)/tan of lead angle.
Addendum and dedendum, same as for worm.
Throat diameter = Pitch diameter + (2 $\times$ addendum).

The center distance, that is, the distance between the axes of worm and wheel, is equal to one-half the sum of the pitch diameters. Worm and wheel must be cut both with either right-hand or left-hand threads. In an automotive drive, with the engine mounted in front and turning right-handedly (looked at from the front), both worm and wheel must have right-hand threads when the worm is on top of the wheel (over-mounted), and left-hand threads when the worm is at the bottom (under-mounted).

If we cut a very thin section perpendicular to the axis from the middle of the worm wheel, we have a spur gear. If we cut a similar section from the worm we have a rack, or a double rack, Fig. 2, and since the flanks of a rack tooth in the standard involute system are plane surfaces, the edges of the teeth in the axial section are straight lines. Worms with teeth of this form are used for industrial purposes, and at one time were used also for automotive purposes. However, a different type of worm tooth, known as a helicoidal involute tooth, is now being used, at least preponderantly, for automotive applications. There is also another form of worm, known as the hour-glass type, in which pitch lines, instead of being straight as in a rack, are circular arcs concentric with the wheel axis. Both of these types of worm gear will be discussed in some detail further on. It will suffice here to point out that the general theory of worm gearing, which is based



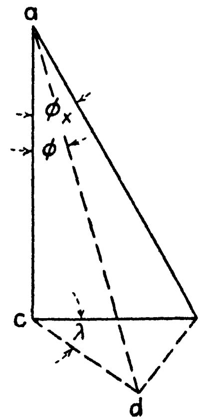


Fig. 2 (*left*).—Axial Section of Worm.
Fig. 3 (*right*).—Showing Relation Between Pressure Angles.

on conditions in a plane containing the worm axis and perpendicular to the wheel axis, applies to all types.

Notation—In discussing the properties of worm gearing we will make use of the following notation:

T, torque on gear, lb-in.;
F_N, tangential force on pitch circle of gear, lb;
F_n, tooth load normal to helix, lb;
F, normal tooth load, lb;
F_t, tangential force on pitch circle of worm, lb;
d, pitch diameter of worm, in.;
D, pitch diameter of gear, in.;
n, worm speed in rpm;
N_W, number of worm threads;
N_G, number of gear teeth;
r, speed-reduction ratio $= N_G/N_W$;
F_r, radial component of tooth load, lb;
λ, lead angle of worm;
ϕ, normal pressure angle;
ϕ_x, radial pressure angle;
ϕ_t, transverse pressure angle;
l_a, thrust load on worm bearings, lb;
l_r, radial load on worm bearings, lb;
L_a, thrust load on gear bearings, lb;
L_r, radial load on gear bearings, lb.

Pressure Angles—In worm gearing we have to deal with three distinct pressure angles—the normal pressure angle ϕ, the axial pressure angle ϕ_x, and the transverse pressure angle

ϕ_t. The pressure angle in any section—normal, axial or transverse—is the angle between either the pitch line (if straight) or a tangent to the pitch curve and a line normal to the tooth profile at the pitch point. The relation between the normal pressure angle ϕ and the axial pressure angle ϕ_x of the worm is illustrated in Fig. 3. In the drawing the line *cd* is supposed to be normal to *ac* and not in the plane of the paper. The lead angle is represented by λ. It will be seen that

$$\tan \phi_x = \frac{bc}{ac}$$

$$\tan \phi = \frac{cd}{ac}$$

$$cd = cb \cos \lambda.$$

Substituting this value of *cd* in the preceding equation we get

$$\tan \phi = \frac{bc \cos \lambda}{ac} = \tan \phi_x \cos \lambda;$$

that is, the tangent of the normal pressure angle is equal to the tangent of the axial pressure angle multiplied by the cosine of the lead angle. The lead angle is found from the equation

$$\tan \lambda = l/(\pi d),$$

where l is the lead of the worm and d the worm pitch diameter, both in inches. An axial pressure angle of 30° has been widely used. With a lead angle of 35° this corresponds to a normal pressure angle of 25° 19′. As in the first quadrant the value of the cosine decreases as the angle increases, it is apparent that for a given axial pressure angle ϕ_x, the normal pressure angle is the smaller the greater the lead angle λ. This places a lower limit on the axial pressure angle for high lead angles, as with too small a normal pressure angle there will be interference and undercutting, which reduces the effective tooth contact surface. With the hour-glass type of worm, moreover, there arises the further difficulty that the worm cannot be assembled with the wheel if the pressure angle is too small.

Center Distance—As the worm pitch diameter

$$d = \frac{np}{\pi \tan \lambda},$$

and the wheel pitch diameter

$$D = \frac{Np}{\pi},$$

and as the center distance C is equal to one-half the sum of the two pitch diameters, we have

$$C = \frac{\dfrac{np}{\pi \tan \lambda} + \dfrac{Np}{\pi}}{2} = \frac{p}{2\pi}\left(\frac{n}{\tan \lambda} + N\right).$$

The center distance, of course, bears a close relation to the maximum power or torque which can be safely transmitted by the worm and gear. With spur gears torques proportional to the cube of the center distance result in equal bending stresses. In worm gearing, however, it is not so much the bending stress on the teeth as the ability to get rid of the heat generated by friction at the tooth contact surfaces that limits the torque capacity. With a given efficiency, the heat loss varies directly with the power transmitted, and at a constant speed it varies directly as the torque. But the rate at which the gears can get rid of waste heat varies directly as their surface area, and therefore as the square of any linear dimension, such as the center distance. On the other hand, the normal speed is not independent of the center distance, but varies inversely thereto. Therefore, while considerations of mechanical strength alone would suggest a torque capacity proportional to the cube of the center distance, and considerations of heat-radiating capacity alone a torque capacity proportional to the square of the center distance, the truth lies midway between, and it is found that the actual torque capacity is closely proportional to the 2.5th power of the center distance.

Load Capacity of Worm Gears—An analysis of modern British practice shows that the center distance of worm gears for a particular type of service is made proportional to the 0.4th power of the maximum torque on the rear axle (0.4 being the reciprocal of 2.5). For buses the center distance is made approximately

$$C = T^{0.4}/13.2 \text{ in.},$$

and for trucks,

$$C = T^{0.4}/14.5 \text{ in.}$$

Here T is the maximum rear-axle torque, obtained by multiplying the maximum engine torque in lb-in. by the low-gear ratio of the transmission and by the rear-axle ratio. These equations give the following maximum torque capacities for worm gears with different center distances, in the two applications:

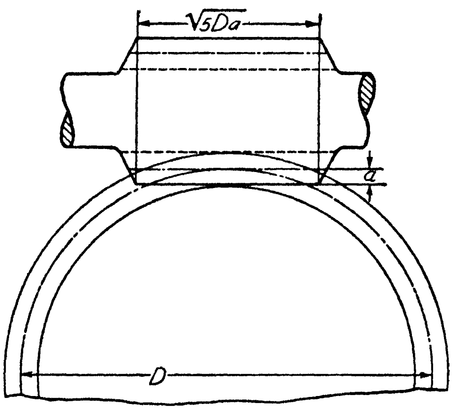
Center distance, C, in.	$6\frac{1}{2}$	7	$7\frac{1}{2}$	8	$8\frac{1}{2}$	9
Max. axle torque (bus)	68,200	82,100	97,500	114,600	133,400	154,000
Max. axle torque (truck)	86,200	100,700	123,000	145,700	168,600	194,500

The chief reason why a worm gear will stand up under a higher maximum rear-axle torque in a truck than in a bus is that trucks usually have a very large low-gear ratio, and are therefore able to develop a very high rear-axle torque for a given engine torque. This maximum torque, however, is used only rarely. Buses have no such large low-gear ratio, and in normal service their average axle torque is a greater per cent of the maximum than is the case in trucks. Consequently, for a given maximum rear-axle torque, the center distance must be greater in a bus than in a truck.

Effects of Speed and Load Factor—While we have here considered the center distance dependent only on the maximum axle torque, actually it should vary also with two other factors. First of these is the worm speed. It is quite evident that a worm gear which is just able to transmit a certain torque at, say, 2000 engine rpm, will not safely transmit the same torque at 4000 rpm. The above torque-capacity or center-distance formulae are based on the fact that in a particular service, such as bus operation, normal engine speeds do not vary materially from one make or one design to another. The speed factor has to be taken into account particularly when worm gears are to be applied to passenger cars, the engines of which usually run at considerably higher speeds than those of commercial vehicles. There the center distance usually is made about $T^{0.4}/12$ in.

The other factor referred to is the average-load factor. For instance, a worm gear may give satisfactory service in a truck when the latter is operated as a separate unit and not required to carry loads in excess of its rated capacity. But if a trailer is hitched to the truck and the total load hauled is greatly increased, it may (and likely will) give trouble. The maximum rear-axle torque is no greater than previously, as neither the maximum engine torque, the low-gear ratio, nor the axle ratio has been changed; but evidently the average rear-axle torque is considerably greater. When either the engine speed is considerably greater than the average in the particular line of application, or the ratio of horse power to gross vehicle weight is considerably lower, the center distance should be made somewhat greater than the value given by the formula, and vice versa.

The limiting load capacity of a worm and gear also depends upon the angle subtended by the wheel teeth at the axis of the worm or the degree to which the wheel teeth envelop the worm. Experiments have shown (what would have been expected) that if the face width of the gear is successively reduced, the drive will fail at lower and lower loads. However, the wheels are now generally so made as to give the largest possible area of tooth contact. According to David Brown & Sons, Ltd., British worm-gear specialists, while the face width should be as great as possible within reason, the throat should not envelop the root of the worm by an amount



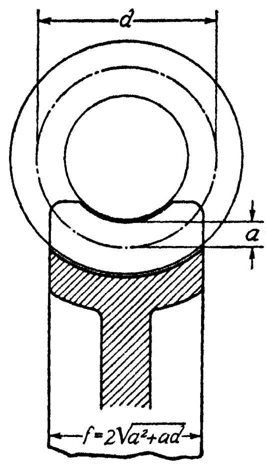


FIG. 4 (*left*).—DIAGRAM FOR LENGTH OF WORM.
FIG. 5 (*right*).—DIAGRAM FOR WIDTH OF WHEEL FACE.

greater than 4.71 $\times$ the normal circular pitch, in order to prevent cutting the wheel teeth to an edge.

Length of Worm and Face of Wheel—It is not advisable to make the worm so long and the wheel face so wide that these parts extend to the very limit of possible tooth contact. A more practical plan is to proportion worm and wheel as indicated in Figs. 4 and 5. This gives for the length of the worm

$$l = \sqrt{5Da},$$

and for the face of the wheel,

$$f = 2\sqrt{a^2 + ad},$$

where a is the addendum; d, the worm pitch diameter and D, the wheel pitch diameter. These rules apply to straight worm sets only.

At one time it was customary to chamfer the toothed portion of the worm-gear rim, the conical surfaces passing through the worm axis. At present the corners of the rim are generally turned square, with a small radius, as shown in Fig. 5, which facilitates machining and gives a larger effective tooth surface.

Rubbing Speed and Normal Pressure—Fig. 6 is a development of the worm pitch surface for a length corresponding to the lead. It is a rectangle of width πd and height l, and it will be seen that the diagonal, which represents the length

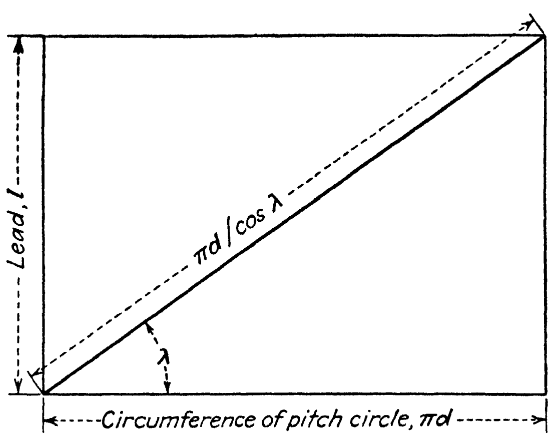


Fig. 6.—Development of Worm Pitch Surface.

of worm thread corresponding to the lead, is equal to $\pi d/\cos \lambda$. During each revolution of the worm a gear tooth slides over this entire length of worm thread, and the rubbing velocity therefore is

$$v = \frac{n\pi d}{12 \cos \lambda} \text{ fpm.}$$

As the pitch line velocity of the worm is $n\pi d/12$ fpm, the rubbing speed is greater than the pitch-line velocity in the proportion of unity to $\cos \lambda$. It is 22 per cent greater for a lead angle of 35° and 30 per cent greater for a lead angle of 40°.

From Fig. 7 it can be seen that the normal tooth load

$$F = F_N/(\cos \lambda \cos \phi).$$

If the coefficient of friction is μ, the friction on the rubbing surface is $F\mu$. From the friction and the sliding velocity the frictional loss can be calculated, and this together with the output found from the tangential force on the gear and the speed of rotation leads to the following expression for the efficiency of the worm gear:

$$\mu = \frac{\cos\phi - \mu\tan\lambda}{\cos\phi + \mu\cot\lambda}.$$

Experiments made in England on a testing machine in which the frictional conditions obtaining in worm gearing are

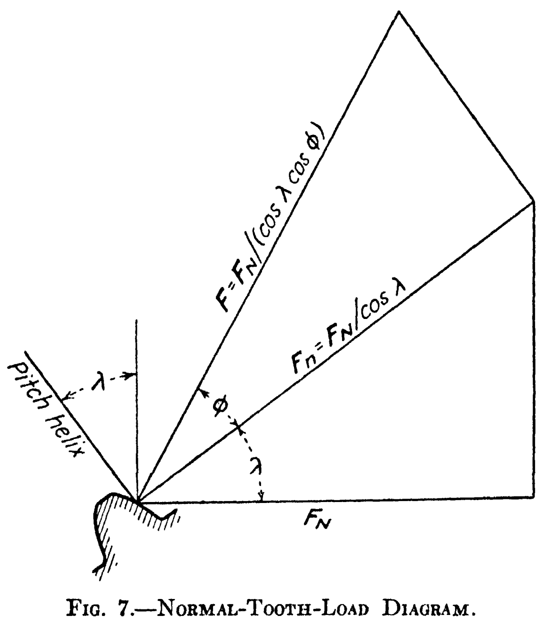


Fig. 7.—Normal-Tooth-Load Diagram.

simulated showed that the friction coefficient varies from about 0.15 at rest to about 0.015 for a rubbing speed of 4000 fpm. From the equation conclusions can be drawn regarding the effect of the lead angle λ on the efficiency. The efficiency is greatest when the lead angle is slightly less than 45°, and if the friction coefficient is low it does not vary perceptibly over the lead-angle range 30°-60°. But as the lead angle approaches 0 and 90°, the efficiency drops rapidly to zero. It should be emphasized, however, that the efficiency here referred to considers only the losses due to tooth friction, and that because there are also losses due to oil churning and bearing friction, the actual efficiencies of worm drives always are lower. It would seem from the foregoing that lead angles close to 45° are most advantageous. However, a smaller lead angle gives a greater worm diameter, and as a fairly large worm diameter is needed for stiffness, lead angles of between 35° and 40° are generally used.

Bearing Loads—The tooth load, that is, the force acting normally on the tooth contact surface, can be resolved into three components at right angles to each other. One of these components acts in a direction parallel with the worm axis,

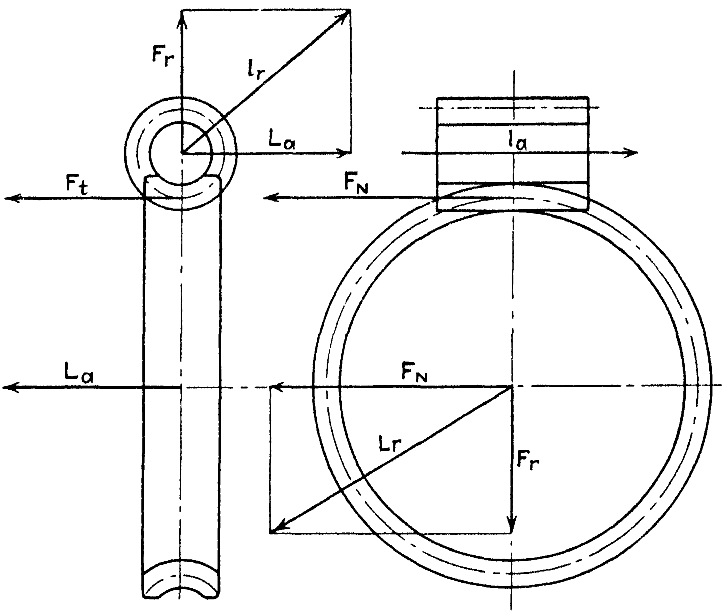


FIG. 8.—BEARING-LOAD DIAGRAMS.

another parallel with the gear axis, and the third normal to both of these axes. The three components, F_N, F_t, and F_r, are shown in Fig. 8. The component F_N is the tangential force on the gear pitch circle and can be readily found from the rear-axle torque by means of the equation

$$F_N = \frac{2T}{D} \text{ lb.}$$

When the worm presses against the gear with a force F_N in the direction indicated in Fig. 8, there is a reaction of equal magnitude on the worm, and this is the worm thrust load l_a. In addition to the tangential force F_N, the worm exerts on the gear a force F_t parallel with the gear axis, and this force creates a thrust load L_a on the gear bearings. F_t, to which L_a is equal, is the tangential force on the worm pitch circle and can be found from the equation

$$F_t = \frac{2T}{dr} \text{ lb.}$$

The component F_r, which tends to force the worm and gear apart, is due to the pressure angle. Its magnitude can be readily determined from a section through the worm axis and normal to the gear axis, Fig. 9. There it is shown that the tangential force F_N acts along the pitch line of the worm section, and the radial force F_r in a direction normal thereto. It

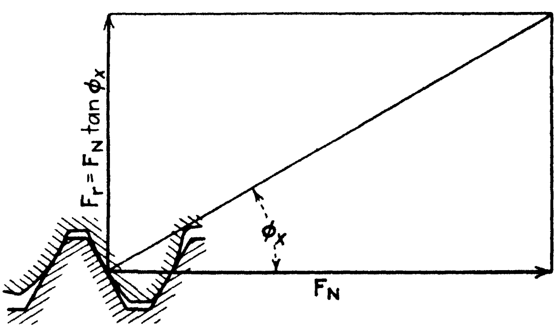


Fig. 9.—Radial-Load Diagram.

can readily be seen that $F_r/F_N = \tan \phi_x$, from which it follows that

$$F_r = F_N \tan \phi_x.$$

Force F_r imposes radial loads on the bearings of both the worm and gear, but each of these is subjected to another radial bearing load acting at right angles to F_r, and the two radial loads on the same bearing can be combined vectorially. When the worm exerts a transverse pressure F_t on the gear, it is subjected to an equal and opposite reaction which in Fig. 8 is represented by a line extending to the right from the worm axis. Both bearing loads L_a and F_r are in the same plane, and their resultant is

$$l_r = \sqrt{F_r^2 + L_a^2} \text{ lb.}$$

In the same manner we find that the radial load on the gear bearings is

$$L_r = \sqrt{F_r^2 + L_a^2} \text{ lb.}$$

To these equations may be added those for the thrust loads on the worm and wheel, respectively, already found:

$$l_a = 2T/D \text{ lb,}$$

and

$$L_a = 2T/(dr) \text{ lb.}$$

Hindley Worm Gears—All worm wheels are throated, that is, the face of the gears, instead of being a cylindrical surface, is machined to a radius slightly larger than the root radius of the worm. This "throating" operation is generally performed simultaneously with the cutting of the teeth, the throat being formed by the hob. The object, of course, is to increase

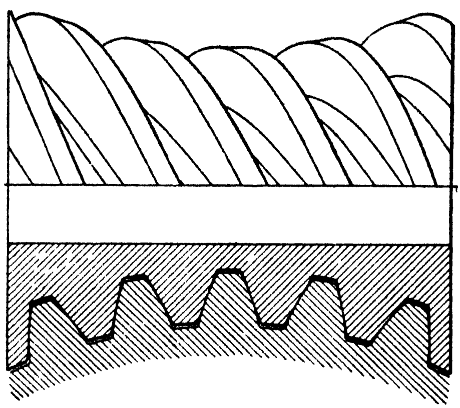


FIG 10—HINDLEY TYPE OF WORM (Shown Longer than Actually Made, to Bring Out the Hour-Glass-Effect More Clearly).

the tooth contact area over that obtainable with an unthroated face. It is also possible to "throat" the worm, and this form of worm is known as the Hindley or "hour-glass" type, Fig. 10. Such a worm ensures increased bearing surface if correctly mounted. However, its machining involves some difficulty, and it requires additional care in mounting. The ordinary straight worm must be mounted accurately in two planes; that is, the worm axis must be at a definite distance from the wheel axis, and it must also be in the median plane of the worm wheel. The Hindley or "hour-glass" worm, on the other hand, must be accurately located in three planes: The worm axis must be a definite distance from the wheel axis; it must be in the median plane of the worm wheel, and the median plane of the worm must include the wheel axis. In cutting a Hindley worm, the cutting tool must be mounted so as to turn around a center at a distance from the cutting edge equal to the radius of the worm wheel, and it must be fed in the direction perpendicular to the plane of rotation of the worm. The teeth of Hindley worms cannot be corrected by grinding after hardening, but must be cleaned or finished by hand. To make it possible to mesh a Hindley worm with the wheel, it must be made shorter than other types, and its length is usually made from one-fifth to one-fourth the wheel diameter. Lanchester, the pioneer of the passenger-car worm drive in England, used Hindley-type worms, but it is the author's impression that they are not now being used for automotive drives.

Helicoid Involute Worm Gears—If an axial section is taken through a conventional involute worm, a double rack is obtained whose straight tooth sides form an included angle of 60°. Practically all theoretical investigations of worm-

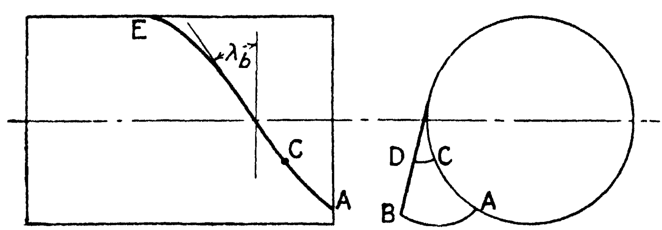


FIG. 11.—ILLUSTRATING PRINCIPLE OF HELICOID WORM.

tooth contact have been confined to the center plane, but it will be realized that, since contact extends over the whole width of the gear, other planes are of equal, or at least nearly equal, importance.

As a result of a study of tooth contact in other planes than the central one, a new form of worm tooth was developed by F. J. Bostock in England in 1920. It has since come into general use there, and is now known as the involute helicoid type of worm. Whereas the conventional worm is generated with a cutter having straight cutting edges which intersect the worm axis, the involute helicoid is generated by means of a cutter whose straight cutting edges are tangent to a cylinder coaxial with the worm and which has a diameter substantially equal to the root diameter of the worm.

The principle of the involute helicoid worm is illustrated in Fig. 11. It may be recalled that an involute is a curve generated by the end of a cord when unwound from a cylinder.

Now suppose a strip of paper or a tape to be wound around the cylinder. The end of the tape, instead of being cut off square, is cut off at an angle equal to what is known as the base lead angle, that is, the lead angle at the base circle. If now the tape is unwound from the cylinder, each point in its inclined end will generate an involute. Thus point A describes the involute AB, point C the involute CD, and so on. Each involute, of course, starts from the helix on the base cylinder. When a straight edge is applied to any point of the base circle and inclined to the transverse plane by an angle equal to the base lead angle, it will contact the worm thread over its whole depth. This makes it possible to grind the teeth of an involute helicoid worm with a straight-sided wheel, and it also affords a simple method of checking the thread profile. An axial section through an involute helicoid worm shows it to have teeth with slightly convex sides, while the

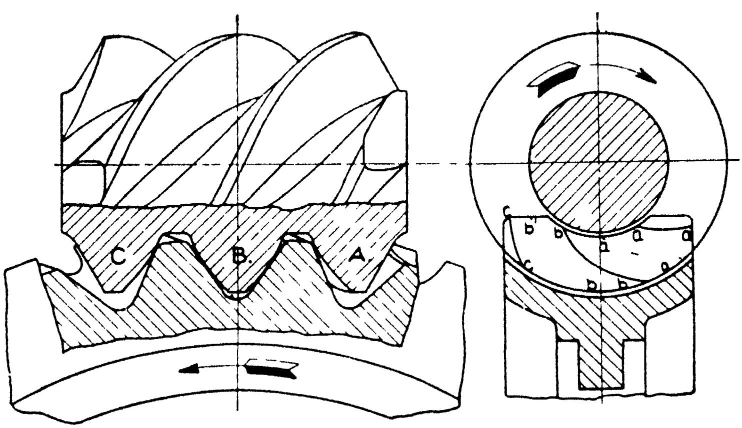


FIG 12—HELICOID WORM AND WHEEL.

sections of the gear teeth in the same plane tend to be concave. The gear is generated by means of a hob having the same form as the worm, with its teeth suitably gashed and relieved, and during the hobbing process the hob and the gear blank are rotated at the same relative speeds as the worm and gear in service.

A worm and gear with involute helicoid teeth are illustrated in Fig. 12. The view on the left shows the forms of the teeth in a plane through the worm axis, while the view on the right shows the gradual progress of the line of contact on the gear tooth from the time the worm thread first contacts it, up to the time tooth contact ceases. With the conventional worm and gear, the line of contact at all times is nearly parallel to the gear axis, and it progresses from the top toward the root of the gear tooth. Advantages claimed for the involute helicoid form of worm tooth are that the whole of the tooth surface is utilized more nearly uniformly, which equalizes tooth wear, and that the line of contact always makes a considerable angle with the direction of sliding motion, which tends to maintain the film of oil between the teeth.

Mounting of Worm Gears—Automotive worms always are made integral with their shafts, as this ensures greater rigidity

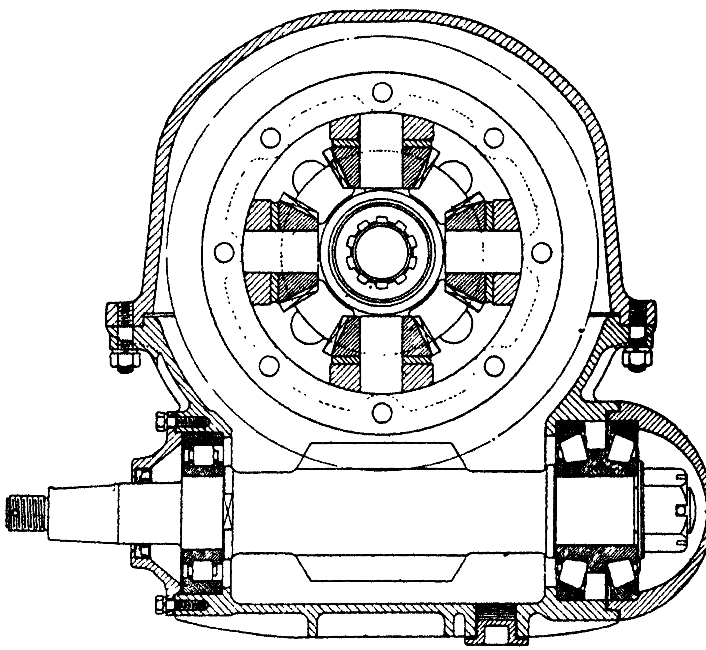


FIG. 13.—GUY BUS-DRIVE WORM GEAR, SECTION ON WORM AXIS.

for a given worm pitch diameter. In bus drives the worm is generally under-mounted, as this permits of a lower floor, which is desirable particularly in double-deck buses. Under-mounting was the usual practice also when worm drive was used for passenger cars. On trucks, on the other hand, over-mounting is more popular, because it gives increased ground clearance. Tooth-contact conditions are better with under-mounted worms, as the contacting surfaces then are always submerged in oil, and there is greater assurance that the oil film will be maintained. With over-mounted worms it is practically necessary to furnish a constant supply of oil to the contacting surfaces by means of an oil pump.

Worm-driven axles are always of the differential-carrier type. Figs. 13 and 14 show the worm-gear drive of the British Guy bus, this being an under-mounted drive. An important feature in any worm-gear mounting is the arrangement of bearings on the worm shaft. Owing to the heavy end thrust on the worm, it was formerly common practice abroad to use a thrust-type ball bearing. It was found, however, that the centrifugal force on the balls at high speeds creates a con-

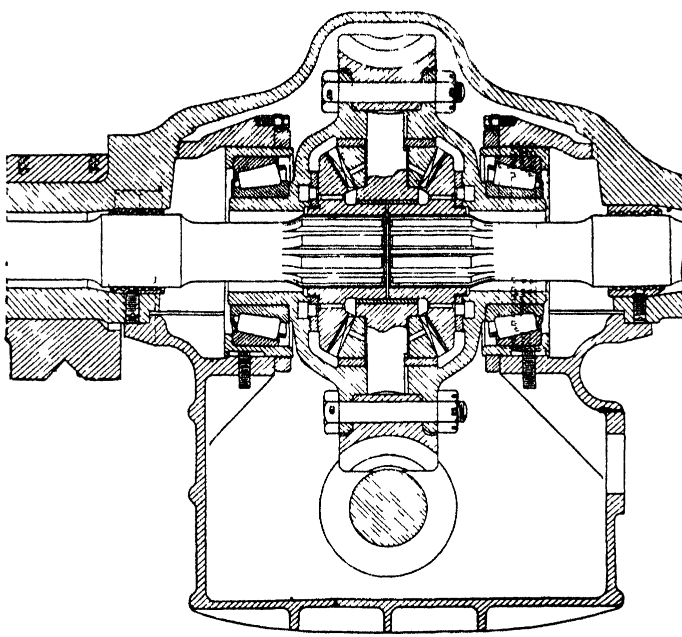


FIG 14—GUY BUS-DRIVE WORM GEAR, SECTION ON GEAR AXIS

siderable radial load, for which these bearings are not suited. Two bearing arrangements are in common use at present. One consists of a pair of conical roller bearings at the rear and a cylindrical (or barrel-shaped) roller bearing at the front end, as shown in Fig. 13, while the other comprises two bearings of a type designed for radial loads only, one a either end, and in addition a ball thrust bearing with a single outer race and two inner races, at the rear end of the shaft. The shaft is always held in position endwise by the thrust bearing, and is not restrained by the bearing at the forward end, so that it will not buckle when frictional heat raises its temperature above that of the housing, in the course of operation.

A feature of the worm drive shown in Figs. 13 and 14 is the large oil reservoir with a flat, ribbed bottom, which tends to keep the oil within the permissible temperature limit. In this design there are both longitudinal and transverse ribs on the outside. It seems to be a better plan, however, to put any transverse ribs on the inside, so that the slip stream can sweep over the longitudinal ribs and make them more effective as heat dispersers. In designing housings for worm drives, care must be exercised to ensure a mounting of adequate rigidity. The bearings at opposite ends of the worm must be brought as close together as practicable, to reduce the span, while the differential bearings must be well spread, so that the worm gear will not deflect appreciably under load. One plan to ensure adequate rigidity is to make the differential carrier in two parts, split through the axis of the differential bearings. A flat banjo-type axle housing is used, and one-half of the carrier is placed over each of the openings in the banjo, the halves being held together by "through" bolts.

Efficiency Tests of Worm Gears—When worm gears were first introduced for passenger-car drives in England, a good deal of prejudice had to be overcome, because they were popularly considered a very inefficient transmission mechanism. The Daimler Motor Company, which had adopted the Lanchester type of hour-glass worm, then had some efficiency tests made at the National Physical Laboratory. The machine used in these tests, of which a sectioned elevation is shown in Fig. 15, is based on the same principle as the electric cradle dynamometer used in making engine horse-power tests.

It has been repeatedly pointed out that a rear-axle housing under load tends to turn in a direction opposite to that of the axle shafts. Therefore, by mounting the axle housing in ball-bearing supports and holding it from rotation by means of an arm secured to it, we can measure the axle torque. If the axle is worm-driven, the housing also has a tendency to turn in a plane normal to the worm axis, in the direction opposite to that of the worm, with a torque equal to that on the worm. The worm-gear testing machine designed by Lanchester therefore supports the worm-gear housing in such a way that it can rock in two vertical planes at right angles to each other. The torques on the propeller shaft and on the rear-axle shafts then are measured by balancing the housing in both planes. This is done by means of a single weight suspended from a knife edge parallel with and at a given distance from the axle-shaft axis.

If we denote the torque on the worm shaft by t, that on the axle shafts by T, and the worm-gear reduction ratio by r, then if there were no loss in the gear we would have

$$tr = T.$$

As a matter of fact, T is always less than tr, and the efficiency is measured by the ratio T/tr.

The testing machine comprises a cradle consisting of two wheels coupled by bridges. The cradle is supported by four ball-bearing rollers, and power is transmitted to the worm

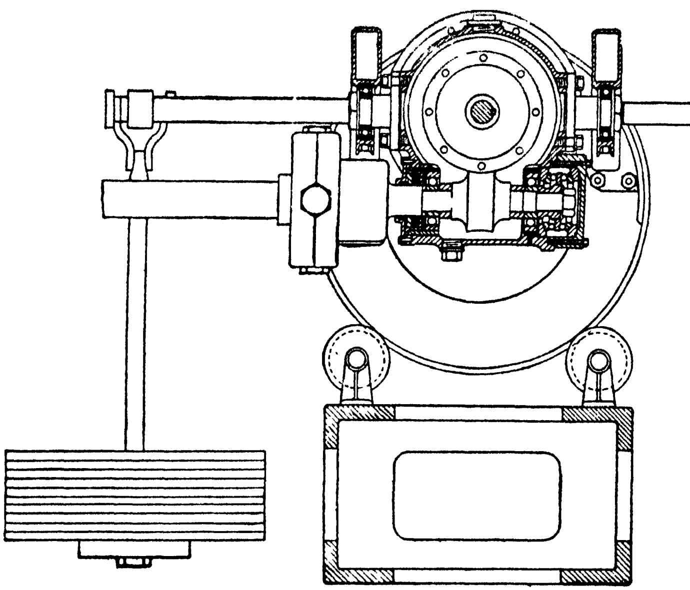


FIG. 15.—LANCHESTER TESTING MACHINE FOR WORM GEARS.

and from the hub by universal-jointed shafts, the joints being of the ball-bearing type. A balance arm is fixed to one side of the case parallel to the worm shaft. At the end of this arm there is a transverse knife-edge arm on which a weight is suspended by means of a rod. The weight can be slid along the transverse arm by means of a finger wheel, and its distance from the axis of suspension can be read off on a dial.

The gear box is supported from the cradle on ball bearings in such a way that the axis of the worm intersects the axis of the cradle wheels. When the worm-gear housing is in equilibrium, the contact point of the knife edge is located in the plane of the two axes of rotation. In operation, the finger wheel is adjusted until the gearbox is in equilibrium. Then, as the same weight is used to measure the torque around each axis of support, the torques are proportional to the distances of the point of knife-edge contact from the two axes of support, respectively. We found that

$$e = \frac{T}{tr} = \frac{T}{t} \times \frac{1}{r},$$

and since

$$\frac{T}{t} = \frac{OA}{AB} \text{ (see Fig. 16)}$$

$$e = \frac{OA}{AB} \times \frac{1}{r}.$$

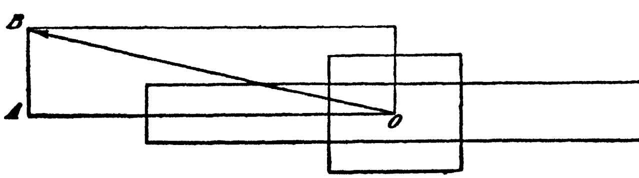


FIG. 16.—DIAGRAM OF TORQUE BALANCE.

To be able to make efficiency tests of large worm gears with a small expenditure of energy, the driven shaft (or axle shaft) is connected through a step-up bevel-gear set and a belt to the worm shaft, the step-up ratio being slightly greater than the reduction of the worm and worm wheel, so that the belt always slips slightly. As a result, only the power lost in the worm gear, bevel gear and in belt slip needs to be supplied from an outside source. The belt tension is adjusted until the weight hung from the knife edge is lifted, and when midway between stops the arm is locked in position. Readings are then taken, and afterwards the arm is released to see whether the torque has changed. Slight changes in torque do not affect the efficiency, consequently it is not necessary to constantly adjust the weight.

CHAPTER XIII

Chain Drive

Transmission by means of chains and sprockets is still used to a certain extent in motor trucks and tractors, although it no longer enjoys the popularity it did in the early years of the industry. The chain possesses the advantage of a slightly greater flexibility than the shaft drive, hence it tends to protect the engine and other parts from shock due to too rapid engagement of the clutch, road shocks, etc. When kept clean, oiled and properly adjusted, the chain is a very efficient means of power transmission, but in the final drive of sprung vehicles the chains must be exposed. They then tend to accumulate dust and grit that get into the joints or bearings, causing wear that results in noisy operation and eventually in the destruction of the chain. Gear drives are generally preferred because the gears can be fully enclosed. When chain drive is used at present, it is because one or another of its various advantages is of special importance under the conditions under which the vehicle is intended to be operated.

The greater flexibility of the chain as compared with gear drive is due to the cushioning effect of the oil films between the pins and bushings. When there is a sudden application of load, or increase in load, the film thickness at each pin is slightly reduced, and as there are quite a number of links in the tension strand of the drive, the cushioning effect is cumulative. Another advantage of chain drive is that it reduces the rear-axle unsprung weight, which results in reduced wear and tear on wheels and tires, and minimizes the effects of road shock on the sprung mass. Furthermore, with chain drive there is more road clearance at the middle of the track, which is of importance where the vehicle is likely to be used on deeply rutted roads. A minor advantage from the vehicle-manufacturer's standpoint is that the reduction ratio can be changed with much less trouble than with gear drive. At the beginning of World War II three American manufacturers listed chain-driven truck models, but by 1950 the number had been reduced to two.

Chains lend themselves particularly to the drive of six-wheeled trucks with drive on the four rear wheels. The two wheels on each side of the bogie then are driven by chains from driving sprockets a short distance apart, on the ends of concentric differential shafts, which usually are located midway between the driving wheels. Chain drives are used also on certain special vehicles, such as a lumber truck in which the powerplant is carried in a high position above the loading space, and the driving sprockets are directly above the driven wheels; and a truck with an extra, chain-driven third axle. One manufacturer of farm tractors drives the axle shafts by two chains enclosed in the gear-and-axle housing.

Construction of Chains—The only type of chain used in automotive final drives is the roller chain, which is one form

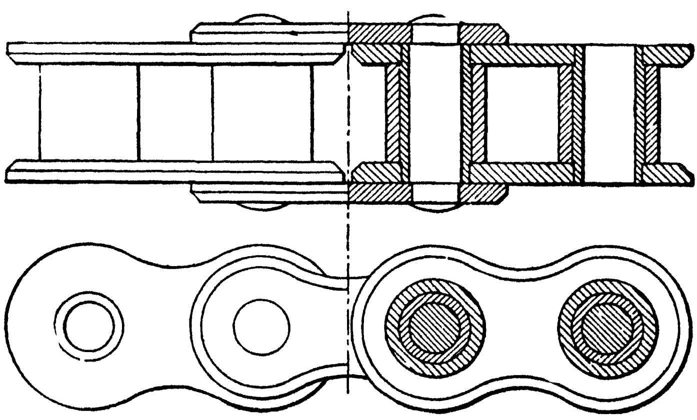


Fig. 1.—Roller Chain.

of the general class of machine-made chains. The chain (Fig. 1) consists of two sets of links, inner and outer, respectively, each set of one kind being joined to two sets of the other kind of means of a bushing and a rivet for each joint. The bushing serves to hold the pair of inner links the proper distance apart, and the rivet has both of the outer links riveted to it. In passing over a sprocket, the rivet turns inside the bushing through an angle which is equal to 360° divided by the number of teeth in the sprocket, first in one direction and then, as it leaves the sprocket, in the other. It is this motion of the joints which is responsible for the wear of chains, and as the motion is less the greater the number of teeth in the sprocket, the advantage of using large sprockets is obvious.

The bushing is surrounded by a roller which contacts the sprocket teeth. Hence the contact between the chain and the sprocket is a rolling contact, and the sliding takes place between the bushing and roller and between the bushing and pin. The rivets are generally made of nickel steel, and the bushings and rollers are hardened.

Offset Link—Whenever possible, the center distance between sprockets should be made such that the chain will have an even number of pitches. An uneven number calls for the use of an offset link (Fig. 2), and this practice is discouraged by chain manufacturers. Offset links are made in two types, plain and combination, and chain manufacturers recommend that the combination type be chosen if the use of an offset link cannot well be avoided. This latter consists of one offset and one roller link, assembled and riveted together, the pin being a press fit in the offset link plates.

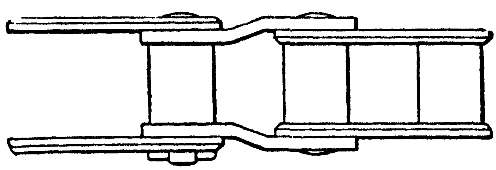


FIG. 2.—OFFSET LINK.

Capacity of Roller Chains—The permissible working load of a roller chain increases substantially as the square of the pitch, because both the pin diameter and the bushing width increase with the pitch, and the product of these two factors is the joint area, to which the working load must be proportional. In addition to the area of the pin-bushing joint there are a number of other factors that affect the maximum safe load, chief among which are the chain velocity and the weight per ft of length. According to a standard formula, the maximum safe load of a roller chain is

$$T = \frac{2{,}600{,}000A}{V + 600} - \frac{WV^2}{115{,}900}\text{ lb},$$

where A is the projected bearing area of the pin-bushing joint, in sq in.; V, the velocity of the chain in fpm; and W, the weight of the chain in lb per ft. Chain manufacturers, however, feel that two other factors should be considered in determining the load capacity, namely, the number of teeth and the rpm of the smaller sprocket, which in automotive drives always would be the driving sprocket. They publish ratings of their chains in their catalogues, based on formulae including all of the above factors, and it is advisable to be guided by these ratings.

Chain Speeds and Sprocket Diameters—In laying out a chain drive for a commercial vehicle it is advisable to make the chain speed comparatively high, provided it does not exceed the maximum safe speed, because the higher the speed the lower will be the load on the chain, for a given horse power transmitted. The maximum safe speed is given by the equation

$$V = \frac{1920}{P\sqrt{A/WP}} \text{ fpm},$$

where P is the pitch in in.; A, the projected area of the roller, and W, the weight of the chain in lb per ft.

Driving sprockets never should have fewer than 13 teeth, and a larger number is preferable, because it reduces the angular motion at the pin-bushing joints on entering and leaving the sprocket, as well as the roller impact and the load per sprocket tooth. The maximum diameter of the driven sprocket usually is limited by considerations of ground clearance required.

The pitch diameter of a sprocket for roller chains may be found by means of the following equation:

$$D_p = \frac{P}{\sin \dfrac{180°}{N}}$$

where D_p is the pitch diameter; P, the pitch of the chain, and N, the number of teeth in the sprocket.

The distance between centers of sprockets for a certain number of links in the chain may be found by means of the equation

$$L = P\,\frac{Z - \dfrac{N}{2} - \dfrac{n}{2} - (N - n)\dfrac{\beta}{180}}{2 \cos \beta}$$

where Z is the number of links in the chain; N, the number of teeth in the large sprocket; n, the number of teeth in the small sprocket, and β, the angle made by the chain with the line of centers. This equation gives the distance required for the chain to run tight on the sprockets. Some slack is necessary, and means for adjusting the center distance are always provided. The center distance affects the angle of wrap on the small sprocket, which should be at least 120°. With reduction ratios of more than 3.5 the center distance should not be less than the difference between the outside diameters of the two sprockets, while with smaller reductions it should not be less than the diameter of the large sprocket. In service, center distances should be adjusted so that with one span tight, the total in-and-out movement of the other span is about 0.5 in. per ft of span.

Sprocket Design—Driving sprockets are drop forged and after having the teeth cut are hardened to show a Rockwell C hardness of 35 to 45. During the early years of the industry, when unhardened sprockets were used, the teeth sometimes wore hook-shaped. This difficulty is now overcome by the use of better material, heat treatment, and better tooth forms. The present standard tooth form, described further on, uses a large pressure angle, which allows the chain, as its pitch increases as a result of wear, to ride farther out on the tooth face. In this way the chain adapts itself to a larger pitch circle, and the load is distributed over a larger number of teeth. It was chiefly concentration of load on the first tooth that caused teeth to wear hook-shaped.

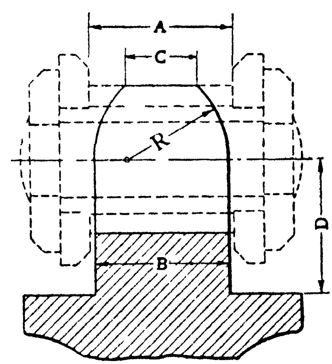


FIG 3—SPROCKET DIMENSIONS.

Driven sprockets usually are of the ring-plate type and are made of a wear-resistant alloy steel which is heat-treated before being machined. They can be bolted or riveted to a flange cast on the brake drum. Referring to Fig. 3, the width B of the sprocket is made equal to $0.93A - 0.006$ in., the tolerance on this dimension being $\pm$ ($0.01A + 0.002$ in.), and the sprocket teeth are rounded off on both sides to a radius $R = 0.43P$, P being the pitch of the chain. The center of the arc R lies on the pitch circle (normally $0.3P$ below the top of the tooth). The clearance between the side links and the full-width portion of the sprocket ($A - B$) is made $0.07A + 0.006$ in., and the width C at the top of the teeth is ($0.93A - 0.25P - 0.006$ in.).

Sprocket wheels are cut by means of formed cutters which have been standardized by the Sectional Committee for the Standardization of Gears of the American Standards Associa-

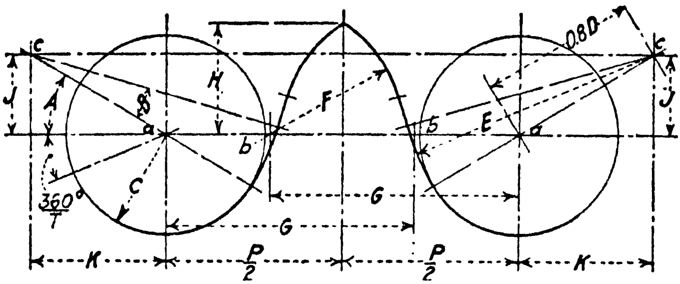


Fig. 4.—Sprocket Tooth Form.

tion, which also standardized chains and sprocket forms. Fig. 4 shows one of two standardized roller-chain-sprocket tooth forms, the dimensions for different pitches P, roller diameters D and numbers of teeth T being calculated as follows:

$$C = \frac{1.005D + 0.003 \text{ in.}}{2}$$

$$A = 35° - \frac{120°}{T}$$

$$J = 0.8D \sin\left(35° - \frac{120°}{T}\right)$$

$$K = 0.8D \cos\left(35° - \frac{120°}{T}\right)$$

$$E = 0.8D + C$$

$$G = 1.24D$$

$$F = 0.8D \cos\left(18° - \frac{56°}{T}\right) + 1.24D \cos\left(17° - \frac{64°}{T}\right) - E$$

$$H = \sqrt{\left[F^2 - \left(G - \frac{P}{2}\right)^2\right]} \quad \text{When } \frac{P}{2} > G, \text{ then } H = F$$

$$O.D. = \text{Outside Diameter} = P \cot \frac{180°}{T} + 2H$$

$$B = 18° - \frac{56°}{T} = \text{Working Arc.}$$

Arcs F and E are connected by a common tangent to complete the tooth form.

The above equation for the outside diameter of the sprocket blank will cause the teeth to come to a sharp edge. A slightly smaller diameter of blank is generally preferred, as given by the equation

$$O.D. = P\left(0.6 + \cot\frac{180°}{T}\right).$$

In order to ensure concentricity of the sprocket and its hub or center when the two parts are made separate, the sprocket blank should be made an accurate fit over a turned portion of the hub or center, against the flange to which it is bolted.

Chain Load—In the design of chain drives the average and maximum loads on the chain are significant figures. The maximum chain pull or chain tension is equal to

$$t = \frac{12Tre}{100d}\ \text{lb},$$

where T is the maximum engine torque, in lb-ft; r, the overall reduction ratio between the crankshaft and the driving sprocket; e, the efficiency of the gearing between engine and the driving sprocket, in per cent, and d, the pitch diameter of the driving sprocket, in in. The thrust load on the radius rod is equal to the chain load. The bearing load on the bearing adjacent to the sprocket, on the other hand, is equal to $t/\cos\phi$, where ϕ is the pressure angle. With standard sprocket teeth the average pressure angle is given by the equation

$$\phi_a = 26° - (92°/T).$$

Chain-Adjusting Rods—Chain-adjusting rods, also known as radius rods, serve a triple purpose. They take up the reaction due to the chain pull, allow of adjusting the slack in the chain, and transmit the driving thrust or braking pull from the rear axle to the frame. These rods must be jointed at both ends so as to permit of free play of the springs, and the joint centers preferably should lie in the axes of the sprockets, so that any play of the springs will not affect the sprocket center distances.

Fig. 5 shows a radius rod which has a spherical joint at its forward and a double pivot joint at its rear end. The turnbuckle is provided with a head whose upper and under faces are turned spherically. The upper face of the head bears against the spherical head of a steel button inserted into a drill hole in the wall of the fitting on the jackshaft bearing bracket, and against the under face of the head

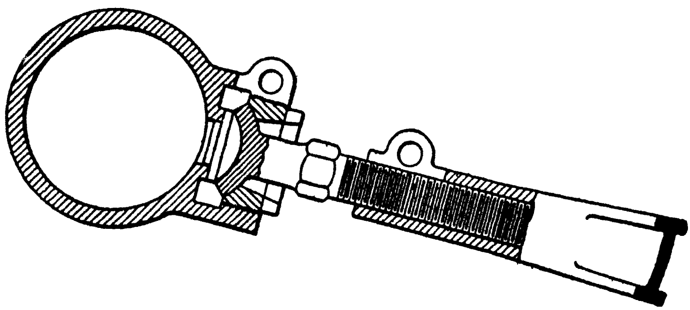


FIG. 5.—RADIUS ROD, FRONT END.

presses an externally threaded ring screwed into a threaded recess in a boss formed on the fitting, which ring is also provided with a spherical surface. After adjustment has been made, the nut can be locked in position by means of a clamp screw, and the same locking means is employed for the threaded shank of the turnbuckle.

The joint of the radius rod to the rear axle is of the double pivotal type. A lug is formed on the hub of the brake support which is swiveled on the rear axle, and the rear end of the radius rod is connected to this lug by means of a pin which is held in position either by means of a bolt head and nut or a locking pin, as shown in Fig. 6.

Effect of Spring Play on Chain Drive—Compression and rebound of the chassis springs cause acceleration and deceleration of the vehicle with practically all types of drive,

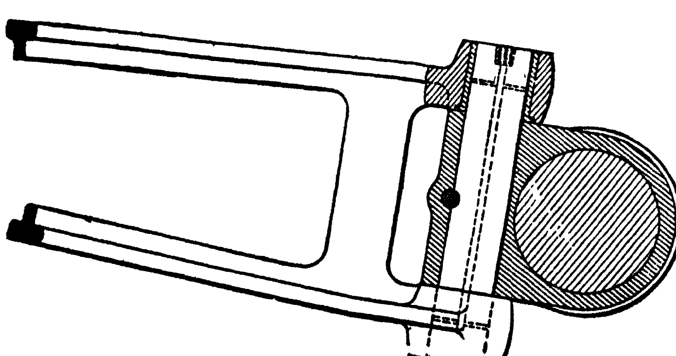


FIG. 6.—RADIUS ROD, REAR END.

but with gear drive, because of the long propeller shafts now commonly used, the effect is very small and is generally neglected. The effect of spring play with chain drive may be investigated with the aid of the diagram shown in Fig. 7, in which the driving sprocket is shown in the positions of maximum spring compression (full lines) and maximum rebound (dotted lines) respectively. The sprocket pinion is supposed to have fifteen teeth, and the sprocket wheel forty-five. The distance between centers is assumed to be 28 in., and the total vertical motion of the springs 6 in. We will assume that when the springs are compressed the line of centers is horizontal. Then when the springs rebound, the line of centers moves through an angle α determined by the relation

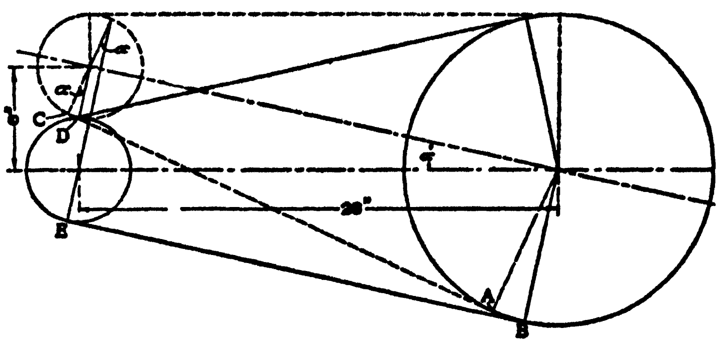


FIG. 7.—DIAGRAM ILLUSTRATING EFFECT OF SPRING ACTION ON CHAIN DRIVE.

$$\sin \alpha = \frac{6}{28}.$$

Referring to the diagram, a portion of the chain whose length is

$$AB = \frac{\pi D\alpha}{360} \text{ in.}$$

winds up on the sprocket wheel, and a portion whose length is

$$CD = \frac{\pi d\alpha}{360} \text{ in.}$$

unwinds from the sprocket pinion. The length of chain between the extreme points of contact on the two sprockets (EB, CA) remains the same. Since the length of chain which unwinds from the pinion is not equal to that which winds up on the wheel, if we assume that the wheel remains stationary the pinion must turn to unwind a length of chain

$$AB - CD = \frac{\pi(D - d)\alpha}{360} \text{ in.}$$

It must turn in the forward direction when the springs rebound and in the backward direction when they compress.

A motion of

$$\frac{\pi(D - d)\alpha}{360} \text{ in.}$$

on the circumference of the sprocket pinion corresponds to an angular motion

$$\Theta = \frac{D - d}{d} \alpha \text{ degrees.}$$

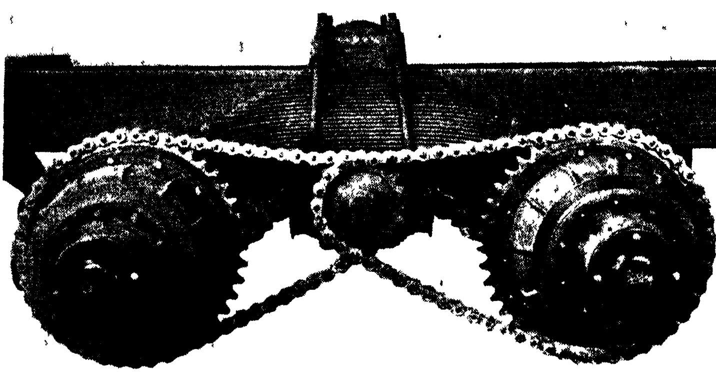


FIG. 8.—DUAL CHAIN DRIVE OF STERLING SIX-WHEEL TRUCK.

It will be seen from this that when $D = d$—that is, when the two sprockets are of the same size—the spring action has absolutely no effect on the drive, and the effect is the less the smaller the difference in the sizes of the two sprockets.

Dual-Chain Drive—An application of chain drive in modern automotive practice is illustrated in Fig. 8, which shows the drive of the Sterling six-wheel truck, with a gvw rating of 60,000 to 80,000 lb. This truck, which is equipped with a 150-hp engine, has a jackshaft located midway between the two rear axles and driven by double-reduction gearing. In the same housing with the driving gears there are three differentials of the non-stalling type. The first of these, which is located inside the spur gear of the double-reduction set, divides the torque between the other two, and each of the latter divides the torque impressed on it between two concentric shafts, one inside the other, which at their outer ends carry the sprocket pinions for the chain drives to the two wheels on one side. The inverted semi-elliptic spring is clipped to a bolster or spring seat on the jackshaft housing, and its ends bear on pads on the axles, which enables the axles to accommodate themselves to rough terrain without unduly stressing the frame. There are radius rods with means for chain adjustment between the two axles and the jackshaft housing, to which latter they connect with ball-and-socket joints.

Chain Lubrication—The following recommendations regarding the lubrication of drive chains are made by Diamond Chain & Mfg. Co., a leading manufacturer of roller chains: "The only efficient method of lubricating chains exposed to grit and dirt is to remove them periodically, or when needed, for a thorough cleaning in gasoline, kerosene, or some other distillate. After having become dry, the chains should be immersed in an oil bath at from 160 F to 200 F, long enough to allow thorough penetration. The lubricant preferably should be a mixture which is very fluid when heated, but rather heavy when cooled to ordinary temperatures. After being removed from the bath, the chain should be laid out horizontally, with the pins in a horizontal plane and with the links pushed together. Although the congealed mixture may collect a lot of dirt, it serves as a seal between adjacent link plates to keep the grit out of the bearing joint and the lubricant in."

CHAPTER XIV

Brakes

The brakes of a motor vehicle serve three distinct purposes. They are used to bring the car to a (relatively quick) stop, to prevent it from attaining too high a speed when coasting down hill, and to hold it in position on grades. Every brake consists of two members or sets of members, one of which is connected either directly to a road wheel or to a shaft in permanent driving relation therewith, while the other is anchored to the car frame or axle; that is, mounted thereon in such a way that it cannot rotate. Means are provided by which the members can be brought into frictional contact (the brake applied). If the brake is applied while the car is in motion, the friction between its members must be overcome; this results in the absorption of energy from the store of kinetic energy of the moving vehicle, the speed of which is thereby reduced.

Legal Requirements—Early motor-vehicle laws usually called for two independent brakes, and sometimes it was stipulated that at least one of these must act directly on the road wheels. The present (1951) New York State law, which is typical, merely requires that "every motor vehicle driven or operated on the public highways of the state shall be provided with adequate brakes and steering mechanism, in good working order and sufficient to control such vehicle at all times." However, the law stipulates that the Commissioner for Motor Vehicles shall make rules providing standards of brake efficiency, and the Commissioner has set minimum stopping distances for both service brakes and hand brakes. The law also provides that all trailers and semi-trailers of more than 3000 lb unladen weight must be provided with brakes. According to the Commissioner's rules, the service brakes on every motor vehicle and every combination of motor vehicle and trailer must be capable of stopping the motor vehicle or combination from 20 mph on hard, dry, approximately level pavement, substantially free from loose material, in 30 ft in the case of four-wheel brakes, and in 45 ft in the case of others. Hand brakes must be capable of stopping the vehicle or combination from 20 mph in 75 ft under the same conditions, and on other than passenger cars with a capacity of less than 8 passengers, must be capable of holding the vehicle on a grade of 25 per cent. All tank vehicles must be provided with four-wheel brakes, and tank trailers or semitrailers must have brakes that go on automatically in the event the trailer breaks away from the towing vehicle. The brakes on the trailer, moreover, must be of such design that they are applied simultaneously with, or ahead of, the brakes of the towing vehicle.

While the early motor-vehicle laws laid stress on independence of the two braking systems, at present the same brake shoes which are actuated by the pedal for service braking often are applied also by the brake lever when the car is to be parked, and the two systems therefore are not entirely independent. Motor-vehicle administrators have held repeatedly that all legal requirements are satisfied if the layout is such that any single failure which might occur anywhere in the system would leave brakes on at least two wheels operative.

The original intention, when two brakes were called for, was to have something to fall back upon if the regular or service brake should fail, and the second brake therefore usually was referred to as the emergency brake. Later service brakes were so improved that their complete failure was a very remote possibility, and at present the term "parking brake," is usually applied to the second brake, to indicate that it is intended chiefly to hold the car in position. The service brake is always applied by a pedal, while the parking brake is applied by a hand lever, as a rule. The parking brake can be set in the "on" position by means of a latch, while the service brake remains on only as long as the driver presses down on the pedal.

Historical Development—All of the earlier motor vehicles were braked only on their rear wheels. With the moderate speeds and the low density of traffic of that period, braking on two wheels was quite sufficient from the safety standpoint. Rear-wheel—rather than front-wheel—braking was adopted because it is much easier to fit brakes to the rear (non-steering) than to the front (steering) wheels.

Within certain limits the retarding force due to brake application is proportional to the pressure with which the bands or shoes are applied to the drums, and to the coefficient of friction between lining and drum. However, when the force of application reaches a certain value, the brakes lock, and the road wheels then slide over the pavement, hence any further increase in the force of application has no effect. Thus the maximum braking effect which can be produced depends on the adhesion between tire and road, which in turn depends on the load carried by the wheels on which the brakes act and on the coefficient of friction between tire and road. When the brakes act on the rear wheels alone, only part of the gross vehicle weight is utilized for retarding purposes.

As the speeds of motor vehicles increased, braking on the rear wheels alone soon proved insufficient. Four-wheel brakes came into use first on racing cars. Considerable interest in such brakes for private cars developed in England about 1908, but some of the systems developed there during the next few years proved faulty in operation, with the result that the movement suffered a temporary setback. It was taken up again at the conclusion of World War 1. Several years later American car manufacturers, led by Buick among large-scale producers, installed four-wheel brakes on their production models, and such brakes soon became very popular. Today a motor-propelled road vehicle without brakes acting on all of its road wheels may be considered impractical.

One of the pioneers in four-wheel brake development was Henri Perrot, a Frenchman, who, as engineer of the Argyll Company of Glasgow, Scotland, designed a four-wheel braking system that later came into wide use. Perrot's American patents were acquired by Vincent Bendix, and they formed the nucleus around which the present very extensive Bendix brake business was built up. However, the Perrot system is no longer being used.

As long as vehicles were braked through their rear wheels only, every vehicle manufacturer produced his own brakes, unless he obtained his rear axles from a parts maker; but with the application of brakes to front wheels a number of rather difficult engineering problems had to be solved, and since that time brake manufacture has become specialized. One of the problems was that of transmitting the actuating motion to the front brakes in such a way that brake application will not be affected by steering, and will not interfere with steering. With brakes on the rear wheels, on the transmission, or on the propeller shaft, the actuating motion can be transmitted by a comparatively simple mechanical linkage. But with brakes mounted on swiveling steering knuckles the problem is not so simple. It could be solved either mechanically or hydraulically, and although most of the earlier four-wheel braking systems were of the mechanical type, a hydraulically operated, or hydraulic brake, the invention of Malcolm

Lockheed of California, appeared soon after the advent of four-wheel brakes in the American passenger-car field. The original Lockheed hydraulic brake was of the (external) band type.

Types of Brake—Motor-vehicle brakes may be divided into two types—those applied directly by pedal pressure, and those applied by some other source of power, such as vacuum, compressed-air, or electric. Power-actuated brakes will be dealt with in the next chapter, and we will here confine ourselves to brakes applied directly by muscular power. These again may be divided into two types, viz., those in which the applying power is transmitted from the brake pedal or lever to the brake proper by a mechanical linkage, and those in which it is transmitted hydraulically. Probably more than 95 per cent of all automotive vehicles now being turned out by the factories have hydraulic service brakes, and as far as the scale of use is concerned, the hydraulic type easily ranks first. But during the first quarter century of automobile history all road vehicles carried only mechanical brakes, and it may serve to give the reader a better perspective of brake development if the mechanical type is discussed first. For, as already mentioned, it was an increase in road speeds and in the density of vehicular traffic that led to the adoption of four-wheel brakes, and certain exigencies of four-wheel-brake design resulted in a preference for the hydraulic type.

Brakes also may be divided into two classes according to the form of the friction surfaces at which the kinetic energy of the vehicle is converted into heat energy, into drum and disc brakes. The great majority of vehicle brakes are of the drum type, but disc brakes have long been used as parking brakes on trucks, and have come into limited use as service brakes in recent years.

The drum type of brake may be either a band brake or a shoe brake. Both band brakes and shoe brakes may be either external or internal, though band brakes generally are external and shoe brakes internal. The non-rotating part of a band brake consists of a flexible steel band, lined with friction material, which can be brought into contact with the rotating part, the drum, by exerting a pull on its free ends when the brake is external, and pushing against its ends when the brake is internal. Internal band brakes, however, have never been used to any extent, largely because they tend to be self-locking. The band is supported at or near its middle by a bracket or plate and an anchor pin. This structure takes up the turning moment due to the friction when the brake is applied.

The chief difference between a shoe brake and a band brake is that in the former the non-rotatable members are comparatively rigid, while in the latter they are flexible. Usually there are two shoes inside a drum, each forming an arc of about 120°. Each shoe is pivotally supported (anchored) at one end. There may be either a separate anchor pin for each shoe, or one pin may serve for both shoes. The free ends of the shoes can be forced apart by a suitable mechanism, and the shoes thus brought into contact with the interior surface of the drum.

Simple diagrams of an internal shoe and an external band brake are shown in Figs. 1 and 2, respectively. These diagrams show only the principal parts, viz., the brake drum, the shoes in the one case and the band in the other, the anchors which hold the brake from rotating with the drum, and the expanding or contracting means. In addition there are needed retracting springs which withdraw the shoes or band from the drum when the brake is released, centering means to ensure that when released the brake will clear the drum all around, adjusting means to compensate for wear and to equalize the clearance around the circumference, and brackets or a backing plate to support the brake anchor and the expanding means. The expanding means shown in Fig. 1 is a cam, the type most commonly used in mechanical brakes. In hydraulic brakes the shoes are forced into contact with the drum by either a single cylinder with two pistons, or by two cylinders with a single piston each.

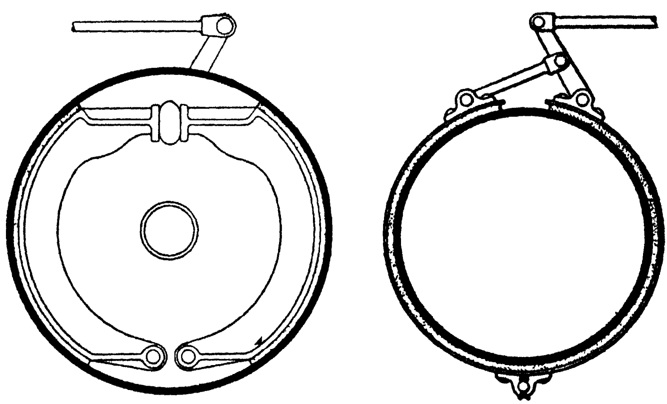


FIG. 1 (*left*).—DIAGRAM OF INTERNAL SHOE BRAKE.
FIG. 2 (*right*).—DIAGRAM OF EXTERNAL BAND BRAKE.

Enclosure of Brakes—When external brakes are fitted directly to the wheels, they are exposed to splashing mud and water, and they usually lose much of their effectiveness in rainy weather. In modern road vehicles the brake is protected by a backing plate, riveted or otherwise fastened to the axle housing or steering knuckle, which envelops the edge of the brake drum in such a way that it is almost impossible for splashing or dripping water to get into the drum. Oil seals are provided in the ends of the axle housing or in the wheel hubs, to keep oil or grease out of the brake drums. Contracting band brakes now are used only as parking brakes, and are mounted either on the transmission or on the propeller shaft, where they are less exposed to mud and water.

Current Practice—All American passenger cars in production at the half-century mark had hydraulic service brakes of the internal type. The majority used the same rear-wheel brakes also for parking, being provided with mechanical actuating means in addition to the hydraulic gear, while the remainder had parking brakes of the contracting band type on the transmission. Ninety-five per cent of all motor bus models listed carried air brakes, the remaining five per cent—most of them of small size—having ordinary hydraulic brakes. All of the buses had mechanical parking brakes on the drive shaft. The truck field was divided between brakes applied by muscular power (mostly of the hydraulic type), vacuum brakes, and air brakes. The first-mentioned type predominated in trucks of up to about 14,000 lb gvw; vacuum brakes in trucks between 14,000 and 24,000 lb gvw, and air brakes in trucks over 24,000 lb. With the exception of a few small truck models, with brakes similar to those carried on passenger cars, in which the rear-wheel brakes serve for parking as well, all of the trucks had separate parking brakes mounted either on the transmission or on the drive shaft. Most of these were external band or shoe brakes, but disc brakes also were extensively used, while a few of the largest trucks had internal parking brakes.

Stopping Time and Distance—When a mass of one pound is acted upon by a force of one pound, as in the case of a free-falling body, the mass is accelerated at the rate of 32.16 feet per second per second (fps^2). This is known as the acceleration of gravity, and is generally designated by the letter g. The acceleration is directly proportional to the force producing it, and inversely proportional to the mass accelerated, hence

$$a = \frac{Fg}{W} \text{ fps}^2.$$

Deceleration of a moving mass may be regarded as negative acceleration, and is subject to the same laws as acceleration. When a vehicle is being brought to a stop by means of its brakes, the maximum value which the decelerating force can attain is substantially

$$F = \mu W,$$

where W is the weight of the vehicle and μ the coefficient of friction between tire and pavement. This maximum decelerating force, however, is available only if brakes act on all of the road wheels and the braking force on each road wheel is sufficient to stop or almost stop rotation of the wheel. The coefficient of friction between the tread of a rubber tire and a clean, dry concrete pavement usually varies between 0.65 and 0.85. However, these ideal braking conditions rarely obtain, and it is usual to assign a value of 0.6 to the coefficient of friction under normal conditions. As the decelerating force is μW and the mass being decelerated has a weight W, the rate of deceleration according to the above formula is

$$a = \frac{\mu W g}{W} = \mu g.$$

When a vehicle travels at a speed of V miles per hour, its velocity is

$$v = \frac{5280}{3600} V = 1.466V \text{ fps}.$$

To bring the car to a stop from an initial velocity of $1.466V$ fps when the rate of deceleration is μg fps^2 evidently requires a time

$$t = \frac{1.466V}{\mu g} \text{ seconds}.$$

With constant deceleration the mean velocity of the vehicle during the stopping period is exactly one-half its initial velocity, or $0.733V$ fps, and as the distance in which the vehicle is brought to a stop is equal to the product of the mean velocity by the time, we get for the stopping distance

$$d = \frac{1.466V}{\mu g} \times 0.733V = \frac{1.075V^2}{\mu g} \text{ ft}.$$

Assuming the coefficient of friction μ to be 0.6, and inserting this and the value of g, we get for the stopping distance for any initial speed V mph,

$$d = \frac{V^2}{18} \text{ ft.}$$

Inserting the same values in the equation for the stopping time we get

$$t = \frac{V}{13.15} \text{ sec.}$$

We thus see that while the time required to bring the vehicle to a stop by means of the brakes varies directly with the initial speed, the stopping distance varies as the square of the speed. Following are the minimum stopping distances from various speeds when the friction coefficient between tire and road is 0.6:

Speed (mph)....	10	20	30	40	50	60	70	80	90
Distance (ft)...	5.6	22.2	50.0	88.8	138	200	272	356	450

Normal Brake Performance—The stopping distances given in the foregoing table correspond to what may be called emergency braking, that is, braking at the maximum rate to avoid an accident. Such severe braking at high speeds involves serious hazards, for one reason because the driver is likely to lock the wheels, thereby throwing the car into a skid, and for another because the passengers are likely to be thrown off their seats. A series of tests on the effects of various rates of car deceleration was run on General Motors Proving Grounds at Milford, Mich., and the results may be summarized as follows: On dry, level concrete pavement a car can be brought to a stop from a speed of 70 mph with an average deceleration as high as 19.5 fps^2 without leaving a 12-ft lane. This corresponds to a stopping distance of 270 ft, or substantially the same as that given for this speed in the table. Such a fast stop, however, is not only uncomfortable, but may result in personal injury. Deceleration at the rate of 13.9 fps^2 is still severe and uncomfortable, and is classed as an emergency stop by the driver. Objects lying on the seats will still slide off. Deceleration at 11 fps^2 does not alarm the passengers, but the average driver would rather not stop that fast. The maximum rate of deceleration which does not interfere with passenger comfort is about 8.5 fps^2, or somewhat less than one-half the maximum, emergency deceleration.

Natural Rates of Deceleration—The tests also covered the subject of car deceleration without brake application, with the engine "in gear" and "out of gear," respectively. The natural deceleration is not constant but varies with the speed. It was found that if the car is allowed to coast with the engine in gear and the throttle fully closed, the deceleration varies substantially linearly from about 2.8 fps^2 at 60 mph, to about 1.2 fps^2 at 20 mph, while with the engine out of gear it varies from about 1.4 fps^2 at 60 mph to about 0.45 fps^2 at 20 mph. On smooth, level concrete highway in still air it takes slightly more than a mile for a car to decelerate from 70 to 15 mph with the engine out of gear.

Conditions Ensuring the Quickest Stop—It was found through experiments made by the Westinghouse Air Brake Company many years ago that railway brakes exert the greatest retarding effect when applied with such force that the wheels do not quite lock, but continue to revolve. This un-

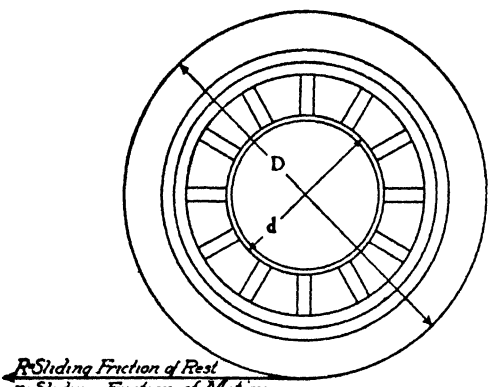


FIG. 3.—MINIMUM-STOPPING-DISTANCE DIAGRAM.

doubtedly applies also in the case of automobile brakes. It may be explained on the grounds that the friction of rest is greater than the friction of motion, and that when the wheels become locked, the rolling friction of the wheel on the ground and the bearing friction of the axle and propeller shaft cease to absorb energy.

The braking effect obviously is a maximum when the energy dissipated by friction in traveling a unit distance is a maximum. Referring to Fig. 3, let R be the starting resistance of the wheel to slippage on the road (friction of rest), and r the resistance of the wheel to slippage once it has begun (friction of motion). Also, let D be the wheel diameter and d the diameter of the brake drum on the wheel. Then the maximum frictional force which can be applied to the brake drum without causing the wheel to slip is $(D/d)\ (R - a)$, where a is a very small quantity. While the car travels a unit distance, the friction surface of the brake drum moves a distance d/D, and the energy absorbed at the brake surface is

$$\frac{d}{D} \times \frac{D}{d}(R - a) = R - a.$$

In addition to this we have the energy absorbed by the rolling friction of the wheel and the bearing friction. We will denote the sum of these two frictional forces, both referred to the wheel rim, by B, and the energy absorbed by these two resistances while the car moves a unit distance may also be represented by B. Therefore, the total energy absorbed while the car travels a unit distance when the brakes are on but the wheels are not locked is $R + B - a$.

On the other hand, after the wheels have stopped rotating and begun to slip, the only resistance encountered is the sliding friction r of the wheel on the road. The energy absorbed in unit distance as a result of this friction is also represented by r. Hence we must prove that

$$R + B - a > r.$$

It has been stated already that the friction of rest R is greater than the friction of motion r. Of the two remaining items of the expression for the energy absorbed with the wheels revolving, B has a definite value which on good roads amounts to from 4 to 5 per cent of the sliding friction. The item a, on the other hand, may be made practically nil, as it represents the margin which, if added to the brake friction referred to the wheel circumference, would cause the wheel to lock. Hence, the retarding action with wheels turning is greater than that with wheels locked by the sum of the following three items: The rolling friction of the wheels on the ground, the axle-bearing friction, and the difference between the friction of rest and the friction of motion between wheel and pavement.

Objections to Wheel Locking—Not only will the brakes stop the car more quickly when they are not locked, but wear and tear on the tires are greatly reduced. Brakes have been designed in which locking of the wheels is prevented automatically, but they have not come into practical use, probably on account of their complication and excessive weight. Locking often occurs when a wheel hits a bump and is thrown into the air, when, if the clutch is out, there is only the rotary momentum of the wheel to resist it. In this connection it is interesting to point out that the maximum braking effect obtainable is greater with low-pressure than with high-pressure tires, as the former hold the road better. The chief objection to locking of the rear wheels, however, is that it is almost certain to throw the car into a skid, and to prevent locking when descending fairly steep grades with slippery pavement, the engine should always be left in gear.

Cause of Skidding—One of the advantages of four-wheel over two-wheel brakes is that the former materially reduce the skidding tendency of a car. When there are brakes on two wheels only, it takes only about one-half the pedal pressure to lock them, and locking, therefore, is much more likely to occur. As long as the rear wheels revolve, their motion is a rolling one, and they can progress over the road surface only in the direction of their planes. The reason for this is that the resistance to rolling motion is much smaller than that to sliding motion—something like 15 lb per 1000 instead of 600 lb per 1000 on hard, dry pavement. On greasy roads even a slight application is likely to lock a pair of rear-wheel brakes, unless the engine is in gear, and once the wheels are locked they will slide as easily sideways as forward. Then all that is necessary to start a skid is that the center line of the resistances encountered by the front wheels does not pass through the center of gravity of the car; in other words, that one front wheel encounter slightly greater resistance than the other. Locking the front wheels will not produce a rear-wheel skid, and it can cause the front wheels to skid laterally only if they are out of the straight-ahead position.

Weight Transfer—In the foregoing discussion it was assumed that each pair of wheels, front and rear, carried a definite proportion of the gross vehicle weight, to which the ground adhesion is proportional. However, when the car is being decelerated by the brakes, the division of weight between front and rear wheels is not the same as with the car at rest. The force of inertia acting forwardly at the center of gravity of the vehicle, and the braking or retarding force acting toward the rear at the points of road contact of the wheels produce a couple which has the effect of pressing the forward wheels more firmly against the pavement and relieving the pressure of the rear wheels on the pavement, thus virtually transferring weight from the rear to the front wheels.

With four-wheel brakes the braking or retarding force has a maximum value μW, and if we designate the height of the center of gravity of the vehicle by h, the value of the couple causing the weight transfer is μhW. This is equal to the balancing couple due to the weight w transferred from the rear to the front wheels, which acts on an arm equal in length to the wheelbase l. We then have

$$lw = \mu hW,$$

hence

$$w = \frac{\mu hW}{l} \text{ lb.}$$

In the case of an average passenger car, h/l is equal to about $\frac{1}{5}$, and if we assume μ to be equal to 0.6 we get

$$w = 0.12W \text{ lb.}$$

That is to say, about 12 per cent of the total car weight is

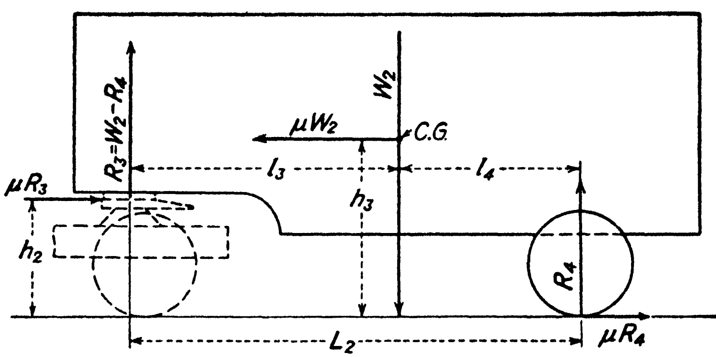


FIG. 4.—FORCES ACTING ON TRAILER WHEN BRAKES ARE APPLIED.

transferred from the rear to the front wheels under extreme brake application, so that if the weight distribution with the car at rest is, say, 45 per cent on the front and 55 per cent on the rear wheels, when the car is being severely braked it becomes 57 per cent on the front and 43 per cent on the rear wheels.

Weight Transfer in Tractor-Trailer Combinations—An interesting problem is that of the transfer of weight under brake action in a combination comprising a tractor and a semi-trailer. Referring to Fig. 4, when the brakes are applied, there are five forces acting on the trailer, namely, an upward reaction R_3 at the coupling, a rearward reaction $R_3\mu$ at the coupling, the force of gravity W_2 at the center of gravity of the trailer, the inertia force $W_2\mu$ at the center of gravity, and the retarding force $R_4\mu$ at the points of road contact of the trailer wheels. Under steady brake action the trailer is in equilibrium, and the moments around any axis then vanish. This gives

$$R_3L_2 + R_3\mu h_2 - W_2l_4 - W_2\mu h_3 = 0.$$

As

$$R_3 + R_4 = W_2, \quad R_3 = W_2 - R_4,$$

and making this substitution and transposing, we get

$$W_2(L_2 + \mu h_2 - l_4 - \mu h_3) = R_4(L_2 + \mu h_2),$$

from which it follows that

$$R_4 = W_2 \frac{L_2 - l_4 + \mu h_2 - \mu h_3}{L_2 + \mu h_2} \text{ lb.}$$

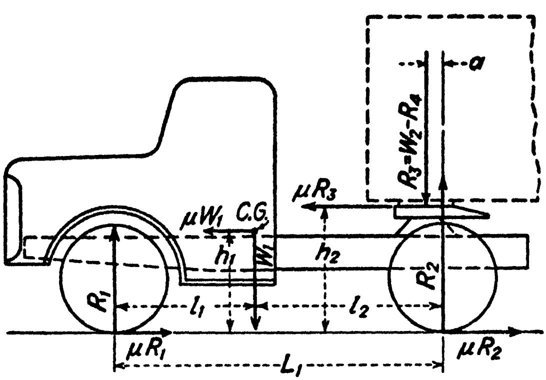


FIG. 5.—FORCES ACTING ON TRACTOR WHEN BRAKES ARE APPLIED.

The tractor is subjected to the following forces (Fig. 5): An upward reaction R_1 against its front wheels, an upward reaction R_2 against its rear wheels, an inertia force $W_1\mu$ acting forward at its center of gravity, the force of gravity W_1 acting downward at its center of gravity, the forward push $R_3\mu$ of the trailer on the coupling pin, a gravitational force equal to the reaction R_3 acting downward on the bolster, and the retarding or braking forces $R_1\mu$ and $R_2\mu$ acting toward the rear at the points of road contact of the front and rear wheels respectively. All except the second and the last two forces create moments around a transverse axis through the points of road contact of the rear wheels. For the condition of equilibrium these moments vanish, and we then have

$$R_1L_1 - W_1\mu h_1 - W_1l_2 - R_3\mu h_2 - R_3a = 0,$$

from which it follows that

$$R_1 = \frac{W_1(\mu h_1 + l_2) + R_3(\mu h_2 + a)}{L_1} \text{ lb.}$$

The reaction on the tractor rear wheels can be readily found after the reactions on the tractor front wheels and the trailer wheels have been obtained by means of the foregoing equations, because the sum of the reactions on all three pairs of wheels must be equal to the combined weight of tractor and trailer:

$$R_2 = W_1 + W_2 - R_1 - R_4 \text{ lb.}$$

We will now apply the foregoing equations to the following practical example: Let W_1 be 8000 lb; W_2, 24,000 lb; L_1, 128 in.; L_2, 180 in.; l_1, 54 in.; l_2, 74 in.; l_3, 108 in.; l_4, 72 in.; h_1, 40 in.; h_2, 46 in.; h_3, 72 in.; a, 6 in. In tractor-trailer combinations of this type, equipped with air brakes, the brakes usually are designed to give a maximum deceleration of somewhat less than $0.5g$, hence we can make μ equal to 0.5. Inserting values in the equation for the reaction (or load) on the trailer wheels under maximum brake application we get,

$$R_4 = 24{,}000 \frac{180 - 74 + (0.5 \times 46) - (0.5 \times 72)}{180 + (0.5 \times 46)} = 11{,}230 \text{ lb.}$$

The load on the fifth wheel

$$R_3 = 24{,}000 - 11{,}230 = 12{,}770 \text{ lb.}$$

The reaction on the tractor front wheels is

$$R_1 = \frac{8000(74 + 20) + 12{,}770(23 + 6)}{128} = 8770 \text{ lb.}$$

The reaction on the tractor rear wheels is

$$R_2 = 8000 + 24{,}000 - 11{,}230 - 8770 = 12{,}000 \text{ lb.}$$

It can be easily shown that the static load distribution of this tractor-trailer combination is as follows: Tractor front wheels, 5075 lb; tractor rear wheels, 12,525 lb; trailer wheels, 14,400 lb. Thus, maximum brake application in this particular case reduces the load on the trailer wheels from 14,400 to 11,230 lb, or about 22 per cent; reduces the load on the tractor rear wheels from 12,525 to 12,000 lb, or about 4 per cent, and increases the load on the tractor front wheels from 5075 to 8770 lb, or about 72 per cent. The proportional weight transfers, of course, increase with the heights of the centers of gravity and as the lengths of wheelbase decrease.

Heat Generation and Operating Temperatures—The chief problem in connection with vehicle brakes is to keep them from attaining excessive temperatures in service, as such temperatures accelerate the wear of the friction linings and may have injurious effects on the brake drums or discs. The maximum rate of heat generation by the brakes varies with the maximum speed of which the vehicle is capable, and the maximum temperature reached by the brakes varies with the average speed of travel. Both of these factors have increased continuously with increase in the proportion of engine power to vehicle weight, with improvement in roads, and with streamlining of the vehicles. On the other hand, the ability of brakes to get rid of the heat they generate has decreased, as the reduction in wheel diameters has made it necessary to use smaller brake drums, and ventilation of the brakes has been impaired by modern styling, which encloses both front and rear wheels in deep, streamlined fenders, and further reduces air circulation over the brakes by the provision of deep bumpers. In trucks the braking problem has been rendered more difficult by increase in gross vehicle weights and operating speeds.

In the brake the heat is generated at the interface between the drum or the disc and the lining, and most of it must flow off through the drum or disc, as the lining is a very poor heat conductor. Under severe braking conditions heat is generated at a very rapid rate, and the time available is too short to allow much of it to be dispersed to the atmosphere during a single stop. Therefore, in an emergency stop from a high speed most of the heat generated must be absorbed by the drum or disc. The material close to the friction surface reaches a high temperature, and there is a steep temperature gradient along the path of heat flow.

Let us assume that one of the smaller passenger cars weighing 4000 lb with load has to make an emergency stop from 60 mph. At that speed the friction coefficient between tire and road may be 0.5, which means that if the brakes are applied to near the locking point the retarding force will be about 2000 lb. Since at 60 mph the car travels one mile or 5280 ft per minute, the horse power absorbed at the beginning of the braking period is

$$\frac{2000 \times 5280}{33{,}000} = 293.$$

Some of this power, of course, is absorbed in overcoming air and road resistances, but by far the greater part is converted into heat by the brakes. Allowing 60 hp for overcoming air and road resistance, there remains 233 hp to be taken care of by the brakes. This represents a heat flow of 1080 Btu per minute from a surface of about 170 sq in. Fortunately, the rate of heat generation decreases in direct proportion to the

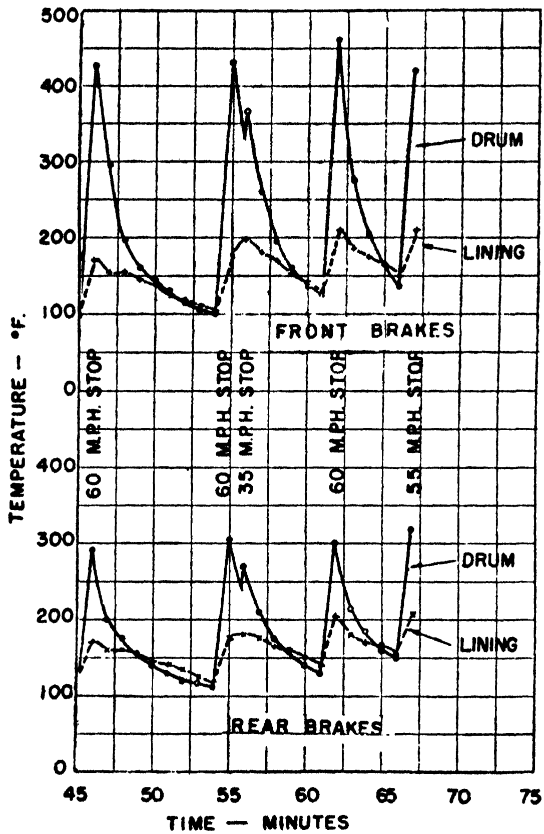


Fig. 6.—Brake Temperatures During Stops and Accelerations.

vehicle speed, and the whole stopping process takes only 5.5 seconds. A single stop of this kind is likely to raise the temperature of the drum by from 350 to 400 F.

Fig. 6 is part of a record of a brake-temperature test reproduced from an S.A.E. paper by T. P. Chase of the GMC Research Laboratories. The figures along the base line represent time from the beginning of the test. That the front brakes reach higher temperatures than the rear ones is due to the fact that they carry the greater part of the brake load, as is usual in passenger cars. It took from 15 to 20 minutes for the brakes to cool to their original temperature after a stop from 60 mph, and near the end of the cooling period the drums were cooler than the shoes, while at the beginning of that period they were much hotter.

With trucks the braking conditions are most difficult when long, steep grades have to be descended with heavy loads. The lower the speed to which the vehicle is held, the cooler the brakes will remain, because the total energy to be absorbed in the descent is nearly independent of the speed, and the rate of heat generation is smaller at low speeds. To prevent the brakes from overheating under such conditions, the descents usually are made in low or second gear, with the ignition cut off, in which case the engine acts as a brake. The braking power of an engine can be increased by so changing the valve timing that air is compressed in the cylinders during each up-stroke, and is allowed to escape at the end of that stroke, or by closing a valve in the exhaust line. Truck engines with either changeable valve timing (sliding camshaft) or an exhaust shut-off valve have been widely used in the Alpine regions of Europe.

Brake Drums—Three different types of material, all ferrous, are available for brake drums. Up to about 1930 practically all drums were cold-pressed from low-carbon steel. Such drums had advantages of low cost and relatively low weight, but they were open to the objections that in severe service they scored badly, especially when used together with the harder grades of lining, and that the brakes usually produced an annoying squeak when applied hard. Low-carbon steel is naturally quite soft, and it is further softened when its temperature is raised by frictional heat. Hard brake linings then pick up small particles of metal from the drum, and these metal particles—possibly hardened by chilling—score the drum. A scored drum causes the lining to wear rapidly.

An increase in the carbon content makes the steel harder and reduces metal pick-up by the lining, but the highest carbon content with which cold-pressing is still possible is only about 0.30 per cent. Most of the regular cold-pressed steel drums were made of 0.25 carbon steel (S.A.E. 1025). Higher carbon steels with about 0.70 per cent carbon also have been tried, and they give good results, but their cost is against them. In the first place, they must be formed hot, which already adds to production costs, and for best results they must be heat-treated. Moreover, such drums cannot be machined, owing to hard spots in them, and grinding is too expensive for quantity production. It may be pointed out in this connection that pressed-steel drums originally were used without machining. Later the demand for increased braking power made it necessary to increase the mechanical advantage of the pedal, which could be done only by decreasing the clearance between linings and drum in the released position. To permit of a small clearance, the drums had to be finished on their working surfaces.

Cold-pressed steel drums have been made also of a high-manganese, low-carbon steel, containing 1.40-1.70 manganese instead of the usual 0.50-0.80 per cent, and this is said to reduce scoring materially.

Cast-Iron Drums—It had been observed quite early that cast iron gives an excellent surface for brake lining to work against; it takes on a high polish in service, and is much less prone to score than low-carbon steel. The favorable effect is generally credited to the graphite flakes in the structure, and is said to be dependent on both the quantity and the distribution of the graphite. Users of commercial vehicles were troubled most by unsatisfactory life of brake linings and brake drums, and they were first to adopt cast-iron drums. Cast drums of alloy iron are now widely used for commercial vehicles of all kinds.

In the case of passenger cars the change from pressed-steel to cast-iron brake drums was beset with difficulties, for in addition to increasing the cost of the drums (owing to the extra machining required), cast drums would materially increase the unsprung weight, thus impairing the riding qualities and increasing the tendency to shimmy and tramp, which already was giving considerable trouble at the time. The problem was solved by the development of composite drums with a pressed-steel backing into which iron was cast to form the friction surface.

While cast iron does not score nearly as easily as mild steel, in very severe service the surface of the cast-iron drum will check or crack. This is due to the fact that in such service the surface of the metal often heats very quickly, and the heat cannot flow off sufficiently rapidly, so that there is a very steep temperature gradient from the surface inward. This sets up stresses in the surface layer which may exceed the strength of the material and cause numerous fine cracks to form. Examination of such surface defects usually reveals that the cracks are filled with iron oxide, which is indicative of the high temperature reached by the surface layer. The cracks usually follow the outlines of the graphite flakes, and it has been found that irons with a dense grain and finely divided graphite particles are less given to this thermal cracking than ordinary irons. Alloying the iron with nickel, chromium and molybdenum adds to the life of the drums, but the over-all economy reaches its maximum value for comparatively small values of these alloying elements. Centrifugal casting, which produces a dense grain, also has a beneficial effect.

Specifications for Brake-Drum Irons—Specifications of two cast irons for automotive brake drums have been standardized by the S.A.E. No. 113 is a high-carbon iron suitable for use where heat-checking is a problem. It contains a minimum of 3.40 per cent total carbon, 1.10-1.70 silicon, 0.60-0.90 manganese, 0.14 sulphur, and 0.20 phosphorus, the alloy contents being left open. The iron must have a Brinell hardness of 179-229, a minimum transverse strength of 2200, a minimum deflection of 0.20 in., and a minimum tensile strength of 30,000 psi. The graphite is to be Type A, size 2–4, and the matrix, lamellar pearlite, with ferrite (if present) not to exceed 15 per cent.

For heavy-duty drums, where both resistance to heat-checking and great strength are required, No. 114 is recommended. For this iron a minimum of 3.40 per cent total carbon is mandatory; contents of silicon, manganese, sulphur and phosphorus are the same as for No. 113, and the alloy contents are to be so chosen that the physical requirements can be met. This iron must have a Brinell hardness of 207-269, a minimum transverse strength of 2600, a minimum deflection of 0.27 in., and a minimum tensile strength of 40,000 psi. The graphite is to be Type A, size 3-5, and the matrix, fine lamellar pearlite, with an excess constituent of either free cementite, free ferrite, or both, not in excess of 5 per cent.

Design of Drums—Three types of brake drums are shown in Fig. 7, the drawing in each case being an axial section of one-half of the drum. That on the left is the conventional pressed-steel drum which was in almost universal use up to about 1930. When the brakes are applied the drum tends to distort, owing to non-uniformity of the radial pressure of the shoes, to heat generated by the braking action, and to internal stresses in the metal set up during the forming process. Heat tends to expand the drum; expansion is resisted by the internal flange on the "closed" side, and to prevent "bell-mouthing" of the drum under these conditions, it was usually provided with an external flange on the open side, which, of course, also resists mechanical distortion.

The view in the center shows a composite brake drum which has been used by a number of British manufacturers in conjunction with a proprietary brake. The rim is made of rolled section steel with a carbon content of 0.40-0.45, which after heat treatment shows a tensile strength of 125,000 to 130,000 psi. A length of this material is rolled into a circle, and the ends are welded electrically to form a rim of great rigidity (owing to the ribs) and great wear resistance. It is piloted on a disc of mild steel, to which it is secured by means of threaded dowels, the ends of which are riveted over and welded. Welds are made also at intermediate points, but there is no continuous joint between disc and rim, and this, together with the fact that the two are made of dissimilar metals, is said to result in a non-sonorous assembly with little tendency to brake squeak and squeal.

At the right in Fig. 7 is shown a half-section of the Centri-

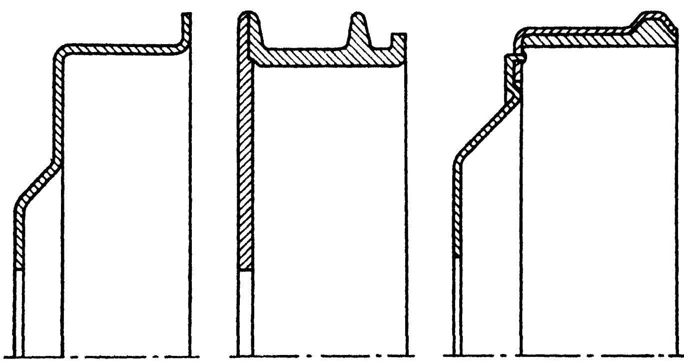


FIG. 7.—THREE TYPES OF BRAKE DRUM.

fuse, a composite brake drum with a pressed-steel backing into which iron is cast centrifugally. The backing consists of hot-rolled strip ranging in thickness from 16 gauge (0.062 in.) to ⅛ in. Strips of the correct length are rolled up into rings and welded electrically. After being trued and slightly expanded, the ring is put through another roll in which the section is formed. A rib usually is rolled in near the open side. In the foundry, after the backing has been fluxed and preheated, the molten iron is cast in a fixture which rotates the backing at about 900 rpm.

An obvious advantage of composite drums is that the material can be distributed to good advantage from the standpoints of both rigidity and ability to absorb large amounts of heat quickly during short periods of severe braking.

Lining Materials—Brakes are lined with friction material composed chiefly of asbestos. Asbestos is a fibrous mineral consisting chiefly of calcium and magnesium silicates. There are a number of different varieties, of which that known as Chrysotile is most largely used for this purpose. It occurs in the form of rock, of which large quantities are mined particularly in the Province of Quebec, Canada. The material, after being mined, is subjected to various mechanical processes to reduce it to the fibrous state, to remove impurities, and to grade the fibers according to length. From the standpoint of its use for brake linings, the most important physical property of asbestos undoubtedly is its high heat resistance, though, of course, its relatively high coefficient of friction in contact with iron or steel also is important. Heat begins to affect asbestos only at about 600 F. At 1000 F the water of crystallization is driven off rapidly, and the material loses its crystalline structure.

Types of Linings—There are four different types of asbestos-base friction material for brakes, as follows: (1) molded pulp; (2) impregnated asbestos sheet; (3) folded and compressed fabric, and (4) woven strip. Molded pulp linings are made of short-fiber asbestos which is mixed with fillers and bonding media, such as asphalts, rubber, synthetic resins, drying oils, etc. Linings in the forms of flexible strips or rigid segments are produced by hot-pressing in hydraulic presses. Impregnated asbestos sheet, which is marketed in the form of rigid segments, is made from plain asbestos sheet by impregnating it with a suitable binder, pressing to shape, and then curing or heat-treating it. Folded and compressed brake lining, which generally is marketed in roll form, is made of woven asbestos cloth by impregnating it with a compound of rubber and pigments or fillers, plying together in strips to the desired width and thickness, and then compressing it hydraulically and curing or vulcanizing. Woven linings come both in roll form and as semi-rigid segments. They are made either by weaving asbestos yarn into a tape, impregnating it with a saturant, rolling to size and curing, or by weaving asbestos yarns which already have been impregnated with fillers and binders, rolling to size, and curing.

Asbestos Yarns—Owing to the shortness of the fibers and their fragile nature, it is impossible to produce strong, tenacious yarns from asbestos alone, and either brass wire or cotton is introduced to obtain yarns better adapted to the weaving process. Both wire and cotton may be used in combination. The addition of cotton and wire results in a harder fabric, the wire especially making it possible to twist the yarns more tightly. Cotton, however, reduces the ability of the yarn to withstand high temperatures, and therefore must be used sparingly. In addition to adding to the strength of the yarns, cotton facilitates the penetration of the impregnating compound or saturant. Brass wire has been used for asbestos yarns for many years, and in recent years wires of lead and zinc, and various alloys of copper, lead and zinc, have been made use of. Lead is said to impart valuable properties to the lining in that it acts as a lubricant and inhibits the formation of abrasive metal particles, thus tending to prevent scoring of the drums. At the very high temperatures reached in severe service, both the lead and the zinc disappear, leaving a surface of non-metallic fibrous material in contact with the drum.

Saturants—Impregnating compounds or saturants are employed to make the fabric impervious to oil, grease and water, to improve its frictional characteristics. Without a saturant the fabric would absorb a great deal of moisture in wet weather, and would then cause trouble from swelling of the lining and seizure of the brakes, or from loss of frictional properties. Most of the same materials which in the case of woven linings act as saturants serve as binders or binding agents in the case of molded linings.

Among the saturants used for friction materials are asphalts, natural gums and oils, and synthetic resins. Asphalts have good impregnating properties when used together with suitable solvents, and impart to the lining a coefficient of friction which is substantially constant throughout a wide temperature range. At high temperatures, however, they become plastic, and to raise their plasticizing temperature, it is now common practice to add natural gums and oils, such as linseed oil, tung oil, kauri and copal resins. Rubber, another natural gum, also is being used as a bonding agent, but not in combination with asphalt. The good impregnating properties imparted to asphalts by the addition of natural oils and gums increase the wear resistance of the linings. Some of these addition agents are claimed to also prevent softening of the binder at high temperatures and consequent "bleeding." Asphalt-bonded linings usually have a friction coefficient ranging between 0.30 and 0.40 up to 500 F. Above this temperature the bonding medium is gradually driven off, and the coefficient of friction rises. New linings sometimes have a tendency to stick, which is due to a surface film of the saturant, with a friction coefficient which may be as high as 0.65.

Linings with a bonding medium containing vegetable oils and gums have a friction coefficient of 0.35-0.45 up to 500 F.

Wearing qualities of such linings are said to be somewhat better, as a rule, than those of linings with a pure asphalt bond, and with some of these linings there is said to be no "bleeding" under any operating conditions. With synthetic resins (such as bakelite) used as bonding material, the friction coefficient ranges between 0.40 and 0.50 up to 450 F, beyond which temperature it drops gradually to 0.20.

Self-Energizing Effect—In band brakes there is what is known as a wrapping effect, which influences the operation. When the band is applied to the drum, the friction on that portion of it between the anchor and the free end toward which the drum moves in its rotation helps to apply the band to the drum, thus increasing its retarding effect. The friction of the band on the drum due to a certain force applied to its end increases rapidly with the length of arc covered by the band. This phenomenon is quite familiar to sailors and dockmen, for by winding a rope several times around a post and holding on to its end, one man can resist a force many times as great that may act on the other end.

In the case of brakes with rigid shoes it would be inappropriate to speak of a wrapping effect, but there is a similar effect in these brakes too, in that the friction between shoe and drum assists the pedal pressure in forcing one of the shoes against the drum. There it is referred to as a self-energizing or self-actuating effect. In a conventional two-shoe brake, while there is a self-energizing effect on the leading shoe, there is a de-energizing effect on the trailing shoe, the normal pressure and frictional force of the latter due to a certain pedal pressure or force of application being reduced in the inverse ratio of that in which it is increased on the leading shoe. The degree of energization (and de-energization) varies along the length of the lining of each shoe, and for the shoe as a whole it depends on the arc covered by its lining and on the location of that arc relative to the anchor pin. Referring to Fig. 8, let N be the reaction of the drum on a very narrow strip of the lining at point A, and let it take a force P applied to the toe of the shoe to produce this reaction when the drum is at rest. In that case P and N are the only forces acting on the shoe (aside from the reaction of the anchor pin), and their moments around the anchor pin therefore must be equal and opposite:

$$Pa = Nb$$

and

$$P = \frac{b}{a} N.$$

When the drum is rotating right-handedly (as when the vehicle is being retarded by the brakes) there is an additional force $F = \mu N$ acting on the shoe, where μ is the coefficient of friction between lining and drum. Assuming the normal force N to be the same as before, the force of application will be smaller, and may be designated by P'. The equation of equilibrium now reads

$$P'a + \mu Nc = Nb,$$

from which we get

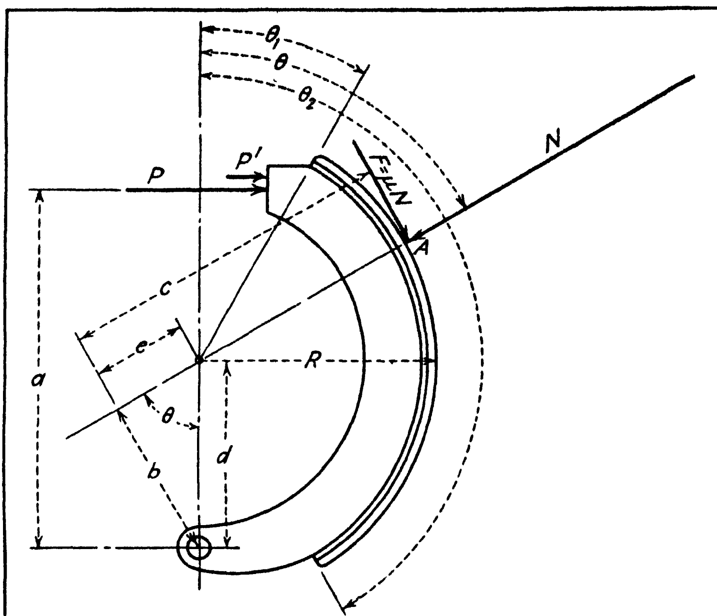


FIG. 8.—DIAGRAM OF FORCES AND REACTIONS ON BRAKE SHOE.

$$P' = \frac{b - \mu c}{a} N.$$

Self-energization thus reduces the force of application required from P to P' or from

$$\frac{b}{a} N \text{ to } \frac{b - \mu c}{a} N.$$

The difference, $P - P'$, is the self-energizing force, and the ratio $(P - P')/P$ has been called the actuating factor. If we substitute the values found for P and P' in the foregoing in this expression we get

$$\frac{P - P'}{P} = \frac{\frac{b}{a}N - \left(\frac{b - \mu c}{a}\right)N}{\frac{b}{a}N}$$

which when simplified gives

$$\frac{P - P'}{P} = \frac{\mu c}{b}.$$

Now it can be seen from Fig. 8 that $c = R + d \cos \theta$, and $b = d \sin \theta$. Making these substitutions we get for the actuation factor

$$\frac{P - P'}{P} = A = \mu \frac{(R + d \cos \theta)}{d \sin \theta}.$$

In Fig. 9 values of the actuating factor for friction coefficients of 0.3 and 0.4 are plotted over the length of the brake lining in degrees as a base. The mean height of the ordinates of each curve is the actuating factor for the whole shoe. It will be seen that when $\mu = 0.3$ the actuating factor for the whole self-energizing or leading shoe is 0.52, and when $\mu = 0.4$ it is 0.67. Therefore, with the smaller friction coefficient we have

$$\frac{P - P'}{P} = 0.52,$$

and $P = 2.08P'$. For the de-energizing or trailing shoe we have

$$\frac{P' - P}{P} = 0.52,$$

and $P = 0.66P'$. That is to say, for a given actuating force P' the leading shoe produces 2.08 times the braking torque of a shoe without self-energization, while the torque of the trailing shoe is only 0.66 times that of a non-self-energizing shoe. The leading shoe therefore produces $2.08/0.66 = 3.15$ times as much torque as the trailing shoe and does more work in the same proportion. In the same way we find that with a friction coefficient of 0.4 the leading shoe produces 3 times and the trailing shoe 0.6 times the braking torque of a shoe without self-energization, hence the former does 5 times as much work as the latter. For a normal friction coefficient of

0.35 the leading shoe does about 4 times as much work as the trailing one.

The foregoing analysis is based on the assumption that all parts of the brake are perfectly rigid. In practice this does not hold true, and the actual actuation factors therefore are somewhat smaller than those calculated.

If the actuation factor is too large, that is, if either the friction coefficient is too high or the lining comes too close to a point diametrically opposite from the anchor pin, brake

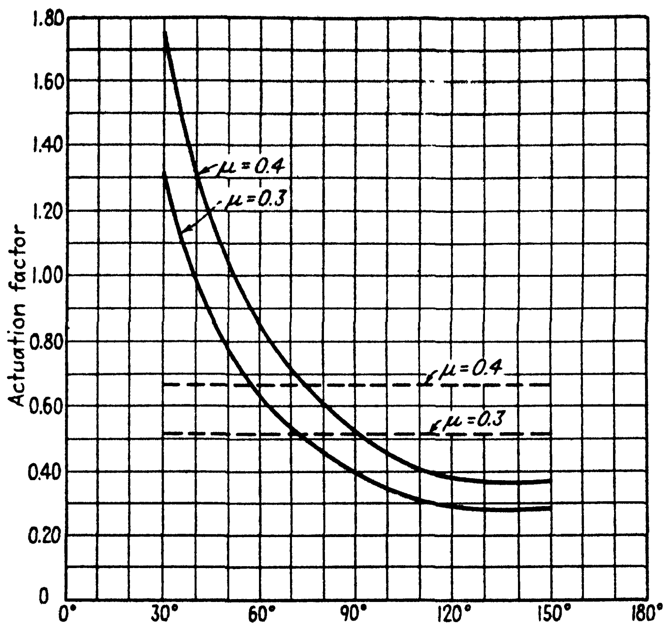


FIG. 9.—CHART OF ACTUATION FACTORS FOR TWO FRICTION COEFFICIENTS.

action becomes unstable, and the brakes have a tendency to lock. It has been recommended that the quotient of the actuation factor by the friction coefficient be limited to 1.4.

The brakes must be able to hold the vehicle on both an up-grade and a down-grade, and must be able to stop reverse as well as forward motion. In brakes of the type discussed in the foregoing, the shoe which is de-energized while the vehicle is moving forward is being energized during reverse motion, and such a brake is equally effective for both directions of motion. Special types of brake in which the self-energization effect is taken advantage of to further reduce the pedal pressure required are described in the latter part of this chapter.

Pressure Distribution over Brake Shoe—Since the shoe of a conventional brake can move only in one way, viz., angularly around its fulcrum on the anchor pin, when the brake is applied, any point on the lining surface moves in a direction at right angles to a line connecting it with the fulcrum, and at a velocity proportional to its distance from the fulcrum. For instance, point *B*, Fig. 10, will move in the direction *BD*, and we may let *BD* represent both the direction and the magnitude of its motion. This motion may be resolved into two components, a radial motion *BC* and a tangential motion *CD*.

Now suppose that while the brakes are being applied, the

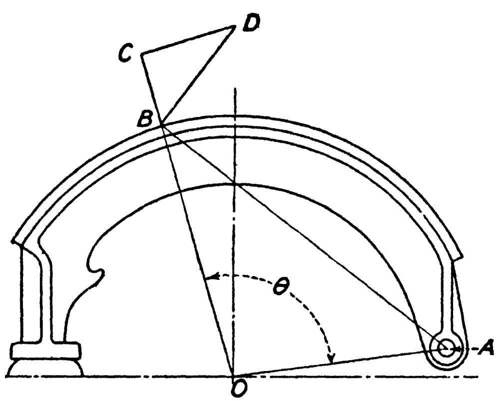


FIG. 10.—RADIAL AND TANGENTIAL COMPONENTS OF MOTION OF A POINT ON THE BRAKE SHOE.

lining has barely come into contact with the drum, and that there is as yet no appreciable pressure between the two. Any further motion of the shoe can take place only by reason of the elasticity of the materials in contact, those of the lining and the drum. Such further motion will result in the lining being compressed and the drum expanded, and the compression and expansion at any point of the surface will be proportional to the radial motion *BC*. Now, according to the theory of elasticity, compression or expansion (extension) is directly proportional to the force causing it, hence the pressure at point *B* will be directly proportional to the radial motion represented by *BC*, and it may also be represented by this vector. It can be shown that the radial motion, and therefore the local pressure, varies as the sine of the angle θ

between radial lines through the axis of the anchor pin and the point of contact, respectively. As the sine has a maximum value of 1.00 for an angle of 90°, the pressure is a maximum at 90° from the anchor pin and drops uniformly to both sides of that point. It would become zero at both ends of a diameter through the anchor pin if the shoe extended over one-half the circumference. Usually the lining extends over only about 120°, and if it is symmetrically disposed with relation to a diameter through the anchor pin, the lining will extend from 30° to 150° from the pin. Since the sine of 30° and 150° is 0.5, the specific pressure will be one-half as great at the ends as at the middle of the shoe.

Lining Wear—Wear of the linings is substantially proportional to the work done by the shoes, and it is the general experience that with shoes covering the same arc of circumference, lined with the same kind of friction material, and subjected to the same pressure at their free ends by the cam or brake cylinder, the self-energizing shoe will wear more than twice as fast as the other. The rate of wear also varies along the length of the shoe, in accordance with the pressure distribution, being a maximum at the middle and decreasing toward the ends. This non-uniform wear is taken into account in the design of some brake shoes in which the linings, instead of being made of uniform thickness over their whole length, are made crescent-shaped as it were, thickest at the middle. When such a brake is relined, the portion of the lining discarded is of substantially uniform thickness throughout, whereas with conventional linings it is much thicker at the ends than at the middle.

Arcuate Length of Linings—For smooth brake action the lining of any one shoe must not extend over too large an arc. If the expanding cam were located substantially 180° from the anchor pin and the lining came close to the cam, the brake would have a tendency to be self-locking, so that if once applied it could be released only by reversing the motion of the vehicle. The self-locking tendency is the more pronounced the greater the arcuate distance of the toe end of the lining from the anchor, and the higher the coefficient of friction between lining and drum. As shown in Fig. 11 locking occurs if the resultant of the equivalent concentrated normal pressure and the frictional force passes through the anchor-pin axis. In two-shoe brakes it has been customary to limit the arc of each lining to 120° and to locate the lining substantially symmetrically with respect to a diameter through the anchor-pin axis. The ends of the lining are usually skived off.

Optimum Length of Linings—The problem of the best arcuate length of linings has received further study in recent years, more particularly from the standpoint of the effect on the ability of the drum to get rid of the heat generated. An interesting series of dynamometer tests bearing on this problem were made by the Timken-Detroit Axle Company and were reported on in an S.A.E. paper by Ralph K. Super. The tests were made on a conventional two-shoe, fixed-anchor, air-operated, heavy-duty bus brake. Originally the linings had a length of 116° each, and this was reduced by steps of 10° (5° at each end), to a minimum of 56°. Readings were taken of the temperatures of the outside and inside of the drum, the air pressure required to produce a predetermined

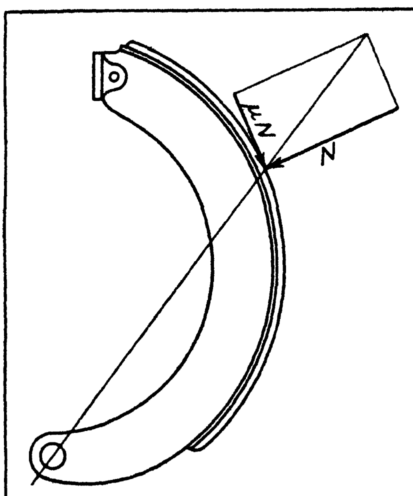


FIG. 11.—ILLUSTRATING CONDITIONS THAT MAKE A BRAKE SELF-LOCKING.

rate of deccleration (10 fps^2), and of the wear of the linings. The results are plotted in Fig. 12. It will be seen from the chart that the air pressure required to produce the predetermined deceleration was a minimum for a shoe length of about 91°, while the drum temperatures were lowest for a shoe length of 86°. Test runs covered a series of 500 stops, and the lining wear during a run remained substantially constant with decreases in the shoe length down to 86°, but it increased rather rapidly when the shoe length was decreased further.

One reason why the brake runs cooler if the length of the lining is reduced below its normal value is that more heat is then dissipated from the exposed inside surface of the drum.

In heavy-duty brakes the shoes usually are mounted on open spiders, and the brakes have no dust shield, hence air can circulate through the brake and carry off heat from the inside of the drum, which normally is much hotter than the outside surface. The temperature of the outside of the drum seldom rises above 400 F, while that of the inside may reach 1100 F, so that even at high atmospheric temperature the temperature

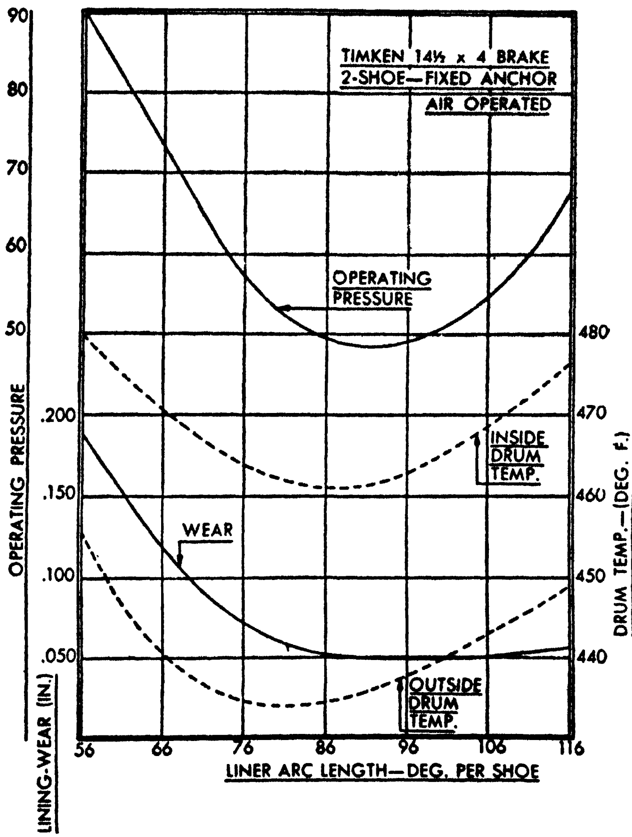


FIG. 12.—EFFECTS OF CHANGES IN THE ARCUATE LENGTH OF BRAKE LINERS.

difference between the drum and the ambient air is more than three times as great on the inside as on the outside of the drum, and a relatively small increase in the exposed inside surface results in a considerable increase in the total heat dissipation.

Buick in 1951 equipped a large passenger car with a Duo-

Servo brake—a brake with both a primary and a secondary shoe—in which each shoe carried three cemented strips of lining, separated by rather wide gaps, the total length of lining on the primary shoe being approximately 56°, and that on the secondary, 80°. Keeping down the total length of the lining helps to keep down the maximum temperatures of drum and linings, while dividing the lining on each shoe into a number of segments was said to be of help in keeping the friction surfaces clean and to remove water when the brakes are flooded. The reason for using a greater total length of lining on the secondary than on the primary shoe is that the normal pressure on the former is much greater, and the unequal lengths of lining tend to equalize the unit normal pressures and hence the rates of wear of the two shoes.

Thickness of Linings—Thicknesses of linings for passenger-car brakes range between $\frac{5}{32}$ in. and $\frac{1}{4}$ in., but most models have $\frac{3}{16}$-in. linings. Up to recently brake linings almost invariably were secured to the shoes by rivets. They were drilled and countersunk so that at least one-half the thickness of the lining could wear away before the rivet heads contacted the drums. Brakes on heavy trucks and trailers are provided with heavier linings, up to $\frac{3}{4}$ in. in thickness, and these heavier linings sometimes are secured to the shoes by means of flat-head bolts, lock washers and nuts, for convenience in relining. Rivets for fastening the linings have been standardized by the S.A.E.

Brake linings also can be cemented to the shoes, by means of heat- and moisture-proof adhesives. This method of fastening has the advantage that the linings can be used until they are practically completely worn away, whereas with the usual method of fastening more than one-half of the total lining material must be discarded. Brake shoes with linings cemented to them were first put on the replacement market in 1940, and toward the end of the forties they had become standard equipment on several makes of car. In the production of such shoes at a Chrysler plant the linings are first wire-brushed on the inner side to remove any dirt, and then have two bands of rubber cement applied to them by extrusion, a narrow strip at the middle of the lining being left uncovered. The cement is dried in a steam-heated oven, and as the linings emerge from that oven they are applied to the shoes in a fixture under a pressure of several hundred pounds, depending on size, the cementing surfaces of the shoes having previously been bonderized. Next the cement is cured for 20 minutes at 350 F, and this is followed by inspection, spray-painting, oven-drying, and grinding of the friction surface.

Hydraulic Brakes—As shown in Fig. 13, the hydraulic braking system normally consists of a master cylinder mounted on the vehicle frame substantially in line with the brake pedal, from which tubular connections are made to wheel cylinders at the four brakes. When the brake pedal is depressed, the piston is forced into the master cylinder, and brake fluid is forced from the later into the line and from the line into the wheel cylinders. Pistons in the wheel cylinders are thereby forced apart, and as a result, the brake shoes are pressed against the brake drums.

Brake Fluid—To be satisfactory under all conditions, a brake fluid must meet certain requirements. It must remain fluid at the lowest temperature at which a vehicle is likely to be operated, and, of course, it must not boil or evaporate at the highest working temperatures. It must have lubricating properties and must have sufficient body so that it can be confined without difficulty. Of course, it must not attack any of the materials with which it comes in contact in the system. A 50 per cent solution of castor oil in alcohol, with the addition of a neutralizer, meets these conditions satisfactorily. The neutralizer is needed to counteract any free acids there may be in either the castor oil or the alcohol.

Master Cylinder—A section through a combined master cylinder, fluid reservoir and pedal bracket is shown in Fig. 14. Within the master cylinder *A* below the fluid reservoir *B* there is a piston *C* which is provided with two rubber cups—a primary cup *D* providing a seal against fluid in the cylinder under pressure, and a secondary cup *E* preventing leakage of fluid from the cylinder. The piston forms a deep conical socket for the end of a pushrod *F*, the other end of which connects to the brake pedal. There are two ports between the reservoir and the master cylinder. Fluid from the reservoir enters the space around the piston through the larger port *G*, while the smaller port *H* serves to prevent building up pressure in the cylinder when the brake is released. There is a piston return spring in the cylinder between the piston and the cylinder head. In the cylinder head there is a double check valve *I*, which will open under fluid pressure in both directions. When the pedal is being depressed and the fluid in the cylinder thereby put under pressure, the poppet-like valve in the center of the head is raised from its seat and permits fluid from the cylinder to enter the line, which results in application of the brake. When the driver releases the brake pedal, the brake-retracting springs draw the pistons in the wheel cylinders together, thereby forcing fluid back into the line. The line pressure immediately closes the valve in

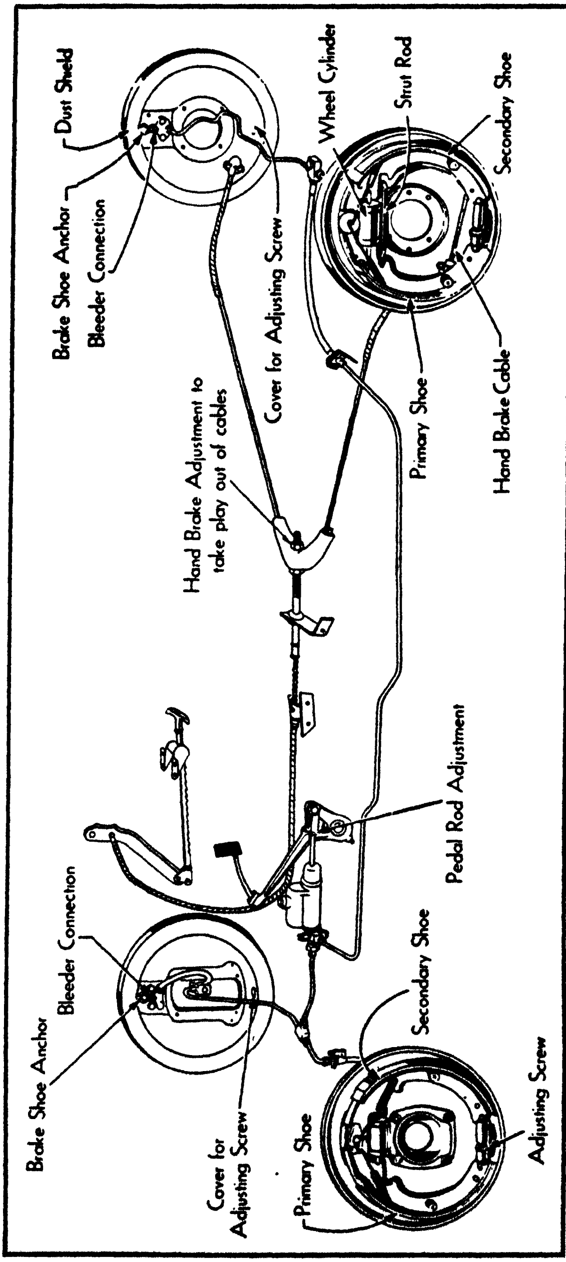


FIG. 13.—HYDRAULIC-BRAKE INSTALLATION ON PASSENGER CAR (CADILLAC).

the cylinder head through which the fluid entered, and presses against the disc serving as the seat for this valve and as a spring rest for the piston return spring, forcing it off its seat on the end plug, thus allowing fluid to return from the line to the cylinder. When the fluid pressure on the disc drops below the spring pressure on it, the disc returns to its seat, shutting off communication between the cylinder and the line. A pressure of between 10 and 15 psi is maintained in the line, to prevent air from filtering in if there should be any small leaks in the system.

As shown in Fig. 14, when the brake is fully released, the piston of the master cylinder uncovers the small relief port, and the fluid in the master cylinder is then under atmospheric pressure. When the pedal is being depressed, a very slight movement of the piston closes the relief port, and any further motion causes a pressure to be built up in the master cylinder,

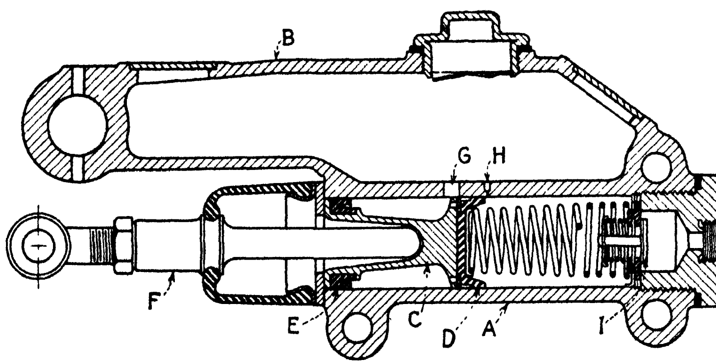


Fig. 14.—Section of Master Cylinder and Fluid Reservoir.

and the brakes to be applied. When the brake is being released, owing to the resistance to flow due to the valve in the master cylinder head, fluid cannot enter the cylinder from the line as fast as the release motion demands, and to keep the cylinder filled during this period, a small amount of fluid from the reservoir and the space surrounding the piston is allowed to pass through holes in the piston crown and past the lip of the primary piston cup. When the piston completes its return stroke (the release stroke), the relief valve is fully uncovered, and any additional fluid returning from the line can pass on into the reservoir. The return motion of the piston is limited by a piston stop secured to the end of the cylinder. Sometimes this piston stop is made to serve also as a pedal return stop, by providing the piston pushrod with a collar which ends up against the piston stop at the end of the release motion.

Compound Master Cylinder—A compound master cylinder for truck brakes is being manufactured by the New York Air Brake Company. It comprises two coaxial cylinders, of which the larger one is stationary, while the smaller one is movable within the larger one and forms its piston. During the first part of the pedal movement the large piston moves in its cylinder and forces fluid into the line, because it has a lighter return spring than the small one. Owing to the large diameter of this cylinder, the brake clearance is taken up rapidly. After the clearance has been taken up, the line pressure increases, and when this pressure reaches a certain value the small piston begins to move in its cylinder and displace fluid through what is known as a staging valve, into an annular space surrounding its piston rod. This increases the mechanical advantage of the pedal, as the high pressure generated by motion of the small piston acts on an area equal to that of the large piston. With this compound master cylinder it is possible to apply the brakes by direct physical effort in larger trucks than with a simple master cylinder.

Wheel Cylinder—A sectional view of a wheel cylinder is shown in Fig. 15. It is a plain, open-ended cylinder in which there are two metal pistons provided with rubber cups. The cylinder is secured to the brake backing plate by cap screws. Each piston is formed with a spherical seat on its inside, to take the strut which transmits the fluid pressure on the piston to the brake shoe. The ends of the cylinder are sealed by rubber caps with reinforced edges lodged in grooves on the outside of the cylinder and on the strut. A spring within the cylinder holds the rubber cups in contact with the pistons. At the center of the cylinder there are two tangential openings, one of which takes a fitting to which the brake hose is secured, while the other takes a bleeder screw.

Ordinarily the wheel cylinder is straight, that is, of uniform diameter from end to end, but so-called stepped cylinders also have been used, with one large- and one small-diameter piston. In some cases the large piston actuates the non-energizing shoe, while in other cases it actuates the self-energizing shoe. The explanation of this apparent inconsistency is that entirely different objects are aimed at in the two cases. Where the larger piston acts on the non-energizing shoe, the object is to equalize the rates of wear on both shoes, whereas when it acts on the self-energizing shoe the object is to obtain a more powerful braking effect for a certain pedal effort. The force applied to the self-energizing shoe is multiplied by the self-energizing action and produces a correspondingly greater braking effect, hence from this standpoint it is advisable to apply more force to that shoe.

Parking Brake—Most cars equipped with hydraulic service brakes use the rear-wheel brakes also for parking purposes. The method of applying these brakes mechanically is illustrated in Fig. 13. A lever within the brake has its fulcrum on the secondary shoe near its upper end, and a strut is interposed between this lever and the upper end of the primary shoe. From the lower end of the lever a cable passes through a guide secured to the backing plate to an equalizer at the center of the chassis, from which connection is made through another cable to the brake lever fulcrumed on the cowl. In this particular design the driver does not grasp the brake lever when applying the parking brake, but instead

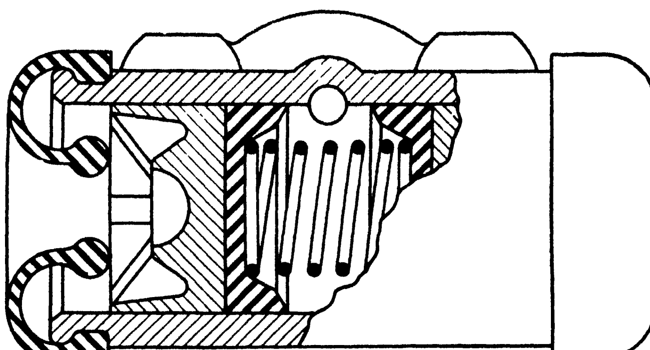


FIG. 15.—SECTION OF WHEEL CYLINDER.

pulls on a rod articulated to the lever. This rod, which extends through the instrument board, has ratchet teeth formed on it that engage a fixed pawl when the brake is applied. The shoes float, and the parking brake is equally effective for both directions of rotation.

Hill-Holder—In driving a car with conventional controls, an awkward situation arises when it has to be stopped—for a traffic light, for instance—on a steep up-grade and then started again. During the stop the driver has his feet on the brake and clutch pedals. To make a smooth, positive getaway, the brake must be released gradually and the clutch engaged simultaneously, but to prevent stalling of the engine, the throttle must be gradually opened at the same time. The driver cannot do this with the accelerator pedal, as he must keep his right foot on the brake pedal. Of course, most cars are equipped with a hand throttle, which is normally used under these conditions, but it is not entirely satisfactory.

For cars with hydraulic brakes a device known as the hill-holder has been devised, which automatically holds the brakes on under the above conditions, thus enabling the driver to shift his right foot from the brake pedal to the accelerator. As shown in Fig 16, it consists essentially of a check valve incorporated in the line from the master cylinder to the wheel cylinders, and it is usually mounted alongside the former. Within the body of the device there are a rubber seal and a steel ball enclosed in a cage. The position of the cage is controlled by a cam operated from the clutch pedal through a linkage. When the car is being stopped, the clutch

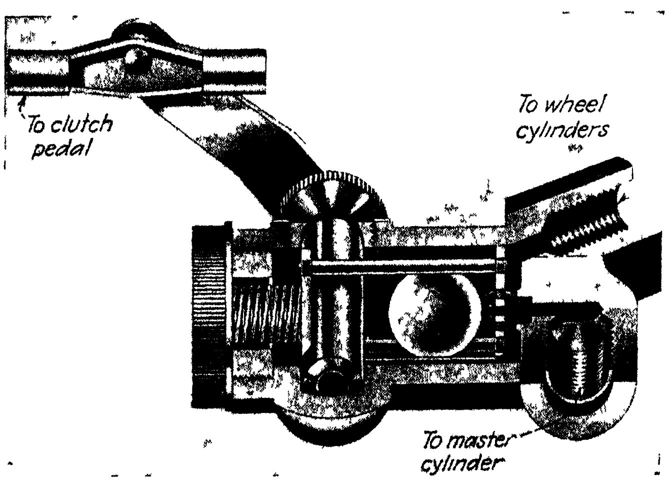


FIG. 16—SECTIONAL VIEW OF HILL-HOLDER

pedal moves the ball cage toward the rear (toward the right in the illustration), and because of the inclination (on the up-grade), the ball is at the rear end of the cage and the fluid passage through the hill-holder is closed. Thus the brakes remain on. When he wants to start again, the driver releases the clutch pedal, with the result that the ball cage is moved forward (toward the left), communication between the wheel cylinders and the master cylinder is reestablished, and the brake is released.

When the car is being stopped on a down-grade, the steel ball rolls forward in the cage, and communication between the wheel cylinders and the master cylinder is maintained through the rubber seal, hence it is not necessary to work against the brakes when backing the car up-hill.

Brake Shoes—The shoes of internal brakes usually are of T section, and are either cast or fabricated from sheet steel. Cast aluminum alloy shoes have been used to a small extent, and have the advantages of low weight and high heat-conductivity. However, the higher material cost is against them, and they also are at a disadvantage in designs where a hardened surface is required for a cam or other actuating member to work against. Brake shoes for heavy vehicles usually are malleable castings to which hardened terminals or cam followers are fastened. Where the shoe is quite wide relative to the diameter of the brake, it is stiffened by ribs between the web and flanges, or a channel section is used instead of the usual T section.

In the older type of cam-actuated brake, the shoes are provided with an eye or a section of an eye at one end, through which the braking force is transmitted to the anchor pin. In hydraulically-operated brakes, and also in some of the newer mechanically-operated ones, what may be called the terminals of the brake shoe are of very simple design. In some cases the web of the shoe is extended, narrowed down, and rounded at its end, which latter has a seating in the cup-shaped piston of the brake cylinder. Alternately, the end of the web may be slotted and engaged by an intermediate member similar in form to a yoked connector, the slotted end of the connector passing over the web of the shoe and the solid end entering the piston of the brake cylinder. Brake shoes of similar design can be used with mechanically-operated brakes, by providing intermediate members between the brake-shoe terminal and the actuating cam or its equivalent, these intermediate members sliding in guides corresponding to the wheel cylinders in the hydraulic system. In the case of a fabricated shoe, sufficient bearing surface on the anchor pin can be provided by riveting reinforcements or bosses to the web.

Lining Area Required—The amount of energy that must be dissipated when stopping a vehicle is directly proportional to the weight thereof, and since the energy-dissipating capacity of a brake varies substantially as its area of frictional contact, the latter should be proportional to the gross vehicle weight. There are, however, a number of other factors to be considered. As the kinetic energy of the vehicle varies as the square of its speed, more braking surface is required in high-speed vehicles than in those of moderate speed. Vehicles that have to make frequent stops, such as city buses, also must have brakes of extra liberal size, as otherwise their linings will not last long. Finally, the cooling facilities of the brakes and the ability of the lining material to withstand high temperatures without injury affect the minimum frictional area required. The modern trend toward streamlining has further increased the difficulty of the braking problem of relatively heavy vehicles, as the greater part of both front and rear wheels is enclosed in fenders or wheel housings, so that the brake drums are shielded from the slip stream.

From the foregoing it may be concluded that any rules regarding lining area required can serve only as a general guide. From the standpoints of effectiveness and long wear of linings, brakes cannot be too large, but in certain vehicle types, such as large buses, it is difficult to find room for brakes of sufficient size. In such vehicles the brake drums usually are located inside the wheel rims, and if they come too close to the rims, air circulation and heat dissipation are interfered with, and tire life may be cut short by excessive heating of the tires. Where disc wheels are used they should have ventilating louvers.

In passenger cars the service brakes usually are given one sq in. of lining area per 25 lb of gross vehicle weight. If the drum diameter can be made $0.17\sqrt{W}$ in., where W is the gvw in lb, and the lining width is one-sixth of the drum diameter, then the brake loading will be about 25 lb per sq in. if the lining extends over two-thirds of the circumference. Brake diameters usually are held to full inches and lining widths to quarter inches. In motor trucks the service brakes are given 1 sq in. of lining area for between 40 and 50 lb of gvw, the brakes of the heaviest trucks being most heavily loaded, because it is there difficult to provide sufficient area. In motor buses the service brakes are given 1 sq in. of lining area for from 18 to 20 lb of *chassis weight.* Buses usually are provided with parking brakes on the propeller shaft or transmission, and these have 1 sq in. of lining area for every 125 lb of chassis weight, on the average.

B. E. House in 1946, while chief engineer of Bendix Products Division, recommended a mean rate of energy absorption of 1200 ft-lb per second per sq in. of lining area, based on the maximum speed of the vehicle and the maximum deceleration. He said that while he usually aimed to hold the rate within that limit, he knew of cases where satisfactory results were being obtained with a rate of 1600 ft-lb per sec per sq in.

Brakes on Outer Side of Wheels—In the case of city buses, especially, a serious objection to brakes of inadequate size is the loss of time (and revenue) occasioned by frequent need for relining. It is of interest in this connection to mention that a German firm (M.A.N.) at one time built a bus on which the brakes were mounted on the outside of the rear wheels, so they could be relined without removing the wheels. These brakes were mechanically actuated, and a member of the actuating mechanism passed through a hollow spindle at the end of a cranked dead axle. Mechanical actuation of "outside" brakes seems impractical with live axles. Some U. S. Army trucks with full-floating axles are fitted with hydraulic brakes on the outside of the wheels, a construction which was adopted after it had been found that in military operations mud, sand, and salt water quickly destroy drums, linings, and actuating

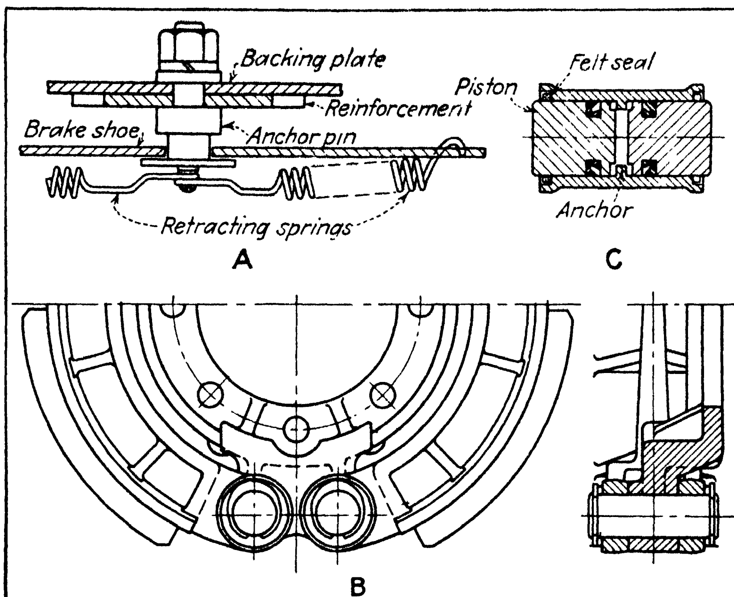


Fig. 17.—Three Methods of Anchoring Brake Shoes.

mechanism of conventional brakes. The brake backing plate, on which the wheel cylinders are mounted, is secured to the end of the axle tube, while the brake drum is secured to a driving plate or cover plate that is splined to the end of the axle shaft and bolted to the wheel. This construction keeps out abrasive and corrosive elements, and permits inspection and adjustment of the brakes without removing the wheels.

Brake Anchors—In connection with passenger-car brakes it has become the practice to secure the anchor pin to the brake backing plate, as shown at *A* in Fig. 17. As the center point of load application is at a distance from the backing plate equal to at least one-half the width of the shoe, and as the backing plate, moreover, usually is of rather light gauge, this does not make a very rigid mounting, and is not suitable for heavy vehicles. It can be readily imagined that under severe brake application considerable distortion takes place. When two anchor pins are used it is, of course, possible to stiffen the construction by slipping a spacer plate over their ends.

On heavy vehicles the brakes usually are anchored (and the actuating device is mounted) on a brake spider comprising a flat ring riveted to a flange on the axle housing, with two arms thereon, one for the brake anchor and the other for the brake-actuating device (*B*, Fig. 17). For heavy-duty brakes separate anchor pins seem to be preferred. The eye of the brake shoe is forked and spans the eye of the brake spider, so that the shoe is symmetrically supported, without overhang. In a recent design of hydraulic brake anchor pins are done away with, the braking force being taken up on an internal flange at the center of the hydraulic cylinder, as shown at *C* in Fig. 17. The braking force then is transmitted through the wheel cylinder to the backing plate, in which the cylinder is piloted and to which it is securely fastened by two cap screws.

Referring to view *C*, Fig. 17, it will be seen that the inner ends of the pistons are reduced in diameter. When the brake is applied there is a slight circumferential movement of the shoes, and the reduced portion of one piston telescopes into the internal flange, forming a chamber from which there is only a very restricted outlet for the fluid. This gives a dashpot effect and eliminates the metallic click usually heard when floating shoes end up against a solid anchor. Another feature shown in the drawing is a felt seal at each end of the wheel cylinder. It takes the place of the usual rubber boot, which is apt to be injured by the heat generated by the brakes in severe service.

Brake shoes are restrained laterally by snap rings set into grooves in the anchor pin. The eye of the brake shoe through which the anchor pin passes often is drilled out to a diameter somewhat larger than that of the pin. This enables the shoe to center itself in the drum, even if its mounting should be slightly off center, thus avoiding heel contact and toe contact and their adverse effect, such as grabbing and chattering, particularly in the case of toe contact.

Expander Mechanisms—The shoes of internal brakes may be expanded or forced against the drum by one or another of four different mechanical devices—the cam, the wedge, the toggle and the screw. Of these four the first two have been used most extensively. It is desirable that the mechanical advantage of the pedal (the ratio between pedal and brake-shoe motions) should increase as the brake goes on, because that results in the greatest braking effect for a given pedal pressure. During the first part of the pedal motion only the tension in the retracting springs and friction in the mechanism must be overcome, and for that a high reduction ratio is not needed. The above requirement is met by a flat-sided cam contacting a flat follower surface on the shoe. With this the mechanical advantage of the pedal is least in the released position, when the flat surfaces of cam and follower are in contact, and it increases with the angle between these surfaces as the cam is turned. The advantage of the pedal can be increased even more rapidly if the cam is "ovaled," as shown in Fig. 18.

These variable-ratio cams are suitable for brakes with comparatively thin linings, on which the point of contact on

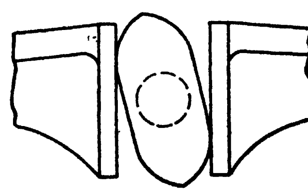


Fig. 18.—"Ovaled" Type of Brake Cam.

the cam corresponding to full brake application does not vary greatly throughout the life of the lining. For brakes with linings of ½ in. and over, it is usual to employ constant-ratio cams acting on rollers, so that a given pedal pressure will always produce the same braking effect, regardless of the state of wear of the linings.

Brake cams can be either rigidly supported (journaled) or allowed to float. A floating cam always will exert equal pressures on both shoes, while a rigidly supported one will not. Moreover, with rigidly supported cams there is no self-energizing action. For that reason floating cams are generally preferred for brakes that are to be actuated by muscular power.

Wedge Actuation—Brake-expanding wedges may have either a radial or an axial motion. Axially moving wedges are in extensive use in Great Britain, for one reason because they can be operated through a linkage of which practically all members are in tension and therefore yield less under pedal pressure. The expanding mechanism of the Girling, a British proprietary brake, is shown in Fig. 19. Steel plungers *3,3* are adapted to slide in a bore of the die-cast housing *5*, which is held to backing plate *6* by means of studs, nuts, and the spring washers *7*. The holes for the studs in the backing plate are oblong, and as the housing is not rigidly secured to the backing plate, but merely held in light frictional contact with it by the spring washers, it is capable of movement in the circumferential direction. It practically floats, and therefore is not subjected to severe stresses by the force of brake application.

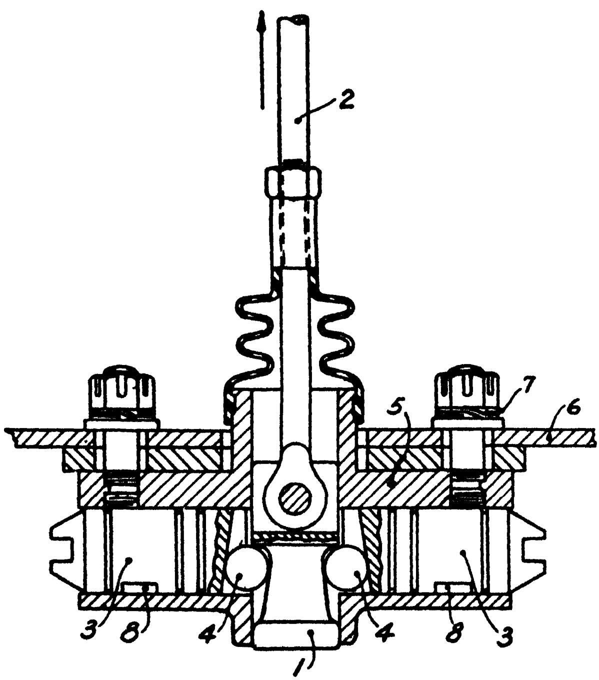


Fig. 19.—Girling Wedge-Type Actuating Mechanism.

Pullrod *2* is pin-jointed to expander wedge *1*. The inner ends of plungers *3,3* are cut off at an angle, and motion is transmitted from the expander wedge to the plungers through the intermediary of rollers *4,4*. The rollers roll up grooves in the plungers and down the wedge, and the wedge moves twice the distance of the rollers, and thus doubles the leverage due to the cone angle. This results in a high step-up ratio, so that for a given radial pressure of the shoes, less force needs to be transmitted through the pullrods. When the brake pedal is released the retracting springs return the shoes to the off position. The expander housing with its plungers, wedge, and rollers is a self-contained unit, the plungers being held in position by pins *8* when the shoes are removed.

More recently Girling has been producing hydro-mechanical brakes. These employ an expander mechanism similar to that described, but the wedge is directly connected to the piston of a hydraulic cylinder mounted on the brake backing plate.

Brake Clearance—Clearance adjustments always are made at the toe and the heel of the shoe, because when there is enough clearance at these points there will be enough everywhere. In two-shoe passenger-car brakes a clearance of either

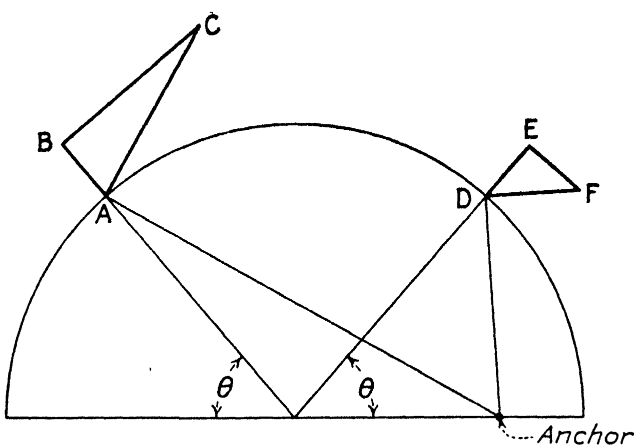


FIG. 20.—DIAGRAM SHOWING THAT HEEL AND TOE OF SHOE APPROACH BRAKE DRUM AT THE SAME VELOCITY.

0.010 in. or 0.015 in. is frequently specified, though some manufacturers call for as small a clearance as 0.006 in. While a small clearance increases the chance of instability (the tendency to become locked), it has the advantage of making possible a larger ratio of pedal movement to brake-shoe radial movement, thus making for lighter pedal pressures.

Specifying the same clearance for both heel and toe seems illogical at first glance, as the toe moves much faster than the heel, owing to its greater distance from the fulcrum. However, if both toe and heel are at equal angular distances from a diameter through the fulcrum, both will approach the drum at the same velocity. This is illustrated in Fig. 20, where point *A* near the toe is at twice the distance from the fulcrum as point D near the heel. The motion of point A, represented by AC, is twice that of point D, represented by DF, but the radial components of the motions of both points, AB and DE, are equal. It is therefore incorrect to specify twice the clearance for the toe as for the heel, as is sometimes done.

Pedal Travel—The pedal travel during a brake application—measured at the pedal pad—may be divided into three parts. The first part is required to take up the clearance between the piston rod and the piston in the released position, and amounts to between 1/8 and 1/4 in. The second part is required to move the piston sufficiently far to completely close the relief port, and amounts to between 5/8 and 3/4 in. The final part, which serves to move the brake shoes from the released to the applied position, is of the order of 1 in., so that the total pedal motion ranges between 1 3/4 and 2 in. Brake clearances specified by car manufacturers average close to 0.010 in., and if it takes a pedal travel of 1 in. to take up this clearance there is a ratio of 100:1 between pedal travel and brake-shoe radial travel. The pedal usually has a total range of close to 6 in. and the difference between this and the 1 3/4 to 2 in. required to fully apply the brakes is known as the pedal reserve. This rather large reserve is needed for two reasons. In the first place, during a rapid stop from a high speed the drum may reach a temperature 300 deg F higher than the shoes, and expand proportionately more. A temperature rise of 300 deg F will increase the radius of a 12-in. drum by about 0.010 in., and with a ratio of 100:1 this calls for an additional pedal travel of 1 in. The other reason for pedal reserve is that allowance must be made for wear of the linings. The figures given in this paragraph apply to passenger-car brakes.

Pedal Pressures—The pedal pressure required to produce a certain deceleration with a given brake installation can be expressed by an equation of the form

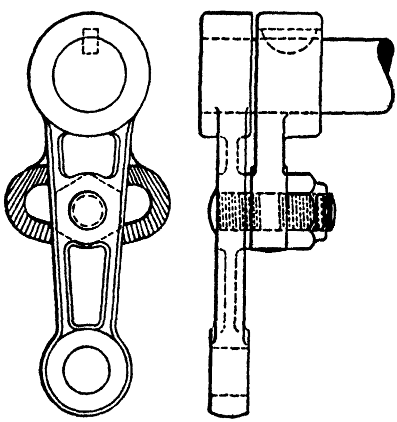
$$P = aD + b \text{ lb},$$

where D is the deceleration in fps^2. In this equation the constant b evidently represents the pedal pressure necessary to overcome friction in the linkage and fluid passages and the forces of the brake-shoe and brake-pedal retracting springs. Light pedal pressure is appreciated by most drivers and is, of course, particularly desirable in the case of lady drivers. But pedal pressures must not be too light, for if they are, there is apt to be frequent locking and skidding of wheels, which not only is destructive to tires, but is dangerous as well. One rule followed by passenger-car designers in the past has been to keep the pedal pressure required to produce the maximum "comfortable" deceleration of 10 fps^2 within the limit of 65 lb. General Motors has been using a chart or bogey giving upper and lower limits on the required pedal pressure as a function of the rate of deceleration. The upper limit is given by the expression $6.12D + 10$ lb and the lower limit by $4.75D + 5$ lb.

To get an idea of the magnitudes involved, let us assume a passenger car weighing 4000 lb with a full complement of passengers. To produce a deceleration of 10 fps^2 of such a car requires a retarding force of $(4000 \times 10)/32.2 = 1242$ lb. If the inside diameter of the brake drums is equal to 40 per cent the effective wheel diameter, the total frictional force on the four brake drums must be $1242/0.40 = 3100$ lb and if the friction coefficient of the lining is 0.35, the normal force necessary to produce this friction is $3100/0.35 = 8857$ lb. With non-self-energizing brakes and a ratio of 100:1 it would take a pedal pressure of 88.5 lb to produce this normal pressure if the mechanism were 100 per cent efficient, but allowing for friction and the tension of the retracting springs, we would have to figure on about 96 lb. This is substantially 50 per cent more than the average pedal pressure required in modern cars to produce the deceleration specified. The difference is accounted for by the fact that practically all modern passenger-car brakes are self-energizing. Self-energization usually increases the effectiveness of the brakes more than 50 per cent, and the ratio of pedal movement to radial motion of the brake shoe therefore may be less than 100:1 without necessitating a higher pedal pressure than the value called for.

Brake Adjustment—As the lining wears, the pedal position corresponding to full brake application moves forward, and unless something is done, the pedal pad eventually will end up against the toe board without the brake being fully applied. This can be prevented only by making an adjustment that will compensate for wear. In early brake installations this adjustment usually was made in the linkage. Shortening the links will serve to keep the pedal position corresponding to full brake application constant, but it is not entirely satisfactory from other points of view. For instance, in the new car the angular position of the arm on the brake shaft may be such that when the brake is fully applied, the link connecting to the arm makes a nearly right angle with it, so that practically the maximum mechanical advantage is obtained. After a number of adjustments have been made, the link will make a quite obtuse angle with the arm, so that the mechanical advantage is materially reduced, and greater pedal pressure is required to produce the same braking effect.

To maintain the mechanical advantage it is advisable to make the adjustment between the cam shaft and its arm. One plan is to use a serrated shaft and to clamp the arm to it. This permits of fine adjustments in the angular relation between arm and shaft. Alternately, as shown in Fig. 21, the lever can be made in two parts, which can be moved angularly with relation to each other and clamped together. The contacting surfaces are serrated, so that friction does not need to be depended on to maintain the adjustment. Another scheme, which has been extensively used, consists in cutting worm-wheel teeth on the brake shaft and mounting a worm



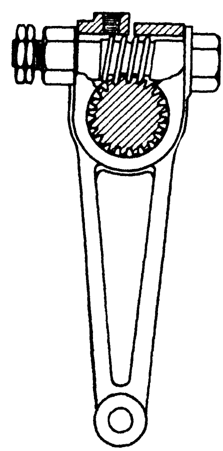


FIG. 21 (*left*).—TWO-PART LEVER FOR BRAKE ADJUSTMENT.
FIG. 22 (*right*).—WORM-AND-WHEEL BRAKE ADJUSTMENT.

in the hub of the brake arm (Fig. 22), so that by turning the worm with a wrench, the arm can be moved angularly with relation to the shaft. The worm may be replaced by a rack meshing with spur teeth on the shaft. Both ends of the shaft are threaded and provided with nuts, and by loosening one nut and tightening the other, the lever is moved angularly on the shaft.

Adjusting Means for Hydraulic Brakes—In the case of hydraulic brakes the means of adjustment must be incorporated in the brake itself. With the conventional two-shoe hydraulic brake, two adjustments are provided, one consisting

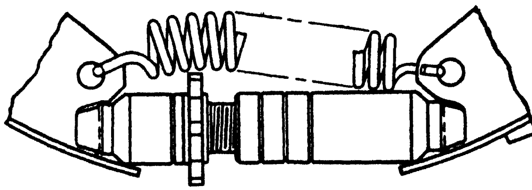


FIG. 23.—ADJUSTABLE STRUT OF BENDIX DUO-SERVO BRAKE.

of a pair of cams which contact the under side of the shoe near the toe when the brake is released, while the other is combined with the anchor pin, which has an eccentric mounting. When the adjusting cams are turned to compensate for wear, the shoes move pivotally around the anchor pins. If after such an adjustment the clearances at the toe and heel of the shoe should be materially different, a further adjustment is required, and this is made by means of the anchor pins, which have eccentric mountings in the backing plate. Loosening the nuts on them and turning the pins around in their mountings will serve to equalize the clearances at the heel and toe respectively. In making the adjustment, use is made of flat feeler gauges, of thicknesses corresponding to the clearance desired. These feeler gauges are introduced between shoe and drum through openings in the backing plate or drum, which are normally closed by snap covers.

In the servo-type of hydraulic brake there is a strut between the two shoes. As the lining wears, the diameter of the brake decreases, and this can be compensated for by lengthening the strut which, as shown in Fig. 23, has a screw adjustment.

In the balanced type of brake, where the toes of the two shoes are opposite instead of adjacent to each other, a separate adjustment for wear must be provided for each shoe. In earlier brakes this adjustment usually was made by means

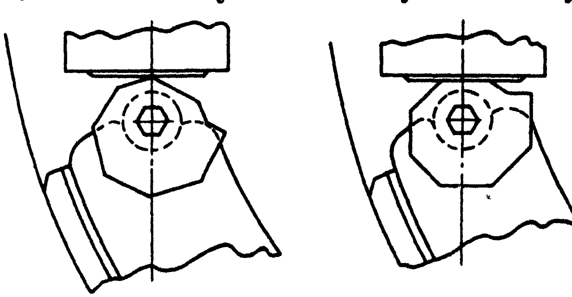


FIG. 24.—POLYGON ADJUSTER FOR HYDRAULIC BRAKES.

of a screw in the piston of the wheel cylinder, the head of which contacted the toe of the shoe. An improvement on this adjusting means is the polygon adjuster illustrated in Fig. 24. It consists of a steel spool comprising a central cylindrical section and two multi-faced end sections. The spool acts as a cam, because the faces of its end portions that come into contact with the shoe successively as adjustments are made, are at successively greater distances from the axis of the cylindrical section. The cylindrical surface contacts the end of the shoe web, which has a circular recess conforming to that surface, while a pair of the flat faces contact the piston. Adjustment is made by means of a wrench, which is inserted in the hexagonal socket of the adjuster. The edge between any two adjacent faces is farther from the axis than either of these faces, and if an adjustment is attempted and the edge will not pass the piston, it shows that not sufficient wear has taken place to make an adjustment necessary.

Automatic Adjustment—Quite a number of automatic brake adjusters have been devised, but only few have come into practical use thus far. An important advantage of the automatic adjuster—aside from the fact that it makes periodic trips to the service station for brake adjustment unnecessary—is that no provision for liner wear need be made in the pedal reserve. The pedal travel for brake application can then be made correspondingly larger, and the mechanical advantage of the pedal increased. As the lining on the brake shoes wears, the clearance between it and the drum in the released position increases, and greater pedal travel is then required to take up the greater clearance. To make the brake self-adjusting, means must be provided which will automatically keep the clearance constant regardless of liner wear.

Studebaker cars for a number of years have been provided with a self-adjusting device that was developed by the Wagner Electric Co. (Fig. 25). In this two-shoe hydraulic brake the shoes are withdrawn from the drum by the usual tension springs. The distance they can be withdrawn is determined by a mechanism comprising a clearance-adjusting eccentric mounted on the backing plate, an adjusting lever in contact with the eccentric and articulated to the inner end of a contact plug which extends through holes in the shoe and lining, and an adjusting wedge interposed between a wedge guide on the web of the brake shoe and the pin connecting the adjusting lever to the contact plug. The contact plug is held in its proper relation to the lining surface by a shear-type spring, while the wedge is under the influence of a tension spring.

Now suppose that the lining has worn slightly, so that during brake release the shoe and the contact plug receded farther from the drum than previously. This causes the pin joining the lever to the contact plug to move away from the wedge guide, and enables the wedge spring to force the wedge deeper into the space between the wedge guide and pin. In this way the adjusting lever is turned slightly around the pin, and that part of it in contact with the eccentric is moved inward to keep the distance from it to the surface of the lining constant.

A somewhat different principle is made use of in a Mercedes automatic adjuster, shown in Fig. 26, which is used on

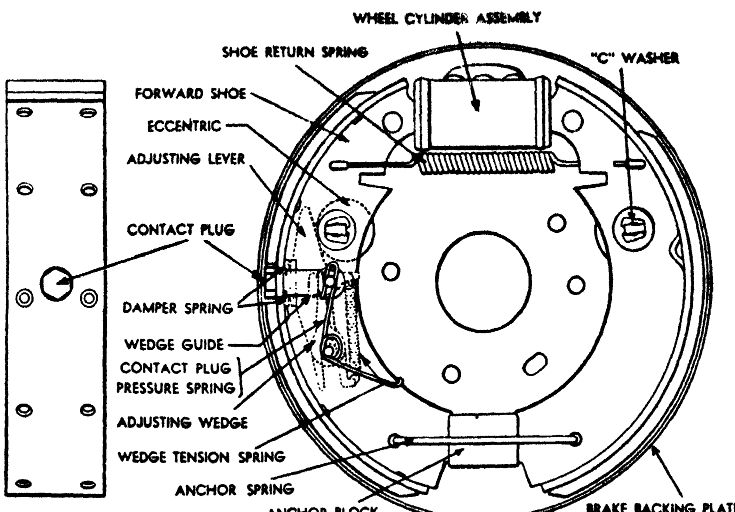


FIG. 25.—STUDEBAKER BRAKE SELF-ADJUSTING MECHANISM.

a large car equipped with two-shoe hydraulic brakes. Oblong holes in the webs of the brake shoes for the anchor pin allow the shoes to center themselves in the drum. Near the free end of each shoe there is another oblong hole in the web, through which passes a stud *A* surrounded by a flanged sleeve *B*. The latter is bored out to a diameter 0.040 in. larger than that of the stud. Threaded over the sleeve are two friction discs, which are held against the web of the shoe by a spring washer *C*, with sufficient force so that the friction between the discs and the brake shoe exceeds the force of retracting spring *D*.

Under normal conditions the sleeve will contact the stud on one side when the brake is applied, and on the opposite side when the brake is released. The releasing force (the force of the retracting spring) is insufficient to move the shoe relative to the sleeve, but the force of application is sufficient to produce such a motion, and any wear of the lining will be automatically compensated for by a sliding motion of the web of the shoe relative to the friction discs and the sleeve. The clearance between shoe and drum in the released position therefore always remains constant, and is determined by the difference in diameters of stud *A* and the bore of sleeve *B*.

Division of Braking Effort Between Front and Rear—When four-wheel brakes were first adopted for passenger cars it was customary to divide the braking effort between the front and rear brakes in the proportion of 40 to 60. At that time the front wheels usually carried considerably less

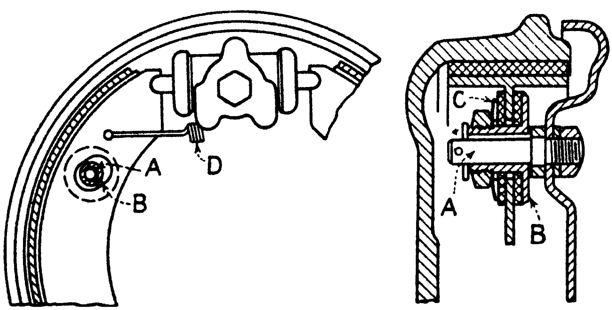


FIG. 26.—MERCEDES AUTOMATIC BRAKE ADJUSTER.

weight than the rear under static conditions, and, besides, it was believed that locking of the front wheels was more dangerous than locking of the rear wheels. Since then the weight distribution in passenger cars has been changed materially by moving the seats and the powerplant forward, so that when the brakes are applied hard there is considerably more weight on the front than on the rear wheels. It also has been pretty well established that under most driving conditions locking of the front wheels is less hazardous than locking of the rear wheels. It has therefore become the practice to divide the braking effort in passenger cars in the proportion of 60 front to 40 rear. This unequal division of the braking effort can be brought about by applying the front brakes with greater force (using wheel cylinders of larger bore), by using linings with a higher friction coefficient for the front brakes, or by providing for a higher degree of self-energization for the front brakes. If both sets of brakes are of the same size and the front brakes do a greater share of the work, they naturally will reach higher temperatures in service and wear faster.

In trucks and tractors, where the front wheels are much less heavily loaded than the rear ones, the front brakes must be made less powerful. The most rational division of the braking effort between the wheels would seem to be in proportion to the loads on the different wheels under full-load conditions and with the brakes fully applied.

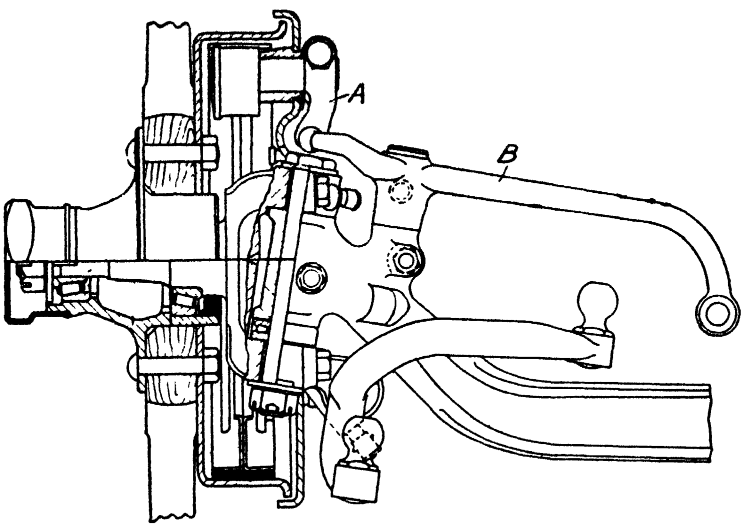


FIG. 27.—LEVER MECHANISM FOR FRONT-WHEEL BRAKES.

Front-Wheel Brake Actuating Mechanism—The problem of transmitting pedal motion to rear-wheel brakes is a relatively simple one, because the only motion of the wheels and their brakes relative to the frame is a nearly straight up-and-down motion due to spring action. With front-wheel brakes the additional motion of the wheels around the knuckle pins in steering must be taken account of, and connections to them must be so designed that neither steering motion nor spring action will affect the functioning of the brakes, and so that brake application will not interfere with steering.

One operating mechanism for front-wheel brakes (no longer used) is shown in Fig. 27. The shaft of the brake cam carried a short lever *A* just outside the backing plate, which lever arm was engaged by the ball-end of a double-armed lever *B* fulcrumed on the front axle in such a manner that the center of the ball was in the axis of the knuckle pin when the brake was applied. Wear of the lining, of course, caused the center of the ball in the "brake applied" position to move away from the knuckle-pin axis, and to make it possible to readily return it to that position, a suitable adjusting means was provided, by which lever *A* was moved angularly with relation to the camshaft.

Brake Pedals and Levers—In a motor vehicle with standard control, the brake pedal is operated by the driver with his right foot, and therefore is located a short distance to the right of the steering column. Brake and clutch pedals are mounted symmetrically on opposite sides of the column. The ordinary control pedal consists of a bent lever of which the part near the fulcrum is below the toe-board, and nearly parallels it, while the part near the pad forms an arc of a circle with the fulcrum as center. This part passes through the toe-board, and when it is arc-shaped, the hole in the toe-board can be relatively small, and therefore can be more easily sealed. Brake pedals vary in length from 13 in. to 16 in. (center-to-center measurement). The part below the toe-board usually is made of I-section, with a section modulus of from 0.08 to 0.12 in.^{3}, while the part passing through the toe-board is usually made of round or rectangular solid section, but can also be made of I-section, with a cross-sectional area of 0.25-0.40 sq in. A rubber grommet or its equivalent is passed over the arc-shaped section and acts as a silent stop for the pedal when released. To permit threading the grommet over the pedal, the shank either is made in two parts, or else the pad is made a separate part and bolted on. One form of brake pedal and brake lever were shown in Fig. 13. In modern passenger cars the brake levers are generally mounted on the cowl and extend downward and backward, so they are out of the way of the occupants of the front compartment. Unfortunately, the cowl does not afford a very rigid base, and in the past the linkage or cable connection to the rear-wheel brakes sometimes has been less substantial than would have been desirable, with the result that some of these brakes were not very dependable.

In motor buses it has become customary to so design and mount the brake lever that in the "brake-released" position it extends forward close to and substantially parallel with the floor, to the right of the driver's seat. This makes it possible to give it a rigid support on the chassis and to obtain a high mechanical advantage (long lever), and it prevents interference with access to the seat. The brake lever used on Yellow coaches is shown in Fig. 28, where the dashed lines at the bottom of the drawing represent the floor. The lever is bent at right angles a short distance from its fulcrum, which necessitates the use of two links and an intermediate rocking member between the latch and the latch handle.

Balanced Brakes—In the foregoing the discussion of two-shoe internal brakes has been confined to the type in which adjacent ends of the shoes are anchored either to a common anchor pin or to two anchor pins close together. In such brakes relatively little benefit is derived from the self-energizing effect, because for each direction of motion one of the shoes is self-energizing, while the other is what might be called "de-energizing"; that is to say, the effect of the friction between it and the drum is to decrease the force with which

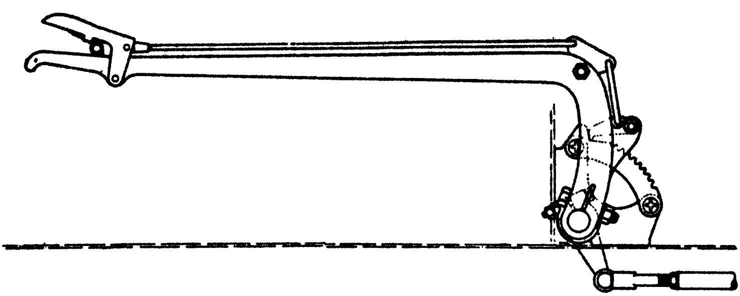


FIG. 28.—BRAKE LEVER PARALLEL WITH FLOOR (YELLOW COACH).

the shoe is being pressed against the drum, and therefore the friction. By a suitable arrangement of shoes, anchors and expanding means, it is possible not only to make both shoes self-energizing, but also to make the frictional force on one shoe help to apply the other shoe to the drum. The first type, in which both shoes are self-energizing, is known as the balanced brake, while the second, in which one shoe helps to apply the other to the drum, is known as the servo-type.

In the balanced type, as both shoes are self-energized, both do the same amount of work and both, therefore, wear equally, provided, of course, the design is symmetrical and both shoes carry similar linings. Another advantage is that both shoes press equally against the brake drum, hence the pressure is taken up completely within the drum and does not increase the load on the wheel bearings. The chief advantage, however, undoubtedly is that, under otherwise similar conditions, less pedal pressure is required with the balanced than with the conventional two-shoe brake to produce a certain braking effect under forward motion.

In one type of balanced brake (Fig. 29) there are two hydraulic cylinders located diametrically opposite each other, each containing a single piston. The shoes have separate anchor pins and each of them is acted upon by the piston in one of the cylinders. As both of the shoes are self-energized, such a brake produces about 50 per cent more "forward" braking torque than the conventional brake, but only

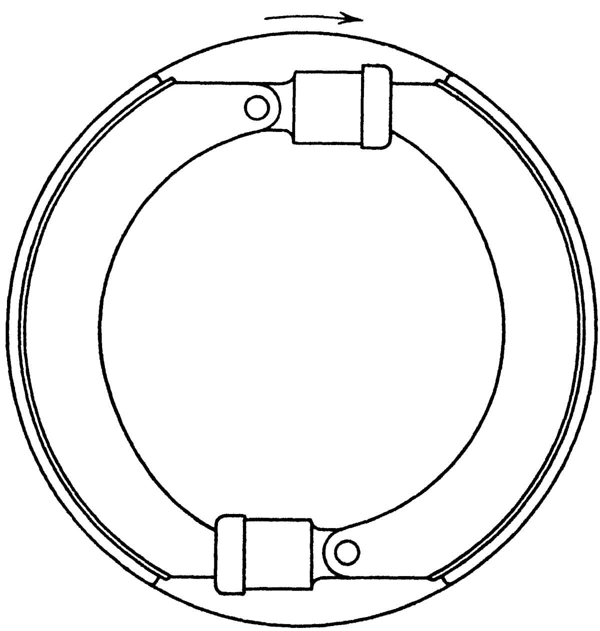


FIG. 29.—BALANCED BRAKE WHICH IS SELF-ENERGIZING DURING FORWARD MOTION ONLY.

about one-third as much "reverse" braking torque. This makes it unsuitable for use on all four wheels, but it has been suggested to combine such balanced brakes on the front wheels with conventional brakes on the rear wheels. Such a combination would approximately give the desired 60:40 division of braking power between front and rear with drums and hydraulic cylinders of the same size for both. The braking power in reverse would be only about two-thirds that in forward motion, but on the rather rare occasions when high reverse braking power is required it would always be available at the expense of a little extra pedal pressure.

In some special cases (large racing cars) use has been made of balanced brakes in which both shoes are trailing shoes, and therefore "de-energizing," as this completely eliminates the self-locking hazard. A much greater force of application is then needed, and power application is resorted to.

Bearing Loads due to Unbalanced Brakes—The loads which application of unbalanced brakes impose on axle and wheel bearings are likely to be underestimated. We found in the foregoing that theoretically, with the conventional two-shoe, single-cylinder brake and a friction coefficient of 0.35, the self-energized shoe should produce four times as much braking torque as the de-energized one. But neither the drums nor the shoes and their supports are perfectly rigid, as was assumed in the theoretical treatment, and elastic yield of the parts will reduce the ratio to about 3. In the example of a passenger-car brake discussed earlier in this chapter we found that the total normal pressure required to produce a deceleration of 10 fps^2 was 8857 lb. If the self-energized shoes do three times as much work as the de-energized ones, the normal pressures of the two sets of shoes will be 6643 lb and 2214 lb, respectively, and the difference of 4429 lb is made up of loads imposed on the bearings through which the wheels support the chassis. That is more than the static load on these bearings, which is somewhat less than 4000 lb. Of course, the loads due to chassis weight and brake unbalance, respectively, do not add directly, because the former are vertical in direction while the latter ordinarily are nearly horizontal.

Bendix "Twinplex" Brake—In the Bendix Twinplex hydraulic brake (Fig. 30) there are two oppositely located wheel cylinders, each containing two pistons. At each end of each cylinder there is an anchor cap. One of the caps of each cylinder is provided with a wear plate, the other with an adjusting screw, and the ends of the brake shoes are held in contact with the wear plate and the adjusting screw by the retracting springs. When fluid is forced into the cylinders, the shoes are forced apart and into contact with the drum, and the frictional force then carries the shoes around until one of the caps on each cylinder contacts the latter and causes it to act as brake anchor.

Timken Balanced Brake—Another brake of the balanced type is the Timken DP (Dual Primary) model, of which an assembly view is shown in Fig. 31. Both shoes of this brake are self-energizing under both forward and reverse motion, and this effect is obtained with a single hydraulic wheel cylinder. The wheel cylinder, shown at the top in the drawing, acts on shoes *E,E* through the intermediary of brake levers *B,B*, which are fulcrumed on the eccentrically mounted anchor pins *I,I*. The "floating" shoes have the actuating force applied to them at the middle through pressure blocks *F*. The

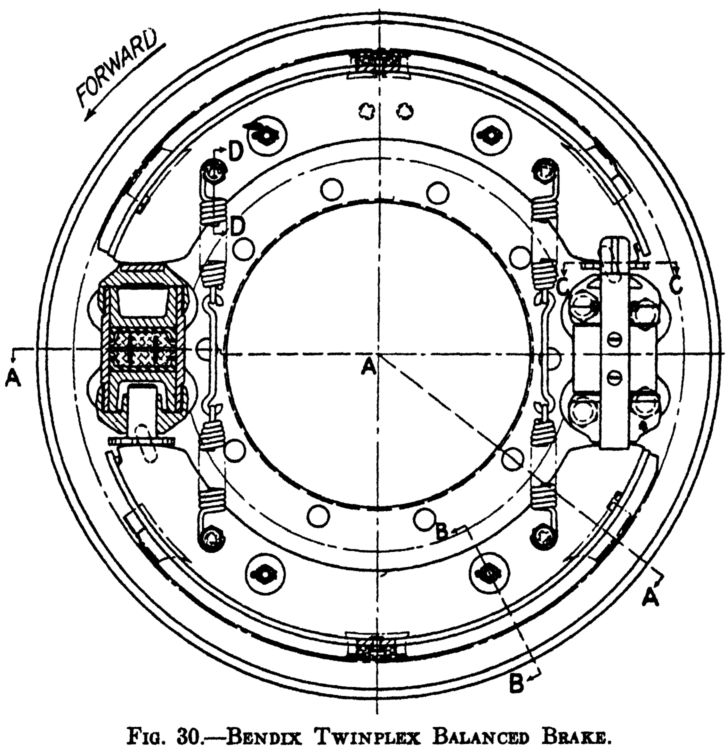


FIG. 30.—BENDIX TWINPLEX BALANCED BRAKE.

brake levers are double, one-half of each being located on each side of the brake-shoe web.

On brake application, when the shoes come in contact with the drum, friction at the contact surfaces carries them around in the direction of drum rotation until (with left-hand rotation) the left shoe contacts the lower abutment block *H* and the right shoe an upper abutment block not shown. This circumferential motion of the shoes is made possible by the fact that the pressure blocks *F* are movable laterally with respect to the brake levers. Under reverse motion, when the drum rotates in the opposite direction, the right shoe ends up against the lower abutment block *H* and the left shoe against the upper one. Retainer springs *G*,*G* hold the pressure blocks *F* in place on the shoes when the brake is disassembled.

Wagner Hi-Tork Brake—Fig. 32 shows the Wagner Hi-Tork brake, which acts as a balanced brake during forward, and as a conventional, non-energizing brake during reverse, motion. It has a single expanding unit, at the top, but this unit contains two communicating cylinders with pistons therein, one cylinder being of about twice the bore as the

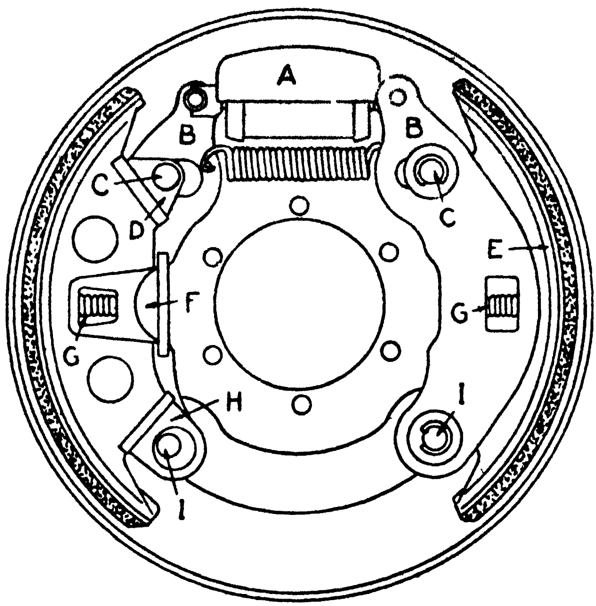


FIG. 31.—TIMKEN DP BALANCED BRAKE.

other. The piston in the large cylinder acts directly on the front shoe, while the piston in the small cylinder acts on the rear shoe through a lever with a 4:1 ratio, so that the force of application is the same for both shoes. The rear shoe pivots around an anchor pin, on which it acts during both forward and reverse motion, while the front shoe acts on an anchor pin at the bottom during forward motion, and on an anchor block at the top during reverse motion.

During forward motion (clockwise rotation of the drum in the drawing), that part of the drum in contact with either shoe moves toward the anchor for that shoe, so that the frictional force is in the same direction as the force of application, and both shoes, therefore, are self-energizing. During reverse motion the friction between the drum and the rear shoe is opposite in direction to the force of application, and this shoe, therefore, is partly de-energized. In the case of the front shoe, the point of anchorage has changed from the bottom to the top, and since the direction of drum rotation also has changed, this shoe is still self-energizing. Thus during reverse motion the brake acts like an ordinary, non-

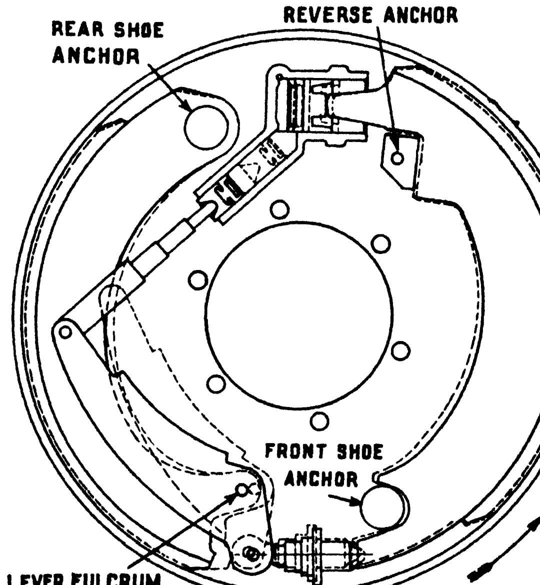


FIG. 32.—WAGNER HI-TORK BRAKE.

energizing brake, in which the two shoes are anchored to either the same pin or to two adjacent pins. In Fig. 32 the brake shoes and expanding mechanism are shown in full lines in the positions they occupy when the brake is applied while the car is running backward, and in dotted lines in the positions they occupy when the brake is released.

Servo Brakes—In a servo brake the fullest use is made of the friction between drum and shoes to help apply the shoes to the drum. This calls for a link between the two shoes at a point opposite the anchor, through which the frictional force of one shoe can be transmitted to the other. Fig. 33 shows the Bendix Duo-Servo brake, which is of this type.

This brake has only one wheel cylinder, which is located at the top. The two pistons within the cylinder act on the brake shoes through struts. Between the lower ends of the shoes there is an adjusting screw with caps, one threaded, the other smooth-bored, which engage the ends of the shoe webs. The upper ends of the shoes normally are held in contact with the anchor pin, located centrally above the wheel cylinder, by the retracting springs.

When fluid is forced into the cylinder, the shoes are pressed apart and into contact with the drum, and the friction then forces them around in the direction of wheel rotation until the

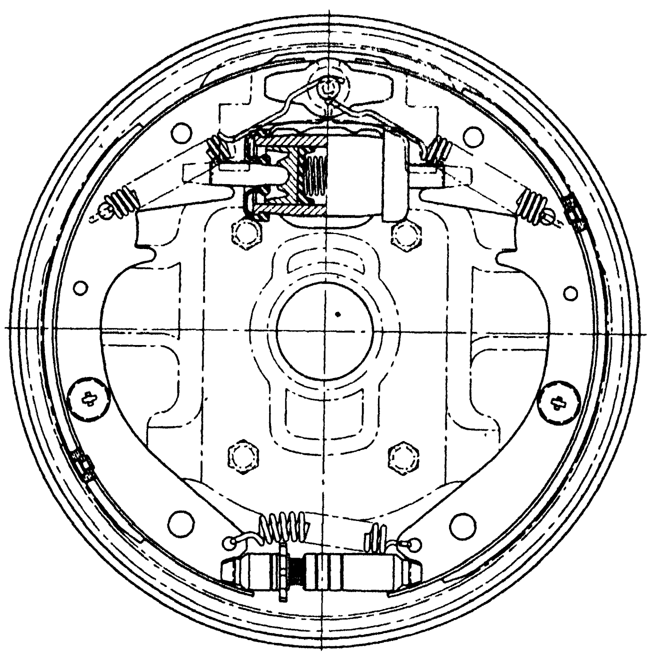


FIG. 33.—BENDIX DUO-SERVO BRAKE.

secondary shoe again contacts the anchor pin. The frictional force on the primary shoe is transmitted to the secondary shoe through the adjusting-screw assembly, and the friction on both shoes helps to press the secondary shoe against the drum, while the primary shoe is energized only by its own frictional force. Thus the secondary shoe is pressed against the drum with greater force than the primary, and consequently exerts a greater retarding effect. With this type of brake a given braking effect is obtained with minimum pedal pressure. For a given force of application, the braking effect with a Duo-Servo is greater than that with a balanced brake in about the same proportion that the latter is greater than the braking effect with a conventional, non-energized brake. Since the two shoes of a Duo-Servo brake press unequally against the drum, their rates of wear are different, and they also impose a load on the wheel or axle bearings when the brake is applied. There is servo action for both directions of motion.

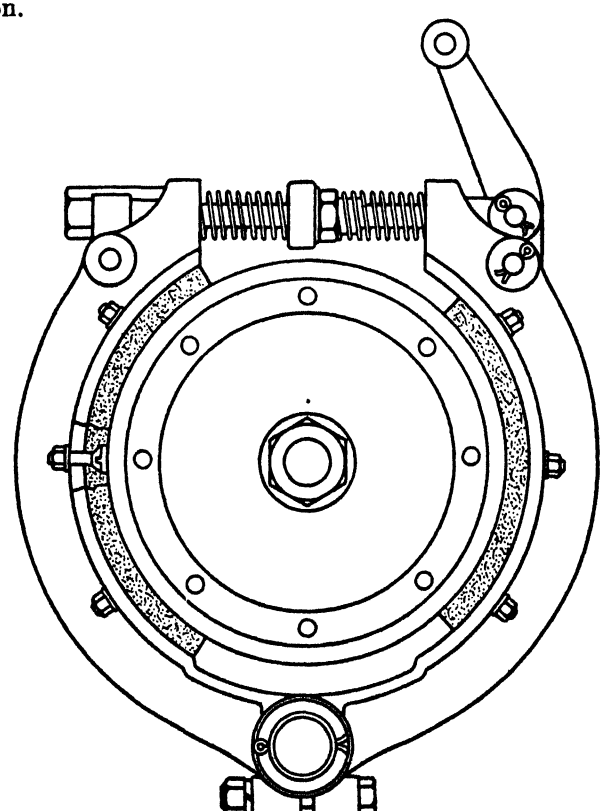


FIG. 34.—EXTERNAL-SHOE BRAKE (GMT).

Duo-Servo brakes are used very extensively on passenger cars. For maximum stability and also to reduce the difference in wear between the two shoes the primary shoe is generally provided with a lining having a high, and the secondary with a lining having a low, friction coefficient. Even then the pressure of the secondary shoe against the drum is about three times that of the primary.

External-Shoe Hand Brake—If the hand brake is used solely for parking purposes, there is, of course, very little wear on its linings, and relatively light, flexible linings may well be used, which go together with flexible steel bands. But where the hand brake is likely to be used a good deal in conjunction with the service brake to prevent the latter from reaching excessive temperatures on long down-grades, considerable wear must be expected, and in that case a shoe brake with heavy, rigid linings is preferable. The contracting type is preferable to the expanding type because it allows the designer greater latitude in the arrangement of the actuating mechanism.

Fig. 34 shows a contracting-type shoe brake used on Yellow coaches of General Motors Truck & Coach Division with angular drive, the brake drum being mounted on the "angular" propeller shaft. Lining blocks are secured to the shoes with flat-head screws and nuts. The lower end of each shoe is swiveled on an eccentric anchor pin. Clearance adjustments are made by turning the anchor pin by means of a tang at its rear end. The anchor pin is supported in a bracket with split boss provided with a clamping bolt. At the upper end of the shoes there are two adjusting nuts, to permit of adjusting the clearance of each shoe independently. It will be noticed that the lining on the left shoe covers a greater arc of the circumference. This tends to equalize the wear of the two shoes, the shoe on the left being the self-energizing one.

Timken Duo-Grip Brake—Another parking or auxiliary brake for commercial vehicles is the Duo-Grip of the Timken-Detroit Axle Co., which has one internal and one external shoe. Each shoe covers about one-fourth the drum surface, and both cover the same part of the drum, hence three-fourths of the drum surface is free to radiate heat. To apply the shoes to the drum, they must be moved toward each other, and this is accomplished by means of a floating lever carrying two parallel bars extending laterally through drilled lugs on the shoes. When a pull is exerted on the lever, the shoes are made to approach each other and to grip the drum. In one design the shoes are supported by two pairs of parallel bars. Since the actuating lever floats and the two shoes press against the drum in opposite directions, the bearing load due to brake application probably is smaller than with some other brakes.

Band Brakes—As already pointed out, band brakes, as a rule, are used only as parking brakes, and then are mounted either on the transmission or on the propeller shaft. A band brake comprises a steel band lined with friction material, which extends almost completely around the drum. The lining may be either in a single piece or in two or more sections. If it is in sections, a portion of the band near the anchor pin usually is unlined.

A typical band-type parking brake, used on some Mack trucks, is shown in Fig. 35. In band-type transmission brakes the anchor and the contracting mechanism usually are located on opposite sides of the drum, at shaft height, as this facilitates making connections to the brake lever. A U-shaped lug riveted to the middle of the band spans a square-section anchor pin, on which it has a limited radial motion. When the brake is released, a coil spring between the lug and the anchor pin withdraws that portion of the band near the anchor from the drum, but this retracting motion is limited by the head of adjusting screw *1*. Brackets are riveted to the band also near its free ends, and a bolt passes through holes in these

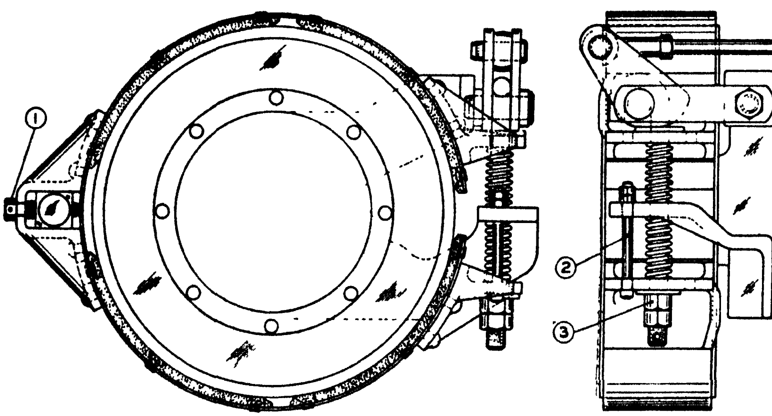


FIG. 35.—BAND-TYPE PARKING BRAKE (MACK).

and in a supporting bracket secured to the rear of the transmission housing. At its upper end the bolt has an eye formed on it, and a pin passing through this eye also passes through holes in a pair of links connecting to another arm of the supporting bracket, and in a pair of cams with lever arms from which a link connects to the brake lever. Retracting springs inserted between the brackets on the ends of the band and the supporting bracket force the ends of the band apart when the brake is released. Owing to its weight, there is a tendency for the upper half to "ride" the drum when the brake is released, and to prevent this, there is provided an adjusting bolt *2* between the supporting bracket and the lower half of the band, which limits the release motion of the lower half and ensures that the upper half also will have adequate clearance. Clearance adjustments are made by means of adjusting screw *1* and adjusting nut *3*. Band brakes on the transmission usually are adjusted for from 0.015 to 0.020 in. clearance.

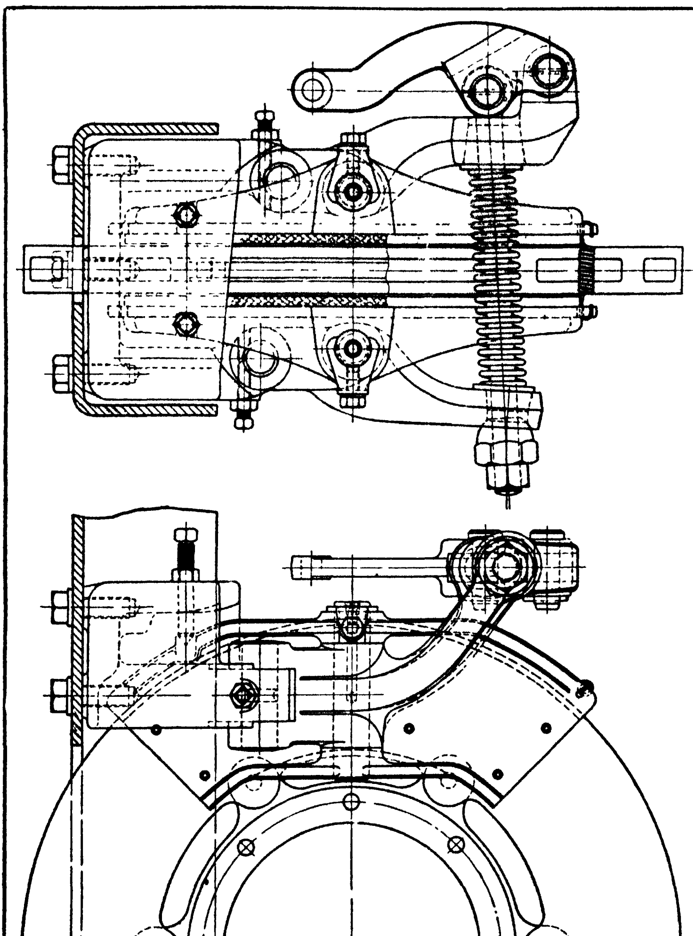


FIG. 36.—TRU-STOP DISC BRAKE.

Disc Brakes—Disc brakes are used principally as transmission brakes on heavy trucks, though they have been used also as wheel brakes, notably on some British four-wheel-drive military vehicles. In the latter application their chief virtue seems to be that they require less space in the axial direction than drum brakes. As a transmission brake, an advantage of the disc- over the drum-type is that the disc can be cooled more effectively than the drum, by "ventilating" it and causing it to act as the impeller of a centrifugal blower. Another advantage sometimes claimed for the disc-type transmission brake is that the overhang over the adjacent bearing is less, hence the bearing load due to brake application is less.

A transmission brake known as the Tru-Stop, and manufactured by the American Chain & Cable Company, is illustrated in Fig. 36. Originally the disc was cast, but to prevent warpage and runout, it was later made of composite construction. Two stamped plates of 1035 steel are riveted to a malleable-iron spider. The plates are riveted to the spider by an inner row of rivets, while an outer row merely holds them together, the ends of the spider arms and short tubes surrounding the rivets serving as spacers. Sector-shaped, ribbed shoes, with friction linings applied, are located on opposite sides of the disc; they are swiveled on levers fulcrumed on a bracket secured to a frame cross member. The ends of these levers can be drawn together by means of another lever and a drawbolt. Adjustments are made by means of the nuts on the drawbolt and the two set screws in the bracket, these latter serving to equalize the clearances at top and bottom of the shoes.

Chrysler Disc Brakes—Chrysler Corporation developed a disc brake for use on its Crown Imperial models introduced in 1950. These brakes are self-energizing and self-adjusting. They provide greater lining area and better cooling facilities than drum brakes which could be accommodated in the same space, and they also are said to be more rigid. A sectional view of the front brake is shown in Fig. 37.

The brake consists of a two-part, ribbed, flat-cylindrical cast-iron housing carried on the wheel, within which there are two flat, cast-aluminum pressure plates, each of which is provided with six sectors of bonded brake lining. Six steel balls are located in sockets formed on the inner sides of the pressure plates, and two hydraulic cylinders are secured to the inner plates. When brake fluid is forced into the cylinders, the piston within the latter moves one of the pressure plates angularly with relation to the other and forces the steel balls to ascend a 32.5-deg ramp. This forces the pressure plates apart and brings the linings in contact with the side walls of the housing. Friction between the housing and the lining tends to cause the balls to ascend the ramps farther, thereby producing a self-energizing effect proportional to the steepness of the ramps. When the brake is released, four coil retracting springs draw the pressure plates together and out of contact with the housing. Fig. 38 shows the brake in both the released and the applied position.

The self-adjusting feature is illustrated in Fig. 39. There are two identical, oppositely located self-adjusting mechanisms on the inner side of the inner pressure plate. Two lugs, A and B, are located opposite each of these mechanisms on the inside surface of the outer pressure plate. Each self-adjusting unit is composed of a pair of guide flanges, a pin, and a friction locking unit between the flanges. The locking unit consists of an adjuster sleeve with an inside taper, eleven small

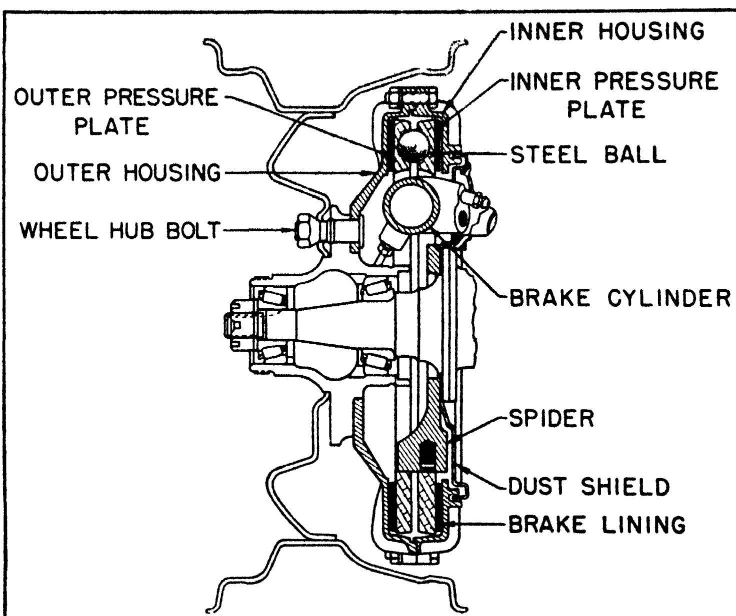


Fig. 37.—Section of Chrysler Disc Brake Mounted on Wheel.

steel balls, a copper washer, a spring, and a release bushing. The spring-loaded balls, in conjunction with the inside taper of the adjuster sleeve, provide a friction lock on the pin against motion toward the left (in the drawing).

When the brake is being applied, the outer plate is rotated with respect to the inner plate by the hydraulic cylinder, and lug A contacts the pin. Normally the brake lining contacts the housing at the same moment, and the pin is not moved relative to its support. However, if the linings have worn,

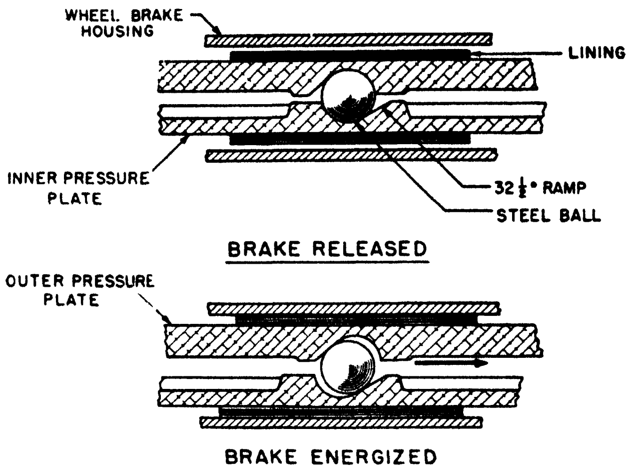


FIG. 38.—DISC-BRAKE ACTUATING MECHANISM IN RELEASED AND APPLIED POSITIONS.

lug A pushes the pin through the guide flanges until the linings contact the housing. Then, when the brake is released, the retractor springs tend to return the outer plate to its former position relative to the inner plate, but the locking mechanism on the pin prevents them from doing so. The difference between the length of the pin and the distance between the finished surfaces on the two lugs determines the brake clearance in the released position. This is a fixed length, and the clearance therefore remains constant regardless of lining wear.

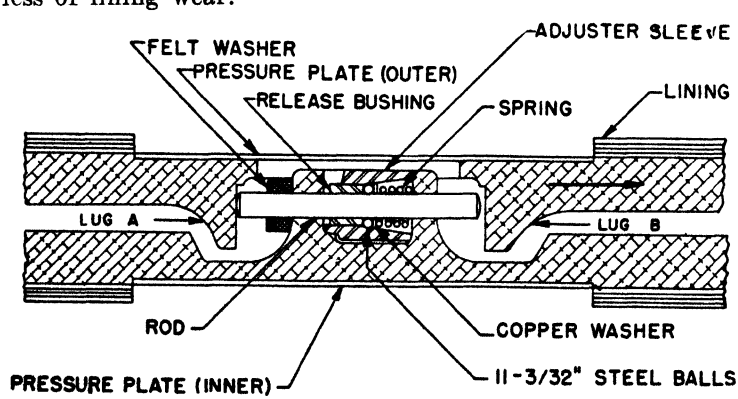


FIG. 39.—DISC-BRAKE SELF-ADJUSTING MECHANISM.

Four-armed malleable-iron spiders (Fig. 40), bolted to the steering knuckles and to flanges on the rear-axle housings, serve both as locators and as anchors for the pressure plates. They prevent the plates from moving radially, and they take the torque produced when the brakes are applied. Spring-backed plungers near the ends of the spider arms or anchors bear against the pressure plates and prevent rattling.

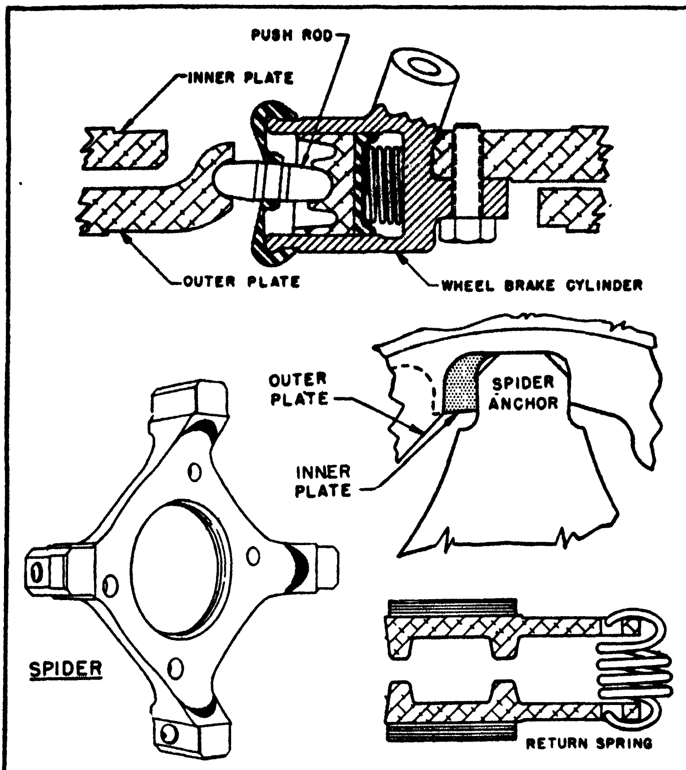


FIG. 40.—DETAILS OF CHRYSLER DISC BRAKE.

In the rear-wheel brakes both of the pressure plates float, and these brakes are equally effective for both directions of motion. During forward motion the inner plate is held against angular motion by the anchor, and when the linings contact the housing, the outer plate moves forward and causes the balls to ascend the ramps, thereby producing the self-energizing effect. In reverse motion the outer plate abuts against the anchor, and a self-energizing effect is produced by movement of the inner plate when it contacts the housing. In front-wheel brakes the inner plate is fixed during both forward and reverse motion, and these brakes therefore are not self-energizing in reverse. Front brakes are provided with $1\frac{1}{4}$-in and rear brakes with 1-in. cylinders, to produce the desired division of braking effort.

Suction-Cooling of Brakes—In the 1951 model Chrysler Crown Imperial the capacity of the disc brakes was further increased by cooling them by a forced draft. As shown in Fig. 41, twenty impeller blades are stamped in a one-piece retainer of cold-rolled steel 0.030 in. thick. The wheel cover is crimped to the periphery of the retainer, and the retainer in turn is held to the wheel disc by the usual spring clips. Air is drawn by the blower thus formed through twelve

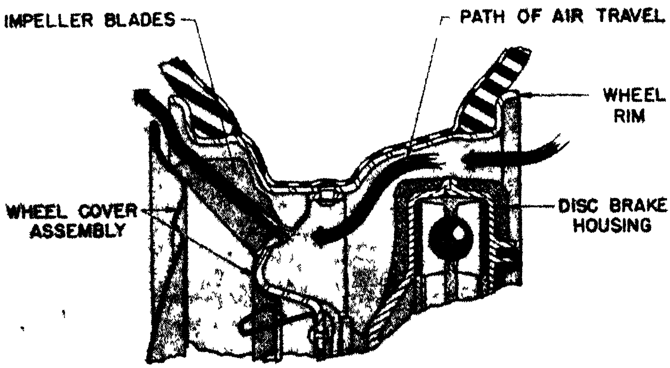


FIG 41—FORCED-DRAFT COOLING SYSTEM FOR DISC BRAKE

D-shaped holes stamped in the wheel disc, and expelled through the gap between the edge of the cover and the wheel rim. This cooling system is said to reduce the internal temperature of the brake by up to 35 per cent, and to reduce lining wear by as much as 50 per cent.

Brake Tests—Brakes can be tested directly on the vehicle, if suitable equipment is available for the measurement of the various factors involved. Such equipment includes a decelerometer for the measurement of the rate of deceleration produced, a pressure gauge indicating the pedal pressure, thermo-couples installed at various points on the brakes, and temperature-indicating or -recording apparatus. Instead of measuring the deceleration by means of a decelerometer, it is sometimes preferred to measure the distance in which the car is brought to a stop from a given initial speed, and then to calculate the mean deceleration from the speed and the distance. In making stopping-distance tests, it is now customary to indicate the point on the highway where the brake was applied by discharging a pistol loaded with powdered chalk and secured to the side of the vehicle in a vertical position. The piston is fired automatically by the motion of the brake pedal.

In developing brakes for new models, it is customary to make use of laboratory equipment with which the conditions of brake operation in service can be simulated. Instead of absorbing the kinetic energy of a heavy mass moving over the highway, the brake is made to absorb the kinetic energy of an equivalent rotating mass. It must be possible to vary the mass through a wide range, to represent vehicles of different weights, and the speed from which the flywheel is stopped also must be variable. A number of such brake-testing equipments have been built by manufacturers of brake linings and brake drums. They usually comprise a flywheel which when rotated at a given speed possesses a store of kinetic energy equal to one-fourth of that of a light car with its wheels rotating at the same speed. The rotating mass can be increased by securing different numbers of steel discs to the flywheel rim. The flywheel is brought up to speed by an electric motor. On the opposite side of the flywheel from the electric motor the shaft is provided with a flange to which a brake drum can be secured, and next to the brake drum, on a pedestal, is mounted a brake backing plate carrying the brake shoes and expander mechanism. When the brake is applied, the drum tends to drag it along, and the torque impressed on it is measured by means of a torque arm, the same as with an engine dynamometer. Both the force of application and the brake torque usually are measured hydraulically.

These brake-testing machines are so equipped that they can be put through a given cycle continuously, the flywheel being stopped at given intervals, which can be varied as desired. As the test proceeds, the brake heats up, and with an increase in temperature the friction coefficient of the lining tends to drop. It is possible to set the machine either so that the force of application remains constant and the brake torque varies with the friction coefficient, or so that the brake torque remains constant, in which case the force of application increases as the friction coefficient decreases.

Machines of this type can be used to study brake drums of different materials for performance and characteristics when stopping under different loads; to study the performance and characteristics of different linings, to determine brake drum and lining wear, and to study temperature gradients and heat flow in brakes.

In order that a rotating flywheel may have one-fourth the kinetic energy of translation of a car weighing W lb and having an effective wheel radius of R ft, when rotating at the same speed as the car wheels, its moment of inertia must be

$$I = \frac{WR^2}{4g} \text{ ft-lb-sec}^2,$$

where g is the constant of gravity (32.16). However, in bringing the car to a stop, the work done by the brakes is not exactly equal to the kinetic energy of translation of the car. This is so partly because certain parts of the car, such as the road wheels, have a rotary motion in addition to their linear motion, and partly because the rolling resistance and air resistance help to bring the car to a stop. The energy required to overcome the resistances to vehicular motion always considerably exceeds the kinetic energy of rotation of the wheels, etc., and this is sometimes accounted for by reducing the moment of inertia of the flywheel as given by the above equation by 5 per cent.

CHAPTER XV

Power Brakes and Auxiliary Energy Dissipators

With the adoption of four-wheel brakes it became difficult even on the larger passenger cars to apply the brakes to the locking point by direct pedal pressure, and with increase in the gross vehicle weight of trucks and buses following the development of the "giant" pneumatic tire, power braking became an absolute necessity on such vehicles.

Mechanical Relays—Power for actuating the brakes can be derived from various sources. As long as the vehicle is in motion, its store of kinetic energy is available, and a number of mechanical relays (boosters or servo-motors) were developed abroad to make use of this energy for brake application. Such mechanical relays at one time were used on passenger cars by Rolls-Royce, Renault, and others. If power actuation were required only to control forward motion, the relay could be of comparatively simple design. However, to be really serviceable, the device must apply the brakes whenever the pedal is depressed, regardless of the direction of motion, and in the event of failure of the relay, it must be possible to apply the brakes directly by pedal pressure. Any relay meeting these requirements is likely to be rather complicated, in addition to being bulky, and relays of this type never came into extensive use.

Air Brakes—Air brakes had made high-speed railroading possible, and had been highly developed for that service before the need for power braking on road vehicles arose. The pioneer firm in the industry, the Westinghouse Air Brake Company, also developed air-brake equipment for road vehicles. The automotive air-brake division was later separated and taken over by the Bendix-Westinghouse Automotive Air Brake Company, now located at Elyria, O.

Bendix-Westinghouse offers a variety of air-brake equipments. The simplest system comprises seven different parts, namely, an air compressor, a governor, a pressure gauge, a safety valve, a reservoir, a brake valve, and a series of brake chambers, all of which are interconnected by lines of tubing.

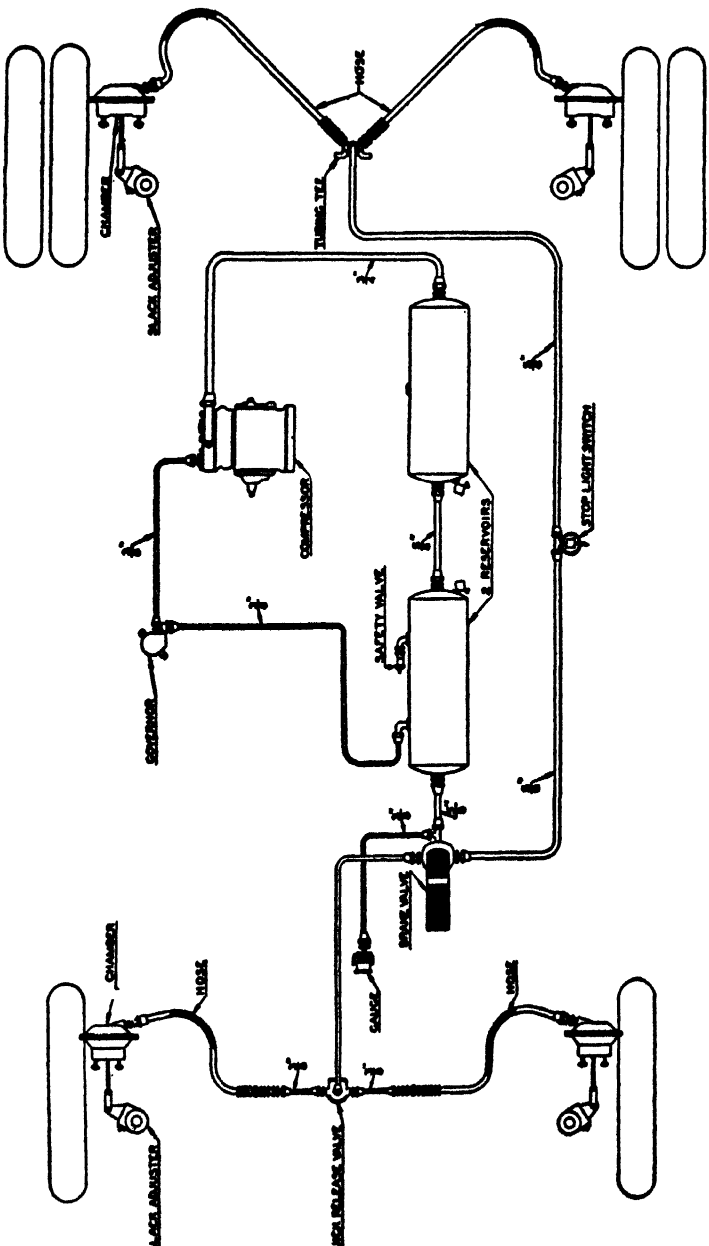


Fig. 1.—Diagram of Air-Brake Installation on Truck or Bus.

Other systems comprise additional members, such as a low-pressure indicator which indicates to the driver when the pressure in the reservoir has dropped below a certain value (usually 60 psi), and when, in consequence, it is unsafe to start out with the vehicle; an air-supply valve by means of which compressed air from the system can be made available for tire inflation or other purposes; a stop-light switch operated by the air pressure in the brake lines, which causes the stop light to light up whenever the brakes are applied; a quick-release valve through which the air in the front brake chambers can be quickly exhausted when the brake pedal is released, and a relay valve to speed up the admission of air to and release of air from the rear brake chambers, to enable the brakes to take hold more rapidly. In some cases a limiting valve is provided, to automatically limit the maximum pressure that can be admitted to the front brake chambers.

Air brakes are well suited also to tractor-trailer combinations, and in such installations an additional unit, a relay-emergency valve, is installed on the trailer. This device, in addition to speeding up the application and release of the trailer brakes, applies the brakes automatically in case the trailer becomes detached from the tractor.

Truck or Bus Installation—An installation diagram of an air-brake system for a truck or bus is shown in Fig. 1. The compressor, governor, pressure gauge, reservoirs, and safety valve constitute the compressing and control units, the remainder are application units. When compressed air is available on a vehicle, it is often used also for other purposes beside brake actuation, such as blowing the horn, actuating windshield wipers, operating bus doors, etc., and this may call for additional units, such as a pressure-reducing valve, an auxiliary reservoir, and valves for controlling the auxiliaries.

The two reservoirs of the installation shown in Fig. 1, which communicate with each other, are constantly kept under pressure by the compressor. From the forward resevoir a tube leads to the brake valve, and a branch from this tube connects to the pressure gauge on the dash. From the brake valve lines of tubing extend to the brake chambers. Single lines of tubing run fore and aft to points directly over the axles, to T-fittings from which connections are made to the brake chambers on opposite sides. The T at the forward end comprises a quick-release valve.

Trailer Installation—An air-brake installation for a semi-trailer is shown in Fig. 2. A reservoir installed on the trailer is combined with a relay-emergency valve, from which latter there are lines of tubing to the two brake chambers on the trailer. Two lines of tubing connect the relay-emergency valve to the brake installation on the tractor. One of these, known as the service line, connects to the brake valve and is under pressure only when the brake pedal is depressed. The other, the emergency line, connects to the tractor reservoir, and is under pressure at all times while the trailer is coupled to the tractor. Connections between the tractor and trailer lines are made by lengths of flexible hose and hose couplings at the forward end of the trailer.

Time Lag—In air-brake systems there is necessarily a slight lag between motion of the brake pedal or treadle and

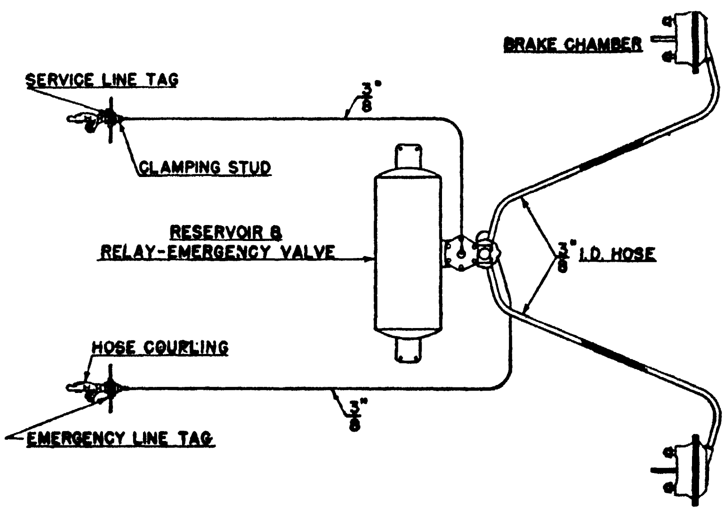


FIG. 2.—AIR-BRAKE INSTALLATION ON SEMI-TRAILER.

full application of the brakes, as it takes time for the air to pass from the brake valve to the brake chambers. Only very little time is required for the air pressure to make itself felt in the brake chambers, but in an emergency application fractions of a second count. Brake action begins as soon as the clearance between brake shoes and drum is taken up, but additional time elapses before the pressure in the brake chambers reaches the full value for which the brake pedal is set. In recent years designers of air-brake systems have devoted considerable effort to reducing the time lag, and with some of the newer installations a brake-chamber pressure of 60 psi, which represents an emergency application, can be reached in 0.4 second. Release of brake-chamber pressure is correspondingly fast. Larger tubing is now being used, and such units as brake valves and relay valves have been redesigned to afford greater air-flow capacity.

Air Compressors—Compressors for air-brake systems usually are of the single-acting type, with either two or three cylinders. A cross section of a two-cylinder Bendix-Westinghouse compressor is shown in Fig. 3. The compressor illustrated has an air-cooled cylinder head, while other models are provided with a water-cooled head, through the jacket of which water from the engine cooling system is circulated.

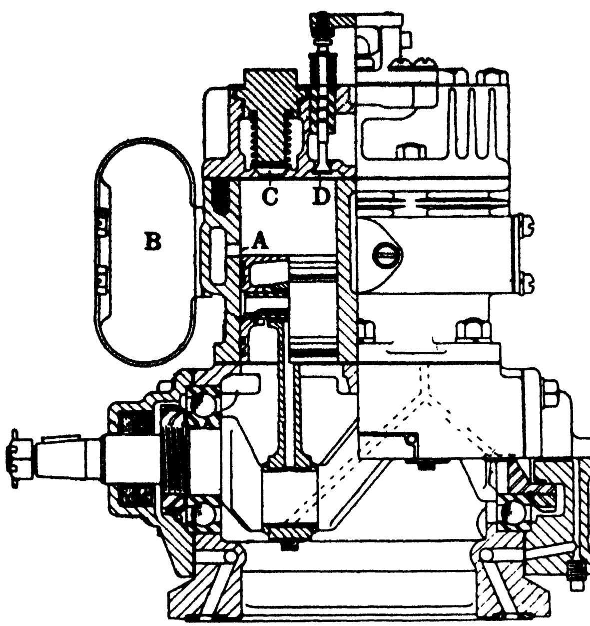


Fig. 3.—Two-Cylinder Air Compressor.

Air is taken in through port *A* in the cylinder wall, which is uncovered by the piston during the latter part of its "in" stroke. The intake is provided with an air cleaner *B*. Air is delivered through spring-loaded discharge valves *C* in the cylinder head. The crankshaft is carried in two ball bearings. Provision is usually made for the lubrication of all moving parts from the engine lubricating system, though compressors are being built also with their own lubricating system, with a plunger-type oil pump. Compressors usually are mounted with the base resting on a bracket on the engine crankcase, and the crankshaft in line with the engine accessories drive shaft. However, they are being built also for flange mounting to the rear of the engine accessories drive-gear housing, or to the engine gear-case cover, if the drive is to be direct from the crankshaft.

In a later design of compressor the cylinder head, cylinder block, and crankcase are cast in aluminum, the cylinders being provided with cast-iron liners. By using aluminum for all structural parts it was possible to hold the weight of a two-cylinder compressor of 12 cu ft rating at 1250 rpm to 30 lb, while an earlier three-cylinder compressor of the same displacement weighed 64 lb. Aluminum is used in air-brake installations wherever practical.

Air compressors are rated by the cu ft displacement of their pistons per minute at their normal speed of rotation, and compressors from 7¼ to 12 cu ft are being manufactured for automotive air-brake installations. The proper size of compressor for any given installation depends not only on the gross vehicle weight and the type of service for which the vehicle is intended, but also on the auxiliary air-operated equipment of the vehicle.

Unloading Mechanism—The compressor is driven from the engine, and therefore runs continuously, but when the pressure in the reservoir reaches a predetermined value (usually 105 psi), unloader valves *D* in the cylinder heads are opened by the governor, so that the air moved by the piston which is on its up-stroke can pass over into the other cylinder, in which the piston at the same time is on its down-stroke, without being compressed. This, of course, prevents unnecessary loss of power and unnecessary wear of the compressor. A sectional view of the governor is shown in Fig. 4. Enclosed within the housing is a Bourdon tube *A* (a device familiar to engineers through its use in steam gauges), which is open at one end and closed at the other. The open end, which is fixed in block *B*, is in constant communication with the air reservoir, while the closed end, which is free, contacts the upper valve *C* through a bracket. Normally the Bourdon tube is under tension and holds valve *C* off its seat, and the unloader chamber in the air compressor is then open to the atmosphere through valve *C* and exhaust port *E*. The lower valve *D* in block *B*, which controls the admission of air from the reservoir to the unloader chamber, is normally closed. As the air pressure in the reservoir and in the Bourdon tube increases, the latter tends to uncoil, and its pressure on valve *C* is reduced. When the pressure reaches the predetermined governor cut-out value (100-105 psi), valve *D* opens and valve *C* closes, owing to the combined effects of the spring pressure and air pressure on valve *D*, which then overcome the pressure of the Bourdon tube. As a result, reservoir pressure is admitted to the unloading chamber, and the compressor is unloaded. When the reservoir pressure drops to 80-85 psi, the governor cuts in again, the elastic force of the Bourdon tube then overcoming the spring pressure and air pressure on valve *D*. Air from the unloader chamber is exhausted at *E* and the compressor begins to function normally again.

A sectional view of the unloader mechanism is shown in Fig. 5. A diaphragm chamber is formed in the cylinder head of the compressor. When air pressure is admitted to the space

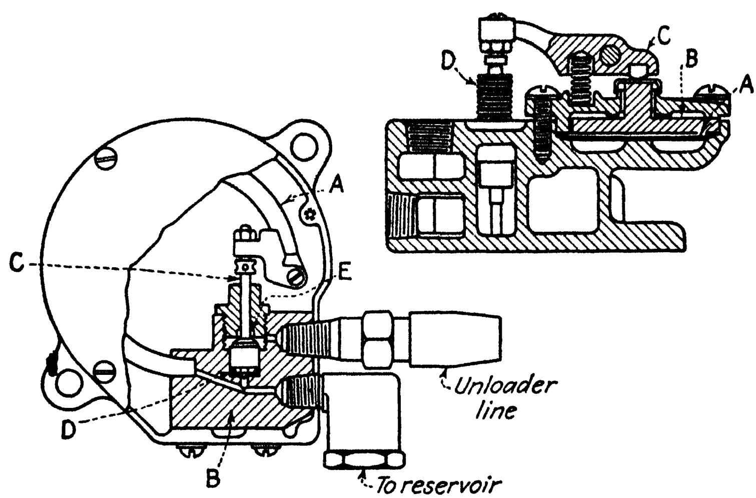


FIG. 4 (*lower left*).—GOVERNOR.
FIG. 5 (*upper right*).—UNLOADER MECHANISM.

below diaphragm *A*, the latter is raised, and through the intermediary of follower *B* and rocker *C* it lifts the unloader valves *D* off their seats. One of these unloader valves is better shown in Fig. 3. The compressor then continues to run without pressure in its cylinders.

The safety valve used with this system is a spring-loaded ball valve which screws into a fitting on the reservoir and is set to blow off at 150 psi.

Brake Valve—A very important part of the air-brake system is the brake valve, which must make it possible to apply any desired pressure to the brake chambers, up to the limit of the reservoir pressure. A sectional view of one type of brake valve used with Bendix-Westinghouse air brakes is shown in Fig. 6. A treadle fulcrumed on the housing of the brake valve presses on a plunger through the intermediary of a roller. The plunger in turn presses through a coil spring on a diaphragm. Pressure in the opposite direction is exerted

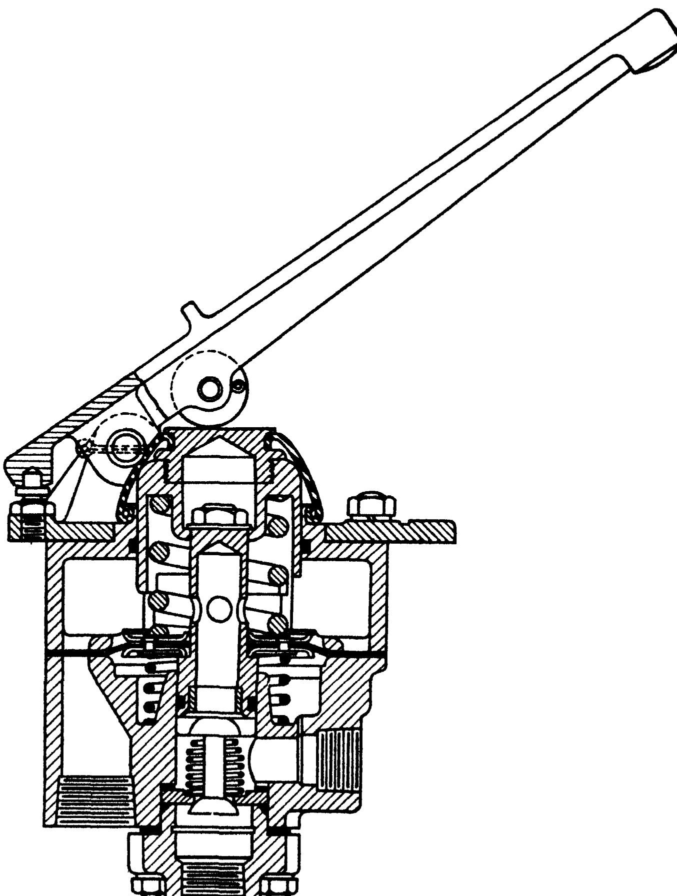


FIG. 6.—TREADLE-OPERATED BRAKE VALVE.

on the diaphragm by another, lighter coil spring below it, and by any air pressure there may be in the space below it, which communicates with the lines to the brake chambers. The line from the air reservoir connects at the bottom of the valve.

Now suppose that the driver forces the treadle down to a definite position. The first effect is to close the exhaust valve and open the valve directly over the inlet from the reservoir, thereby allowing compressed air from the reservoir to enter the lines to the brake chambers. As the pressure in these lines increases, that in the space below the diaphragm also increases, and this increase in the air pressure on it forces the diaphragm up again and allows the inlet valve to close. At what line pressure the valve closes depends on the position of the

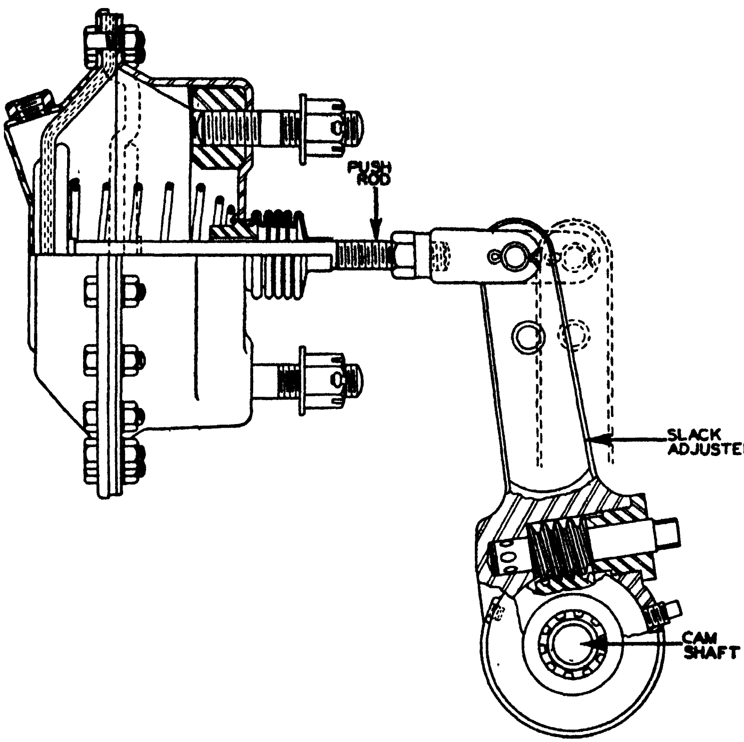


FIG. 7.—BRAKE CHAMBER AND "SLACK-ADJUSTER."

In later design the worm of the "slack-adjuster" is provided with a self-locking device which prevents it from backing off.

treadle. The force of brake application is always in proportion to the pressure exerted on the treadle, and the driver is thus able to gauge and closely regulate the pressure with which the brakes are applied.

Brake Chamber—A brake chamber (Fig. 7) consists of two dished steel pressings flange-bolted together, with an oil-proof rubberized fabric diaphragm between. At the center of the diaphragm is located a pushrod with a wide base or flange. Normally the diaphragm is held against one of the pressings by a spring surrounding the pushrod. The pushrod passes through a central opening in the other pressing, and connects through a yoke to an arm on the brake camshaft, this arm being known as a slack adjuster. Adjustments can be made in the length of the pushrod and in the angular position of the arm on the camshaft.

Brake chambers are made in diameters ranging from 6 to 11 in. and with maximum strokes of 1¾ to 3 in. When the brake is newly adjusted the stroke during the brake application should be as short as possible without the brakes dragging. The pushrod passes through the head of the chamber with liberal clearance, so the arc-shaped motion of its clevis end will not cause any binding. Brake chambers are mounted on

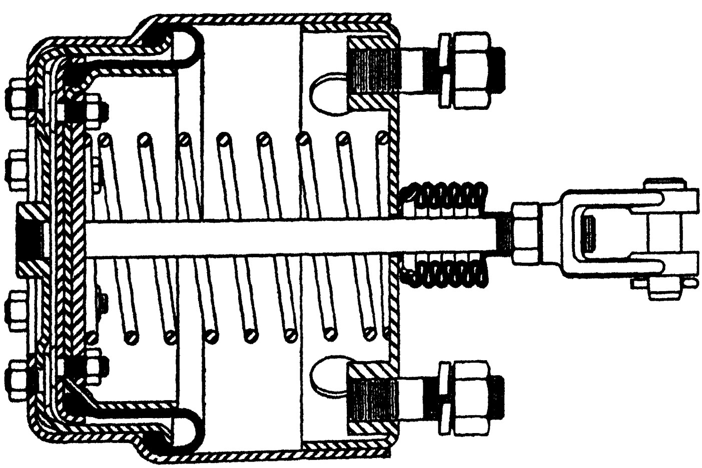


FIG. 8.—ROTOCHAMBER TYPE OF BRAKE CHAMBER.

either the brake backing plate or the axle, in direct line with the operating lever.

Instead of the type of brake chamber illustrated in Fig. 7, Rotochambers or cylinders are now much used in air-brake installations. The Rotochamber (Fig. 8), a Bendix-Westinghouse development, is essentially a cylinder containing a piston or pressure plate of smaller diameter than the cylinder bore, the space between piston and cylinder being sealed by a rubberized diaphragm of the form shown. The Rotochamber has two advantages over the conventional brake chamber, one being that for a given thrust on the pushrod its outside diameter is less, and the other that the thrust is substantially uniform throughout the stroke.

In connection with the slack adjuster shown in Fig. 7 it may be mentioned that it has been found advisable to provide it with a lock, so it cannot back off in service. Several different types of lock have come into use. That on the Bendix-Westinghouse is so designed that it is released automatically when the wrench used to make adjustments is pushed over the hexagon at the end of the worm shaft.

Relay and Relay-Emergency Valves—Air-brake installations on vehicles in which the brake chambers are at a considerable distance from the brake valve sometimes include a relay valve, which is intended to speed up the admission and release of air to and from the brake chambers and make the

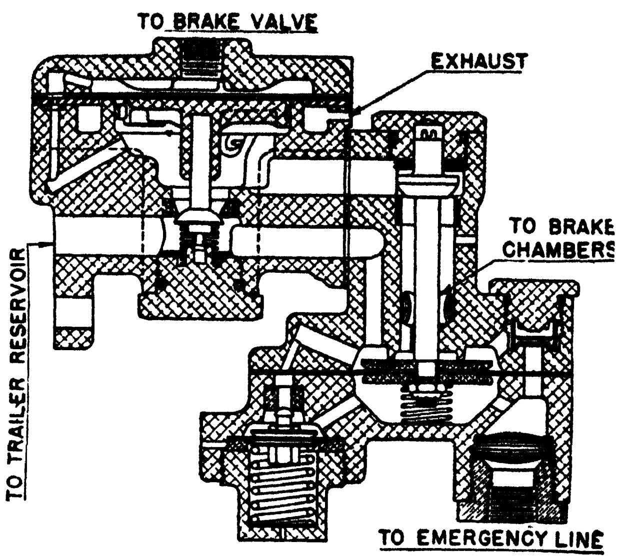


FIG. 9.—RELAY-EMERGENCY VALVE.

brakes take hold and release more rapidly. In that case, opening of the brake valve by means of the brake pedal causes air to flow from the reservoir to the relay valve, where it serves merely to open the valve. The relay valve then permits air from a near-by reservoir to flow to the brake chambers. Relay valves are needed particularly in the case of trailers, owing to the greater distance of the trailer brake chambers from the brake valve on the tractor; and as a trailer also requires an emergency valve, to apply the brakes automatically in case it should become uncoupled from the tractor, the two devices are combined in a single unit in the relay-emergency valve (Fig. 9). As shown in Fig. 2, trailers are fitted with a separate air reservoir, and there is a connection from this reservoir to the relay-emergency valve. There are three lines of tubing leading to and from the relay-emergency valve: One, known as the service line, to the brake valve on the tractor; another, known as the emergency line, to the air reservoir on the tractor, and the third to the brake chambers on the trailer.

Referring now to Fig. 9, when the driver depresses the brake pedal, air from the brake valve enters the relay valve through the connection at the top. This forces the diaphragm below the connection down, thereby closing the exhaust and opening the conical valve connected to it. Air from the trailer reservoir then enters the relay valve at the left and passes through the valve which has just been opened and through another conical valve on the right—which is open as long as the emergency line from the tractor is intact—to the brake chambers, applying the trailer brakes. When the driver releases the pedal, the relay valve is closed by its spring, and the pressure in the brake lines is released through the exhaust.

From Fig. 9 it can be seen that the emergency line from the tractor connects to the emergency valve at the lower right. The emergency-line pressure enters the space below the emergency-valve diaphragm and holds the conical valve connected to it open. At the same time it shuts off communication between the trailer reservoir and the outlet to the brake chambers. Trailer-reservoir pressure acts on the diaphragm. As the emergency-line pressure acts on a larger area of the diaphragm than the trailer-reservoir pressure, the former normally predominates and holds the diaphragm and the attached valves in the "up" position. But if the emergency-line pressure drops to a fraction of its normal value, or vanishes completely (as if the trailer became uncoupled and the hose connection between tractor and trailer were severed), the trailer-reservoir pressure would force the diaphragm down, closing the upper and opening the lower valve controlled by it, and admitting air from the trailer reservoir to the brake chambers.

Quick-Release Valve—The quick-release valve, which is installed on the frame close to the front brake chambers, serves to exhaust the air from the brake chambers rapidly when the brake pedal is released. A sectional view of the device is shown in Fig. 10. A line from the brake valve connects centrally at the top, and there is an exhaust port centrally at the bottom, in addition to which there are threaded openings at opposite sides for connections to the brake chambers. Within the valve housing there is a diaphragm which when the pedal is released and the pressure in the line from the brake valve vanishes, is pressed upward by a spring and closes off communication with the brake valve, thus allowing the air in the brake chambers to escape rapidly through the large central exhaust port. When the pedal is depressed, the air pressure forces the diaphragm down, thereby closing the exhaust port and establishing communication between brake valve and brake chambers. In the drawing the valve is shown in the "brake released" position.

Figs. 11 and 12 show designs of a low-pressure indicator and a stop-light switch, both of which are electric contactors. Both devices comprise a diaphragm which is acted upon by air pressure on one side and a spring on the other. In the low-pressure indicator the spring tends to close the contacts, while the reservoir pressure tends to hold them apart. When

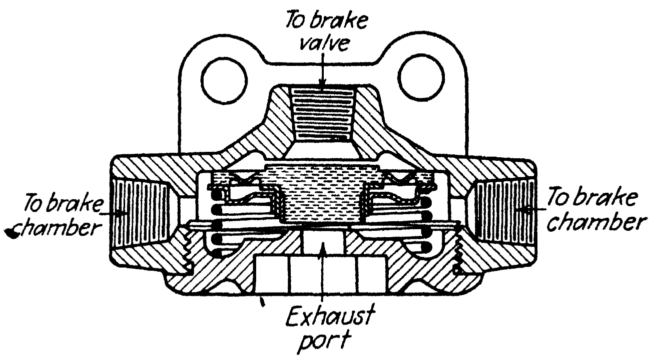


Fig. 10.—Quick-Release Valve.

the reservoir pressure drops below a predetermined value (usually 60 psi) the spring overcomes the air pressure, the contacts close, and an electric bulb is lit or a buzzer sounded. The stop-light switch comprises similar elements, but the contacts are normally held apart by the spring. As soon as pressure begins to build up in the brake chambers on brake application, the diaphragm exposed to this pressure overcomes the spring and presses the contact points together. The stop light at the rear of the vehicle then lights up.

Limiting Valve—In Fig. 13 is shown a sectional view of the limiting valve, which enables the driver to set a definite limit to the force of front brake application, to prevent locking of the front wheels and losing steering control on slippery roads. The device is mounted on the instrument board and is connected in the line between the brake valve and the front brake chambers. It is provided with a milled knob with an arrow on it pointing to marks on a dial back of the knob. The knob is secured to a screw with coarse thread, by means of which the spring *A* can be put under more or less pressure. Pressure on spring *A* forces piston *B* downward, and the motion is communicated to intake valve *C*, which is thus opened. When the intake valve is open, air from the brake valve can pass freely through the limiting valve to the brake chambers. However, the air presses upward against piston *B*, and when the pressure reaches the value for which the

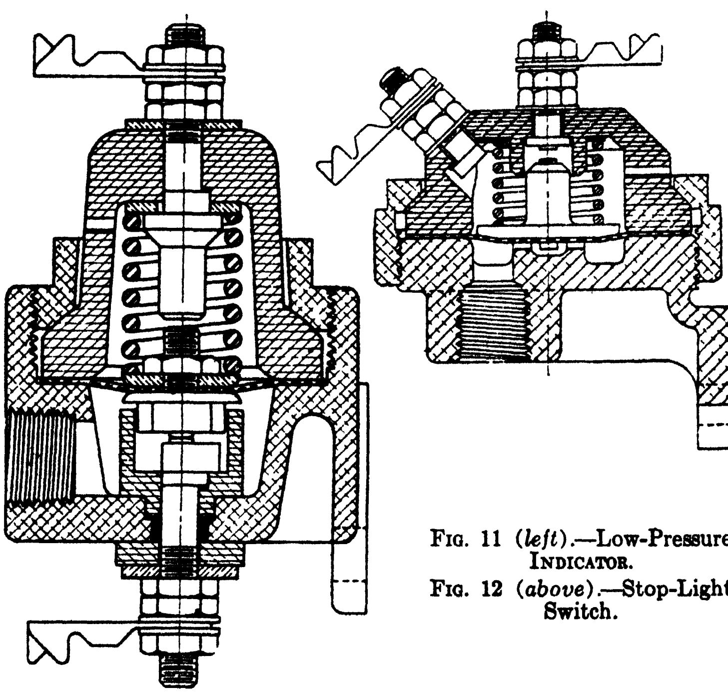


FIG. 11 (*left*).—Low-Pressure Indicator.
FIG. 12 (*above*).—Stop-Light Switch.

limiting valve is set, the piston is forced upward, spring *A* is compressed, and intake valve *C* is allowed to close under its spring pressure. Hence, no more air can pass through the limiting valve to the brake chambers, and the force of application is thus limited. There are four markings on the dial, 20, 45, 53, and full limiting pressure, respectively.

Foundation Brake—The term "foundation brake," which has come into use among manufacturers of power-braking

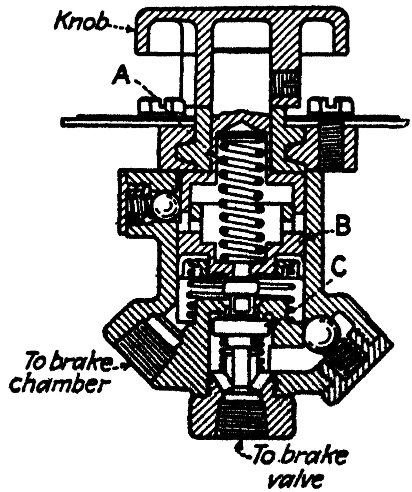


FIG. 13.—LIMITING VALVE.

equipment, is applied to the brakes proper—the drums, shoes, and expanding mechanism—as distinguished from the equipment by which the brakes are applied and released. Foundation brakes for air-operated heavy-duty vehicles usually comprise two-shoe brakes of the fixed-anchor type, operated by a "constant-rise" or involute cam (Fig. 14) and a slack adjuster of the worm-and-gear type. It is recommended that with this type of brake at least 1 sq in. of lining area be provided for every 40 lb of gross vehicle weight. The advantage of the constant-rise cam lies in the fact that it ensures constant force of brake application for a given air pressure throughout the life of the lining. As the lining wears, adjustments are made in the slack adjuster, so that with the brake fully applied the adjuster arm is always substantially at right angles to the

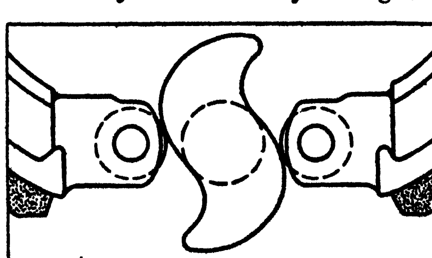


FIG. 14.—"CONSTANT-RISE" CAM AND BRAKE-SHOE TOES WITH ROLLER FOLLOWERS.

pushrod. At each adjustment the angular relation between the cam and the adjuster arm is changed. Relatively heavy linings are commonly used on air-operated brakes, hence numerous adjustments can be made during the life of the lining. The line of contact on the brake cam moves through a considerable arc during this period, but the linear motion of the roller cam follower per degree of cam rotation always remains the same.

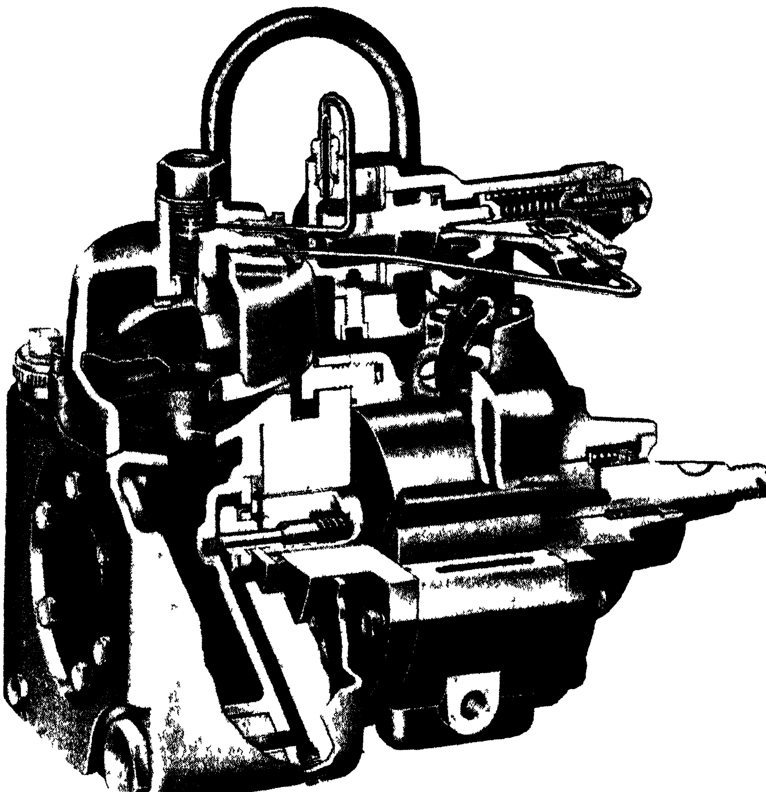


FIG. 15—CUTAWAY VIEW OF WAGNER AIR COMPRESSOR.

Wagner Air Brakes—Automotive-type air brakes are being manufactured also by Wagner Electric Corporation of St. Louis. The Wagner system comprises a compressor of the rotating type, an air dome, and a control-valve assembly. These parts usually are combined in a single unit (Fig. 15), which is mounted on the engine, but where space limitations interfere with the installation of such a unit, the air dome and valve assembly are made a separate unit that can be installed in any convenient position and connected to the compressor by tubing. Fig. 16 is a schematic drawing of the compressor system.

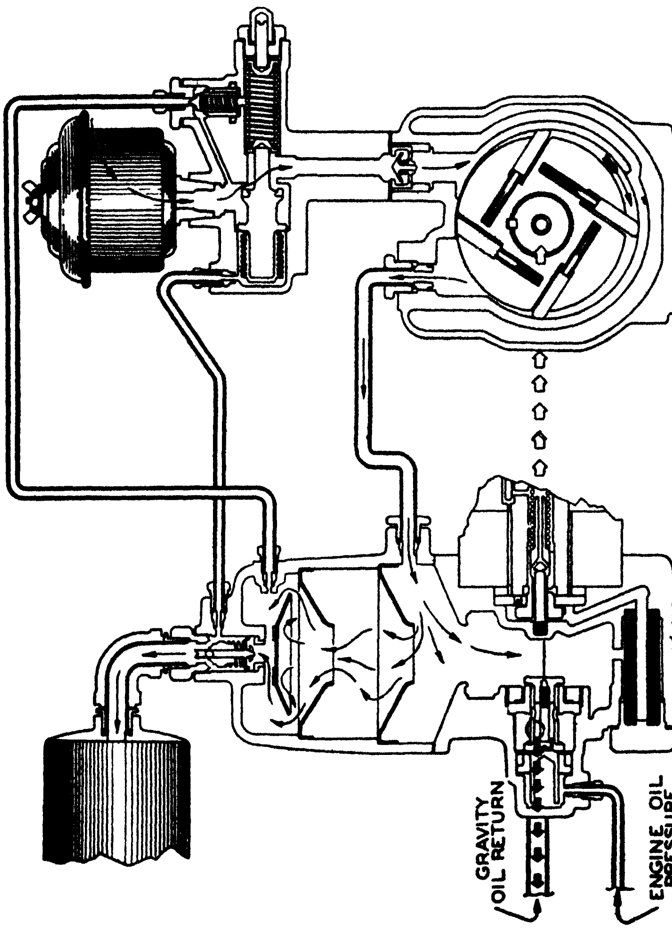


FIG. 16.—SCHEMATIC DRAWING OF WAGNER COMPRESSOR SYSTEM.

The Wagner compressor consists of a stator with a cylindrical bore, and a rotor eccentrically mounted therein, with a clearance of 0.001 in. at the top and $\frac{1}{4}$ in. at the bottom. The rotor carries four sliding blades, at an angle to the radial, which are held in contact with the wall of the stator by springs and sliding pins behind them. These blades divide the space between the rotor and the stator wall into four chambers. Each chamber increases in volume during one-half revolution of the rotor and decreases during the next half. While its volume increases the chamber is open to the inlet port, and air is drawn into it through an air cleaner. Then, as the volume of the chamber decreases, the air in it is compressed, and during the last part of the compression period the chamber is open to the discharge port and compressed air is forced from it into the air dome.

To ensure copious lubrication of the blades and other rotating parts, and also to improve the seal of the chambers, oil is supplied to the compressor from the sump of the air dome in a constant stream. In the compressor the oil becomes mixed with the air, and the discharge consists of a mixture of air and oil mist. In the air dome the oil is separated from the air by centrifugal action. The mixture enters the dome tangentially and is set whirling therein, and as a result of the centrifugal force on it the heavier oil collects on the outer wall of the dome and drains into the sump, while the lighter air is forced toward the center and passes through a check valve in the head of the dome to the reservoir. Oil returning from the air dome to the compressor has to pass through a filter in the sump.

Ordinarily the compressor is driven from the engine through a V belt and operates as long as the engine is running. In the reservoir the pressure is maintained between 85 and 100 psi. The governor or pressure-regulating mechanism is shown at the upper right in Fig. 15 and directly below the air cleaner in Fig. 16. In the latter illustration the inlet valve is located directly below the air inlet port. At the left the valve is surrounded by a bellows subjected to the reservoir pressure, while at the right is shown the valve spring, by means of which the reservoir pressure can be adjusted. When the reservoir pressure reaches the value for which the control valve is set, this valve closes and the pumping action ceases. The reservoir pressure has to drop by from 10 to 15 psi before the valve opens again, and air, therefore, is being compressed intermittently.

Above the spring of the air-inlet valve in Fig. 16 is located a relief valve. When the air-inlet valve is closed or nearly closed, the pumping action creates a partial vacuum in the intake line. This vacuum pulls down the relief valve and opens the air dome to the atmosphere. A small amount of air is allowed to bleed past the relief valve to facilitate oil circulation and cooling. Oil from the sump of the dome enters the compressor through the oil passage in the rotor shaft. The air pressure in the dome always is in excess of that in the compressor, even when there is no compressor action, and this excess pressure forces oil from the sump into the compressor.

The Wagner compressor is made in two types, self-lubricated and engine-lubricated. Fig. 15 shows the self-lubricated type. In the engine-lubricated type the circular end plate of the oil sump shown at the left in Fig. 15 is replaced by an oil supply valve to which are connected a pressure line from the engine-lubricating system and an overflow or return line to the engine crankcase. While air is being compressed there is a relatively high pressure in the air dome, and this pressure immerses a charging valve in engine oil to the overflow level in the supply valve housing. During this period the valve is partly filled with oil. At the end of the compression period the engine-oil pressure overcomes the reduced

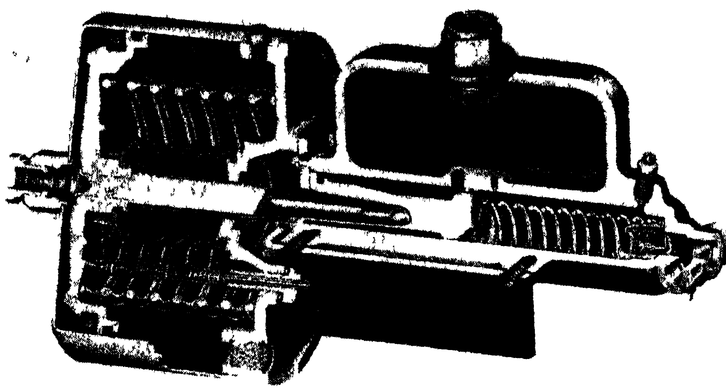


Fig. 17.—Wagner "Power Cluster."

sump pressure and moves the charging valve into the sump. If the oil level in the sump is low, the charging valve discharges into the sump, whereas if the level is high, oil passes from the sump into the charging valve and during the next compression period returns to the engine crankcase through the overflow line. Thus a substantially constant oil level is maintained in the air-dome sump.

Air-Hydraulic Brakes—Another power-braking unit developed by Wagner, known as the Power Cluster, is essentially a device for converting air pressure into hydraulic pressure. A sectional view of the unit is shown in Fig. 17. An air power cylinder is combined with a hydraulic master cylinder and reservoir. This unit makes possible the actuation of conventional hydraulic brakes by air power. The bore of the air power cylinder is four times that of the master cylinder, and the ratio between the hydraulic pressure and the air pressure is maintained at 15 to 1. A stroke indicator incorporated in the unit shows when the piston in the master cylinder nears the end of the latter as the brake is being applied, so that brake adjustment is called for. The unit originally was developed for field installation in trucks with conventional hydraulic brakes, but is being offered also for factory installation in new trucks.

Bendix Air-Pak Unit—A device for operating hydraulic brakes of trucks and tractor-trailer combinations by means of compressed air is being manufactured also by Bendix Products Division of Bendix Aviation Corporation and is marketed as the Air-Pak unit. A sectional assembly view of the unit is

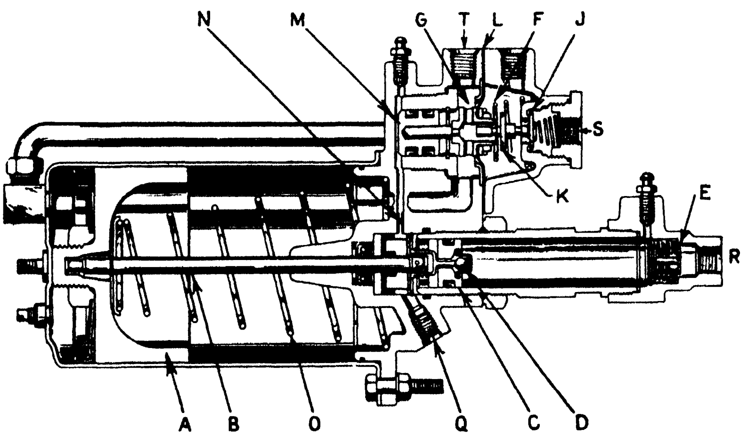


FIG. 18.—BENDIX AIR-PAK UNIT IN SECTION.

shown in Fig. 18. It consists of three sections, as follows: (1) An air cylinder containing a piston *A*, a pushrod *B*, and a return spring *O*; (2) an hydraulic slave cylinder containing the hydraulic piston *C*, a by-pass valve *D*, and a residual-line pressure check valve *E*; and (3) a hydraulically actuated air-control valve, including chambers *F* and *G*, a diaphragm *L*, and interconnected poppet valves *J* and *K*. The air-control valve regulates the air pressure delivered to the air cylinder in accordance with the hydraulic pressure created in the master cylinder by the pressure of the driver's foot on the brake pedal. The air pressure against the left surface of air piston *A* is always proportional to the pressure the driver exerts on the brake pedal.

It will be understood that the Air-Pak unit is used in a power braking system which also includes an hydraulic master cylinder, a compressor and air reservoir, hydraulic brakes on the wheels, and hydraulic and air lines connecting the different units. When the driver presses down on the pedal to apply the brakes, hydraulic pressure in the master cylinder and in the line which connects to the Air-Pak unit at *Q* is increased. The pressure in passage *N* and in the space to the left of piston *M* also increases, and when it overcomes the force of the valve-return spring, piston *M* and the diaphragm assembly move to the right, bringing the seat of valve *K* up against the valve. Communication between chambers *F* and *G* is thereby shut off.

Further increase in the pressure of the master cylinder causes the interconnected poppet valves *J* and *K* to move to the right, thereby opening air valve *J*. Air now enters through compressed-air inlet *S*, flows past valve *J* into valve chamber *F* and thence through the connecting tube directly above the air cylinder into that cylinder. The entrance of compressed air increases the pressure in chamber *F*, and since chamber *G* on the opposite side of the diaphragm is always open to the atmosphere at *T*, the increased pressure differential on the diaphragm causes the diaphragm assembly to move piston *M* to the left against the hydraulic pressure on its left face. Valve *J* then closes and prevents further inflow of compressed air. Thereafter the air pressure in the cylinder remains constant and the valve mechanism is said to be in its "lap" position, since valves *J* and *K* are both closed.

If the master cylinder pressure is increased further, the whole valve mechanism again moves to the right, valve *J* is reopened, and additional compressed air is admitted to chamber *F* and to the air cylinder. When the pressure in chamber *F* balances the increased hydraulic pressure against piston *M*, the valve mechanism again moves to the left into another lap position, in which the brakes are applied with a greater, constant force.

Any decrease in the pressure on the brake pedal initiates a reversed sequence of events. Complete release of the pedal causes the pressure in the master cylinder to drop to zero and the piston *M* and the other valve parts to be returned to the released positions, in which they are shown in Fig. 18. All compressed air is then exhausted from the Air-Pak through chamber *F*, ports in the valve, and chamber *G*. In Fig. 18 *R* is the outlet port to the wheel cylinders.

Brake Cylinders—While most of the early automotive air brakes comprised diaphragm chambers, more recently use is being made also of single-acting cylinders containing pistons fitted with piston rings. The diaphragm chamber undoubtedly is lighter and less bulky than a cylinder, and it possesses the further advantage that in it there is no air leakage. The cylinder, on the other hand, has the advantage that the stroke is not as limited as with a diaphragm chamber, hence it is

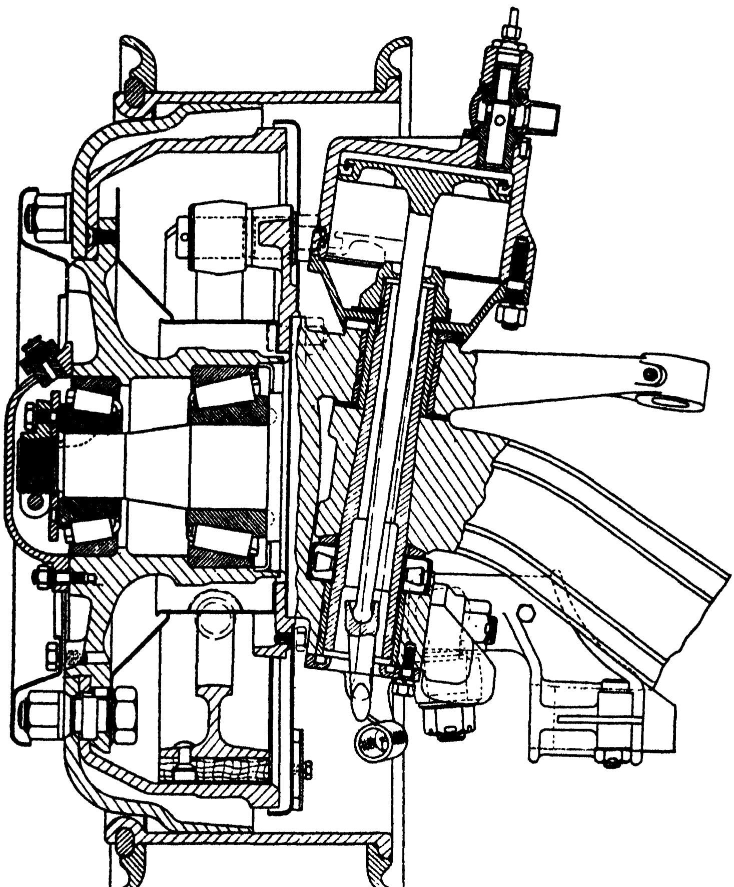


FIG. 19.—AIR-BRAKE POWER CYLINDER MOUNTED ON FRONT AXLE (AEC).

not necessary to make such frequent adjustments to compensate for lining wear. A brake cylinder mounted on the front axle of an AEC (British) bus is shown in Fig. 19. The cylinder is mounted on the axle concentric with the hollow knuckle pin, and the piston rod, which is integral with the piston, passes through the knuckle pin and acts on a brake lever below the knuckle. Steering motion is provided for by a ball-and-socket connection between the piston rod and the brake lever.

Vacuum Brakes—With air brakes, the pull on the brake rods, or the force of brake application, is due to a difference of pressures on opposite sides of a diaphragm in a chamber, or of a piston in a cylinder. Assuming a cylinder to be used, one side of the piston is exposed to atmospheric pressure, and the other to the much higher reservoir pressure. The braking effect is proportional to the difference, the so-called "pressure differential." A similar effect can be produced by subjecting one side of the piston to atmospheric pressure and the other side to a pressure below atmospheric, by exhausting air from the corresponding end of the cylinder. This is the principle of the vacuum brake. With vacuum brakes power cylinders are used predominantly, but diaphragm chambers operated by vacuum are also available.

Much of the pioneer work in connection with the application of vacuum power to brakes was done by Victor Kliesrath and Caleb Bragg, who organized the Bragg-Kliesrath Corporation of Long Island City, N. Y. That firm was taken over by Bendix Products Corporation in 1930.

Air can be exhausted from the power cylinder most conveniently by connecting it to the engine inlet manifold, in which there is a partial vacuum as long as the engine is running. When the engine is under heavy load, the vacuum is only light, but whenever the vehicle is to be stopped, or its speed reduced materially, the throttle is closed, and the vacuum then may attain 24 in. of mercury column, and should be at least 16 in. As 1 in. of mercury column is substantially equal to 0.5 psi, the under-pressure in the manifold at closed throttle ranges between 8 and 12 psi. This vacuum or under-pressure is admitted to one end of the power cylinder by means of the control valve. The maximum pressure differential, of course, is much smaller in the case of a vacuum brake than in that of an air brake, and for a given braking effect the power cylinder of a vacuum system must be made proportionately larger.

Atmospheric- and Vacuum-Suspended Systems—When one end of the power cylinder is in communication with the atmosphere at all times, while the other is placed in communication with the inlet manifold when it is desired to apply the brakes, the system is said to be "atmospheric-suspended." With such a system, when the brake valve is opened, air passes from the power cylinder to the inlet manifold, and dilutes the charge passing through the latter, and if the cylinder happens to be quite large relative to the engine displacement, the charge may be so diluted that it becomes unignitable, in which case the engine stalls. This difficulty is overcome in the "vacuum-suspended" system, in which both ends of the power cylinder are normally in communication with the inlet manifold, and both sides of the piston therefore are subjected to the same under-pressure. When the brakes are to be applied, one end of the cylinder is shut off from the manifold and opened to the atmosphere, and the excess pressure in that end then forces the piston toward the low-pressure end, thereby applying the brakes. The atmospheric-suspended sys-

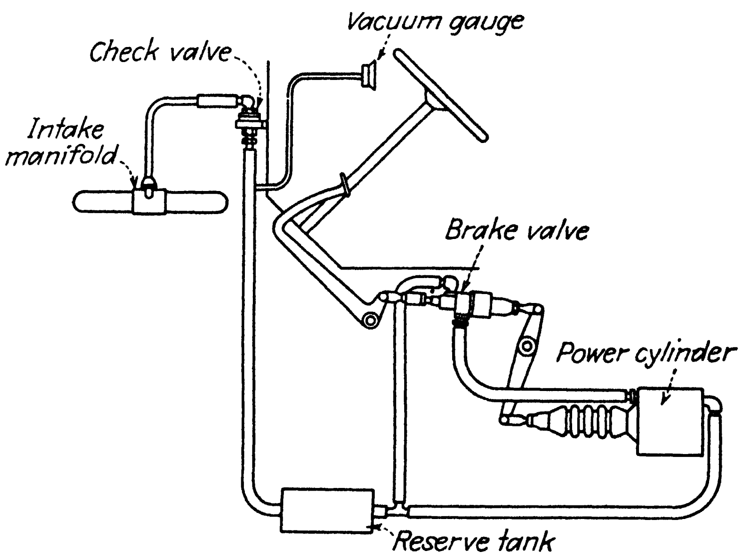


FIG. 20.—INSTALLATION DIAGRAM OF VACUUM-SUSPENDED SYSTEM.

tem has the advantage of greater simplicity, for which reason it was used to a certain extent during the early years of vacuum-power braking, but all modern vacuum brakes are vacuum-suspended.

Control Valve—With vacuum brakes, the same as with air brakes, it must be possible to closely control or graduate the brake effort, which means that it must be possible to hold the degree of vacuum in one end of the cylinder at any point between zero and the inlet-manifold vacuum, and a suitable control valve is provided. The valve, of course, is actuated by means of the brake pedal, and the arrangement is such that in the event there is no vacuum in the inlet manifold, the operator can apply the brake directly, by exerting sufficient pressure on the pedal. It is also desirable that the force of brake application be in direct proportion to the pressure exerted on the pedal, so that the operator may "feel" the force of application. This is made possible by a floating valve inserted in the linkage between the brake pedal and the cross shaft (or master cylinder) from which the force of brake application is transmitted to the individual brakes.

An installation diagram of a vacuum-suspended system is shown in Fig. 20. A tube connecting the intake manifold to the power cylinder leads to the dash, where a check valve is inserted in the line. This valve permits air to flow from the power cylinder into the intake manifold, but prevents flow in the opposite direction. It serves the double purpose of preventing combustible mixture from getting into the power cylinder and of maintaining a vacuum in the braking system if the engine should stop for any reason while the brakes are applied. A branch from the vacuum line connects to a vacuum gauge on the instrument board. A reserve tank or vacuum tank inserted in the vacuum line serves the same purpose as the air reservoir in an air-brake system. From the reserve tank one line leads to the control valve, while another leads directly to the power cylinder, connecting to it at the end opposite the piston rod. From the control valve there is a connection to the piston-rod end of the cylinder. With this system, when the brakes are released, both ends of the cylinder are under vacuum, and when the brakes are to be applied, air is admitted to the piston-rod end through the control valve.

Vacuum-Boosted Hydraulic Brakes—The vacuum-power system described in the foregoing was designed for the operation of mechanical brakes, which were connected to the power cylinder by a linkage. Since that time, so far as the service brakes of road vehicles are concerned, mechanical brakes have been almost completely replaced by the hydraulic type. Hydraulic brakes at first were used on passenger cars only, but larger sizes have been placed in production, and such brakes now are available for the whole range of gross vehicle weights for which vacuum power is well suited. For this reason the original vacuum brakes as described in the foregoing are of little more than historical interest, and a detail description of their mechanism therefore is omitted. Their place has been taken by vacuum-actuated hydraulic brakes, of which a number of different makes were available in 1951.

In vacuum-boosted hydraulic brakes the brake pedal connects to the piston of a hydraulic master cylinder, and brake

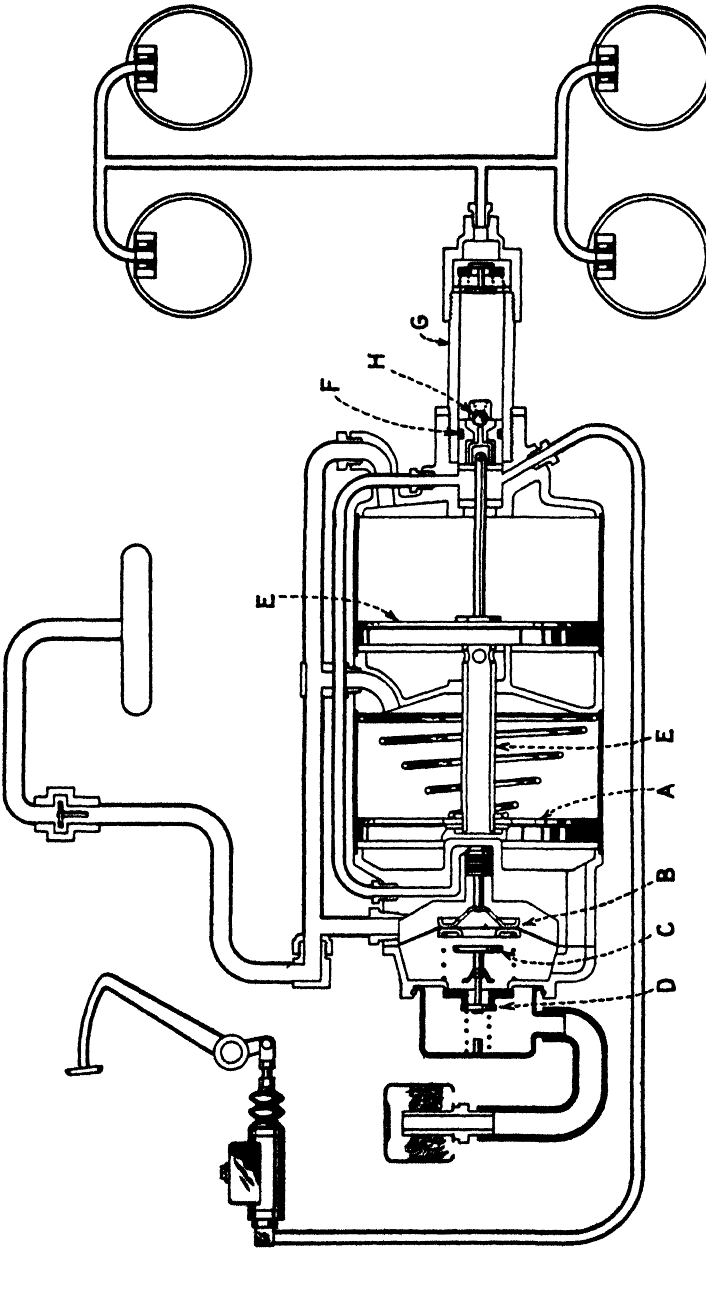


FIG. 21.—SCHEMATIC DRAWING OF BENDIX HYDROVAC BRAKING SYSTEM.

fluid forced from this cylinder when the pedal is depressed actuates the valve which connects one end of the power cylinder either to the intake manifold or the atmosphere. The piston of the power cylinder acts directly on the piston of a hydraulic slave cylinder, from which latter there are lines to the hydraulic wheel cylinders.

Bendix Hydrovac—A vacuum power booster for hydraulic brakes is being manufactured by Bendix Products Division in both single-piston and tandem-piston types. A schematic diagram of the tandem-piston type is shown in Fig. 21. The entire assembly consists of three units. First at the left is shown the hydraulically actuated vacuum-control valve, which controls the degree of brake application or release. The control valve comprises a hydraulically actuated piston *A*, a diaphragm *B*, a vacuum poppet *C*, and an air poppet *D*. In the center is shown the power cylinder, which contains the tandem pistons *E,E* and a pushrod connecting these pistons to piston *F* of the hydraulic slave cylinder *G*. This slave cylinder, which forms the third unit of the assembly, has a check valve *H* incorporated in its piston. In some models a residual line-pressure check valve is incorporated in the outlet from the slave cylinder. The reason for the use of tandem pistons is that the thrust on the piston rod is twice as great as with a single piston of the same diameter. Owing to the relatively low limiting vacuum pressure, it takes a large cylinder to produce the necessary piston thrust, and space conditions usually limit the size of cylinder that can be conveniently installed on a truck or bus.

As the brake pedal is depressed, the hydraulic pressure generated in the master cylinder is transmitted to piston *A* of the control valve and to piston *F* of the slave cylinder. Hydraulic pressure on piston *A* closes vacuum poppet *C* and opens air poppet *D*, thereby admitting atmospheric air to the control side of the power cylinder. The resulting unbalance of pressure on opposite sides of pistons *E,E* creates a thrust that is transmitted to slave-cylinder piston *F* through the pushrod. As the slave-cylinder piston starts to move, check valve *H* closes, trapping fluid under pressure in the brake lines. The total hydraulic pressure generated and transmitted to the wheel cylinders is the sum of the pressure generated in the master cylinder by pedal pressure and that generated in the slave cylinder by thrust of the power-cylinder pushrod.

When the brake pedal is released, the pressure on hydraulic piston *A* is reduced, and as a result air poppet *D* closes and vacuum poppet *C* opens again. All compartments of the power cylinder now are in communication with the intake manifold, pressures on opposite sides of pistons *E,E* are equalized, and the pistons are returned to their released positions by the return spring. When piston *F* nears the end of its release stroke, check valve *H* opens to permit full release of the brakes.

Vacdraulic Brake—This vacuum-actuated booster for hydraulic brakes, which in the previous edition was described as a product of Empire Electric Brake Company, has been completely redesigned and is now being produced by Kelsey-Hayes Wheel Company of Detroit. A cross section of the Model 130-S, used on certain Chrysler, DeSoto, and Dodge models in 1950 and 1951, is shown in Fig. 22. This unit is used together with the conventional hydraulic master cylinder and comprises a diaphragm chamber, a power cylinder, and a valve assembly. Connection from the master cylinder is made to the unit at the point indicated, and from the power cylinder connections are made to the wheel cylinders. In the drawing the parts are shown in their relative positions immediately after the driver has begun to press down on the brake pedal.

Hydraulic fluid from the master cylinder passes through a space within the power cylinder between the power piston and a trip plate, to a cylinder containing the large end of a control piston. The small end of that piston is located in a cylinder that communicates with the power cylinder. Consequently, the large end of the control piston is subjected to the hydraulic pressure in the master cylinder, and the small end to that in the power cylinder. The hydraulic pressure generated in the master cylinder acts on the power piston. An air valve under the influence of the control piston is now open, and atmospheric air is drawn in through an air cleaner and passes through this valve and through a tube from the valve chamber to the outer compartment of the diaphragm chamber. Since air is being exhausted from the inner compartment by manifold suction, there is an excess of pressure in the outer compartment over that in the inner one, and as a result the diaphragm is being forced toward the right. The force on it is transmitted through a pushrod to the power piston and adds to the pressure exerted on that piston by the hydraulic fluid from the master cylinder. As the power piston moves toward the right, the hydraulic pressure generated in the power cylinder increases. This pressure acts on the small end of the control piston, which has about one-fourth the area of the large end, and when the total pressure on the small end becomes equal to that on the large end the control piston moves toward the left and allows the air valve to close.

The valve-actuating plate is then said to be "poised." No more air can enter the outer compartment of the diaphragm chamber. There is then a definite excess of pressure in that compartment over that in the inner compartment, and the brakes are being applied with a definite force. Of this force a fraction of the order of one-fourth originates at the brake pedal and the remainder in the diaphragm chamber.

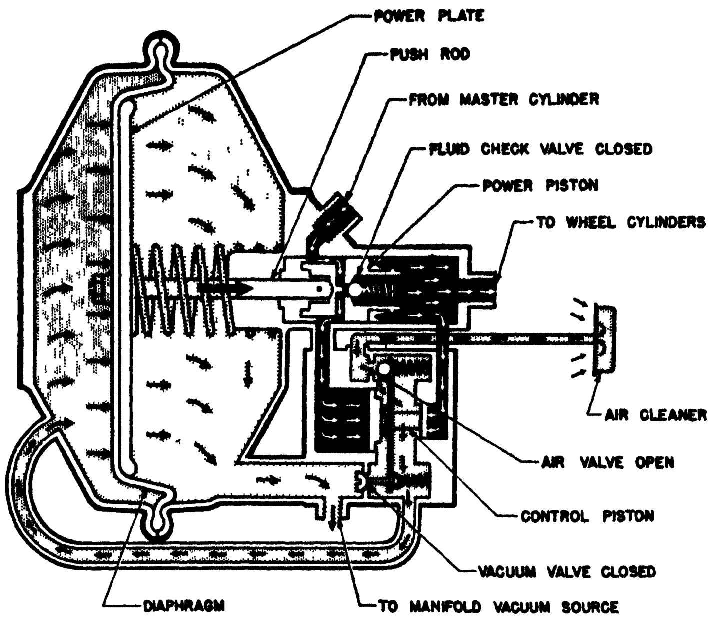


FIG. 22—SECTIONAL DIAGRAM OF VACDRAULIC BRAKE UNIT MODEL 130-S.

If the driver increases his pressure on the brake pedal, the air valve opens, more air is admitted to the outer compartment of the diaphragm chamber, and the brakes are applied with greater force, in direct proportion to the increase in pedal pressure. If, on the other hand, the driver lets up slightly on the pedal, the master-cylinder pressure is reduced, and the excess total pressure on the small end of the control piston causes that piston to move toward the left and allows the vacuum valve to open. Air is then drawn from the outer to the inner compartment, the excess of pressure in the outer compartment over that in the inner one is reduced, and the brakes are applied with less force.

When the brake pedal is released, the ball check valve in the power piston is held open by a pin on the trip plate.

There is then practically no hydraulic pressure on either end of the control piston, and the vacuum valve is open and is held in position by its spring. Thus the two compartments of the diaphragm chamber communicate with each other, both are under the same degree of vacuum, and the diaphragm and power plate are held at the beginning of their stroke by the return spring.

An improvement in the Model 130-S over earlier designs results in the recovery of hydraulic fluid that may seep past the low-pressure piston. This seepage is trapped by a return-system seal and returned to the master cylinder through a passage around the power cylinder, a small check valve, and a tube.

Midland Hy-Power Vacuum Brake—Midland Steel Products Company produces a vacuum-boosted hydraulic brake for motor trucks. The installation (Fig. 23) comprises a con-

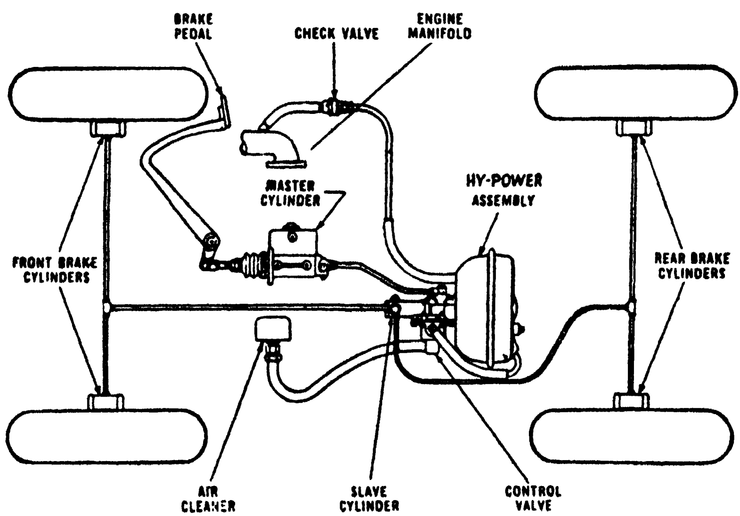


FIG. 23.—INSTALLATION DIAGRAM OF MIDLAND HY-POWER VACUUM BRAKE.

ventional master cylinder, a vacuum power chamber with a diaphragm which acts on the piston of a hydraulic slave cylinder, and hydraulic lines from the latter to the wheel cylinders. Combined with the power chamber and slave cylinder is a valve which controls the admission of atmospheric pressure and inlet-manifold pressure to the rear compartment of the power chamber. The front compartment of the chamber communicates permanently with the inlet manifold and therefore is under vacuum whenever the engine is running. Fig. 24 is a part-sectional view of the Hy-Power unit, while Fig. 25 is a sectional view of the valve mechanism and the slave cylinder on a larger scale.

When the brakes are released, the return springs hold the valve in such a position that the rear compartment is shut off from the atmosphere, but communicates with the front compartment, so that both compartments then are under vacuum. If now the driver presses down on the brake pedal, he forces hydraulic fluid from the master cylinder through

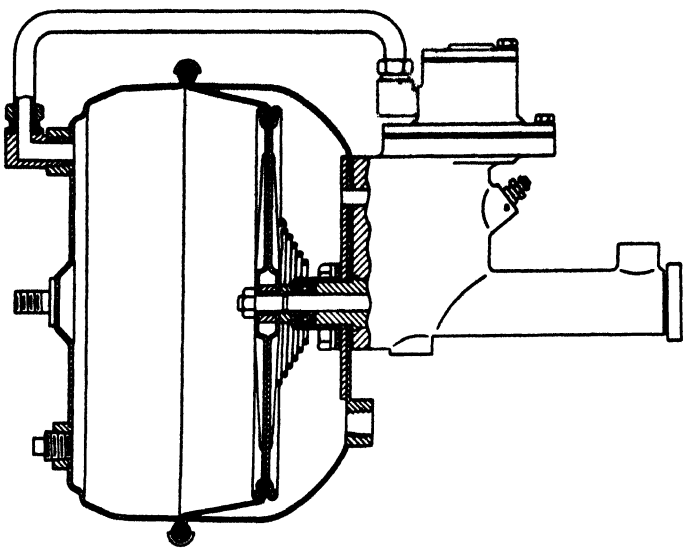


FIG. 24.—PART-SECTIONAL VIEW OF MIDLAND HY-POWER UNIT.

a tube to the slave cylinder. The entrance to the latter is at a point which is straddled by the slave-cylinder piston when in the released position, but the fluid can pass through openings in the piston, a check valve in the piston, and through piston and cup orifices into the brake lines. It can also pass upward to the control-valve plunger and seals.

As the driver depresses the brake pedal, hydraulic pressure builds up in the line, and when a pressure of about 40 psi is reached, the pressure on the control-valve piston, forcing the latter upward, shuts off communication between the two compartments of the power chamber and opens the rear compartment to the atmosphere. Atmospheric air then enters the rear compartment, and the resulting excess pressure in that compartment forces the diaphragm and pressure plate forward. This motion is communicated through a pushrod to the piston of the slave cylinder, which is moved in the same direction. The check valve in the slave-cylinder piston closes, and any farther motion of the piston forces fluid into the

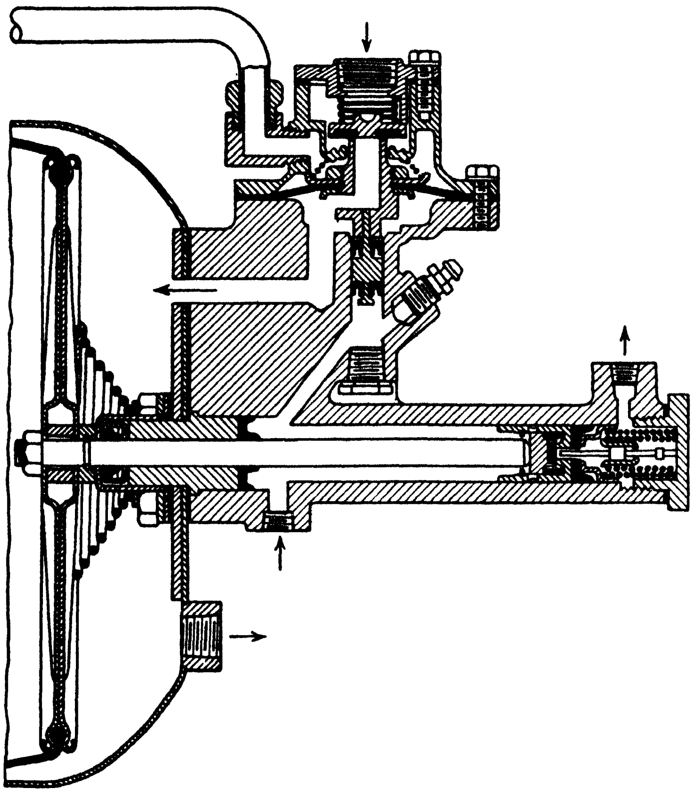


FIG. 25.—MIDLAND CONTROL VALVE AND SLAVE CYLINDER IN SECTION.

line and puts it under pressure proportional to the excess air pressure in the power chamber.

The control valve is carried on a diaphragm which is subject to the excess air pressure in the rear compartment, and the valve therefore floats between the air pressure on the diaphragm and the hydraulic pressure on the piston. With the brake pedal moved to any given position, there will be a definite hydraulic pressure in the line between the master cylinder and slave cylinder, and atmospheric air will enter the rear compartment until the pressure therein, acting on the control-valve diaphragm, balances the hydraulic pressure on the control-valve piston. Then the valve will close and the brake will produce a certain retarding force, depending on the amount of air which has entered the rear compartment. If now the driver wants to brake harder, he increases his pressure on the pedal. This increases the hydraulic pressure in the line to the slave cylinder. The greater pressure on the control-valve piston then opens the valve again, allowing more air to enter the rear compartment, and the resulting greater excess pressure in that compartment produces greater hydaulic pressure in the slave cylinder and the lines to the wheel cylinders. Thus the force of brake application is at all times proportional to the pedal pressure.

When the driver takes his foot off the pedal to release the brakes, the pressure in the line from the master cylinder to the slave cylinder vanishes and the control valve is forced down by its return spring, reestablishing communication between the two compartments of the chamber and equalizing their pressures. All movable parts of the braking system are then returned to their released positions by their respective return springs. If vacuum power should fail for any reason, the hydraulic pressure generated in the master cylinder will apply the brakes directly, but, of course, much more pedal pressure will be required to produce a certain braking effect.

Electric Brakes—Brakes applied by the attractive force exerted by an electro-magnet on its armature are being used to a limited extent in the automotive field, especially on trailers. Their advantages are that they are simple in design and installation, and that the necessary connection between tractor and trailer can be easily made and broken. As compared with brakes that are applied by air pressure, electric brakes also have the advantage that they operate practically without lag. While they consume power as long as the brake is on, whereas other types consume power only while they are being applied, this is not a serious disadvantage, as the power consumption is quite moderate. According to figures issued by one manufacturer, it amounts to approximately 0.01 watt per lb-ft of brake torque under full application in small sizes, and to only about 0.002 watt per lb-ft in large sizes.

Fig. 26 is a cutaway view of one type of the Warner electric brake, which has an annular magnet core of U-section concentric with the brake drum. This makes a rather efficient form of magnet, as the magnetic circuit has a large cross section and is quite short. The driver applies the brake by moving the handle of a controller, which first closes the battery circuit through the magnet coil and the controller resistance, and upon further motion cuts out the resistance step by step, thus increasing the braking power. Referring to the illustration, when magnet *2* is energized, it tends to adhere to armature *3*, secured to the brake drum, and is carried

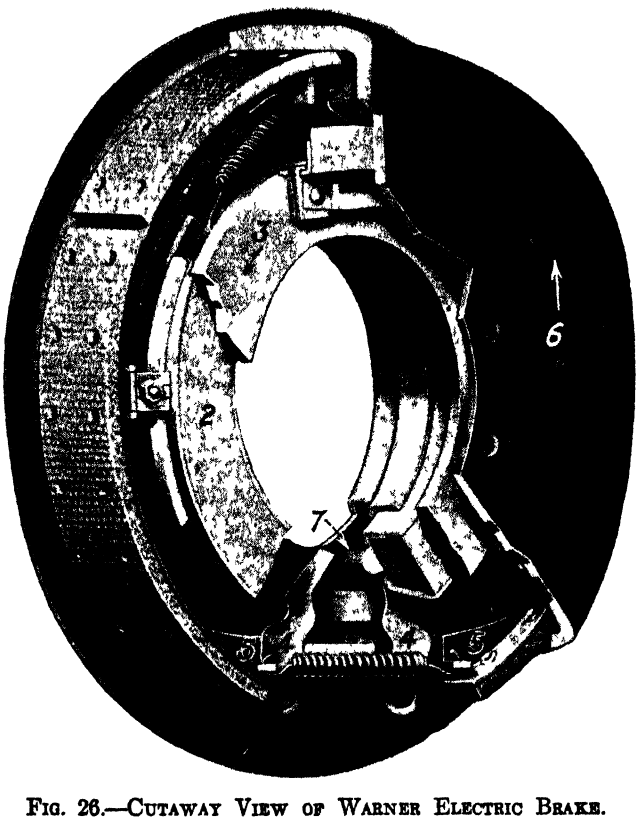


FIG. 26.—CUTAWAY VIEW OF WARNER ELECTRIC BRAKE.

around with it through a small arc. A lug *7* on the magnet then comes in contact with an arm on cam *4*, and by turning the cam, expands the brake band into contact with the drum. There are two cams, one for each direction of motion, and the brake is equally effective for both. The actuating mechanism has sufficient range so no adjustments are required throughout the life of the lining. When the lining is worn down to the rivet heads, the magnet abuts against a stop, thereby automatically preventing scoring of the drum.

A battery and a safety switch are installed on the trailer. The switch is connected to the tractor by a light chain which snaps when the trailer becomes uncoupled, and the trailer brake is then applied automatically by battery current.

Auxiliary Brakes or Energy Dissipators—When heavily loaded trucks or trailers descend long grades, a great deal of energy must be dissipated by their brakes, and the latter often reach excessive temperatures, which result in rapid wear of their linings. Overheating of the brakes involves the risk of brake failure. It is customary to use the engine as a brake on long down-grades, but this practice, while usually affording adequate safety, has its disadvantages. With the transmission in low or an intermediate gear, the engine will be turning over at high speed, hence bearing loads and rates of bearing wear are likely to be high. Moreover, as the throttle is closed, the piston motion creates a powerful suction that draws oil from the crankcase into the combustion chambers, where it forms carbon deposits and fouls the spark plugs. The desirability of some means for dissipating the heat generated in long descents, somewhere else than at the brake surfaces, has long been recognized, but such auxiliary brakes or energy dissipators have come into use only comparatively recently. Since they are actually needed only on the heavier types of vehicles operating in districts with long, steep grades, they are not likely to be fitted as regular equipment. They seem to be needed most on trucks used in the lumbering industry and in other "off the highway" operations.

Auxiliary electric or hydraulic brakes have two distinct advantages. With the conventional brakes long, steep grades must be descended at low speeds in order to prevent the brakes from reaching excessive temperatures. With auxiliary brakes, since the heat-dissipating capacity is greatly increased, higher speeds can be maintained safely. The auxiliary brake serves only to hold the vehicle down to a safe speed and cannot be used to bring it to a dead stop. But as the regular brakes are applied only lightly or not at all during the greater part of the descent, they are always in good condition and capable of effecting a rapid stop. Therefore, with auxiliary brakes it is possible to maintain higher average speeds and to save wear and tear on both the engine and the regular brakes, in addition to which the safety of operation is enhanced.

Warner Eddy-Current Brake—An eddy-current auxiliary brake is being manufactured by the Warner Electric Brake Manufacturing Co. of Beloit, Wis. As shown partly in section in Fig. 27, it consists of an electric generator comprising an internal, multi-polar field frame and an external combined armature and fan, mounted on the propeller shaft of the vehicle. The field frame is of the type in which a single field coil excites all of the poles of a magnet ring, but two such field units are used. The armature is a mere shell without windings, in which eddy currents are caused to circulate as it moves past the field poles. Exciting current is derived from a special generator driven by belt from a pulley on the dissipator shaft. As the armature shell is an integral part of the fan and the air can circulate freely past it and the field poles, the heat generated is carried off rapidly, and the field coils do not attain excessive temperatures.

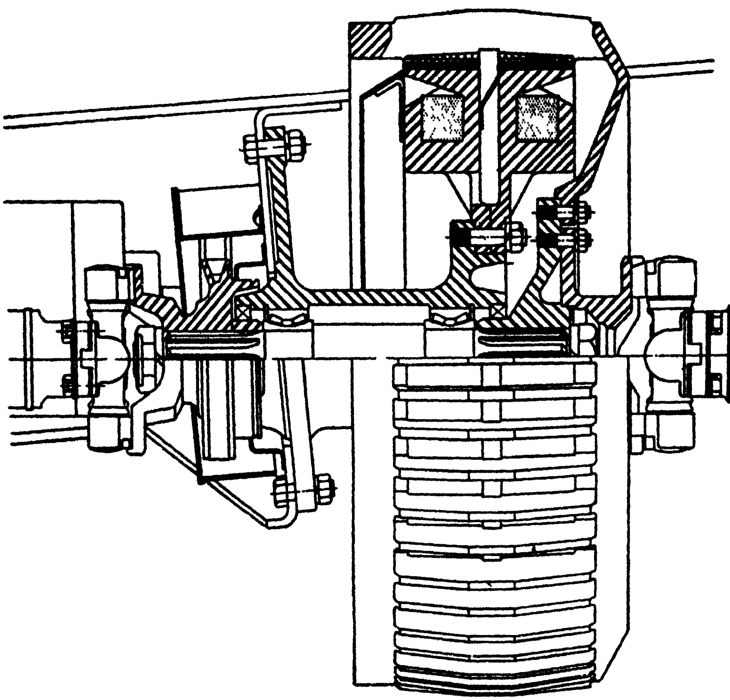


FIG. 27.—WARNER EDDY-CURRENT BRAKE.

During normal operation of the truck, no current flows through the field coils of the dissipator, and the latter, therefore, is inactive. There are two switches in the exciter circuit, mechanically connected, respectively, to the accelerator and the brake pedal. When the accelerator pedal is released the dissipator cuts in at a speed which is determined by the setting of a control knob on the instrument board. If the vehicle speed at the moment of pedal release exceeds the cut-in speed, the dissipator cuts in immediately; otherwise only after the cut-in speed has been reached. If the brake pedal is depressed slightly, the dissipator cuts in immediately regardless of the vehicle speed and the setting of the control knob. The dissi-

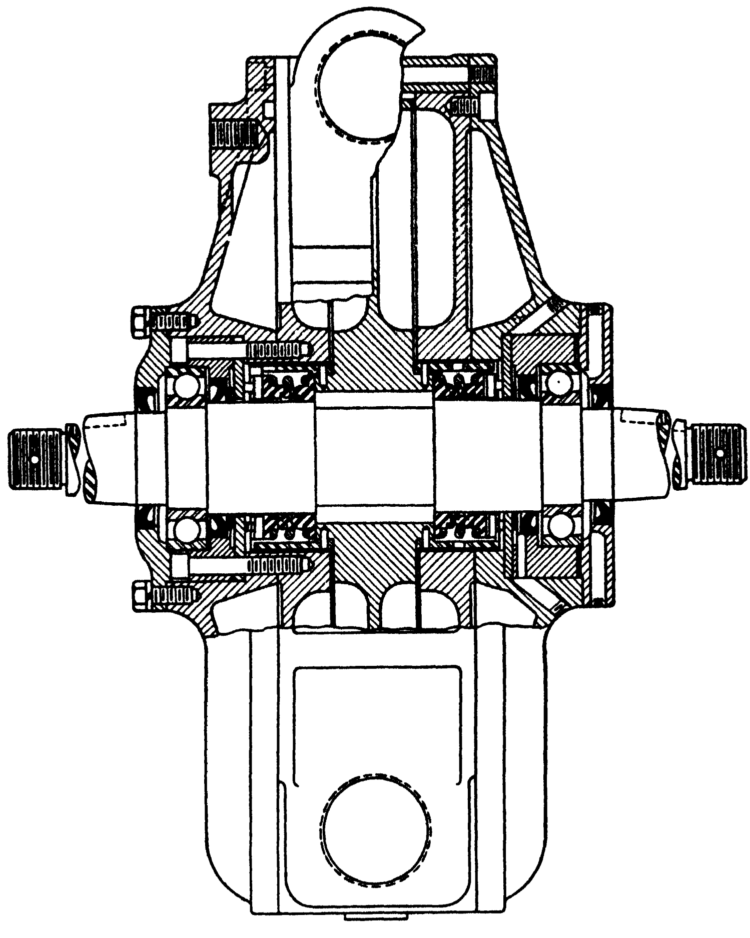


FIG. 28.—PART-SECTIONAL VIEW OF THE HYDROTARDER.

pator then continues to develop approximately its full retarding torque down to about one-third the maximum vehicle speed. Thereafter the retarding torque decreases with the speed, down to the cut-in speed of the generator (8-9 mph), at which speed it vanishes completely.

Hydrotarder—A hydraulic energy dissipator known as the Hydrotarder is being manufactured by the Parkersburg Rig & Reel Co., Parkersburg, W. Va. The unit, which is incorporated in the propeller shaft of the truck, consists of right- and left-hand stator castings and a rotor located between them and mounted on the retarder shaft, both stators and rotor being enclosed in a cast housing, as shown in Fig. 28. Pockets are formed in both the rotor and the stators by tangential or sloping partition walls. If the unit is filled with water, rotation of the rotor causes it to circulate between the rotor and stators, and as the flow is continually interrupted by the partition walls of one unit passing those of the other, it is highly turbulent and results in the generation of much heat. Provision is made for circulating the heated water through the truck radiator, to disperse the heat.

There are tangential inlet and outlet connections on the circumference of the housing. From the housing the water passes to a surge tank mounted back of the truck cab. A pipe from this tank leads to the top tank of the radiator. From the bottom tank of the radiator the water is drawn by the engine pump, which forces it through the cylinder jackets. At the outlet from the jackets there is a three-way valve which is controllable from the dash. All of the water moved by the engine pump enters this valve, and when the Hydrotarder is not in use, all of it returns directly to the radiator. When the Hydrotarder is active, more or less of the water flows to it, according to the setting of the control valve. The Hydrotarder then contains more or less water and produces a retarding torque which depends upon the amount of water in it and on the speed of its rotor, which latter is proportional to the vehicle speed. For a given setting of the control valve the retarding torque varies substantially as the square of the speed. The radiator is vented to the top of the surge tank.

CHAPTER XVI

Types of Chassis Springs and Their Characteristics

Motor-vehicle chassis are supported on the axles or their equivalents through the intermediary of springs, whose primary function it is to cushion the shocks imparted to the wheels by road obstacles. All of the early motor vehicles had leaf springs, which were then in universal use on horse carriages, and were fitted also to railway cars. Coil springs came into limited use at a relatively early date, however, particularly on low-priced cars. Probably their first application in American practice was on the Brush runabout, brought out in 1908, which had its frame supported on four coil springs working in tension. Coil springs working in compression entered the field when independent front springing was adopted and rigid front axles were eliminated, during the early thirties.

A third type of chassis spring that has seen limited use, particularly abroad, is the torsion bar. It consists of a spring-steel bar, usually cylindrical, of which one end is rigidly secured to the chassis frame, while the other is supported in a bearing mounted on the chassis and carries a lever arm (or connects to a crank) adjacent to the bearing. The free end of the lever is articulated to the steering head in the case of front springs, and to the wheel spindle in the case of rear springs. The reaction on the end of the lever arm due to the load which it imposes on the steering head or wheel spindle, causes the bar to twist or deflect torsionally, and the elasticity of the bar gives the needed cushioning effect.

A fourth variety of steel spring is the volute, which is made of strip steel wound to form a distorted spiral. The center line of the strip or blade does not lie in a plane, but describes a paraboloid, and the spring is loaded axially. A peculiarity of this type of spring is that as the load is increased, the coils "bottom" one after another and become inactive, and the rate of the spring, or its stiffness, increases in consequence. This makes it suitable for use as a buffer spring or a helper spring, for it can be so designed that while it is relatively soft under normal load, it will take a relatively large load before it closes up solid.

Rubber springs, in which the rubber is subjected to torsion or shear, came into limited use during the late forties, mainly on buses. Rubber has a higher modulus of elasticity than steel, and is able to store a greater amount of energy elastically per unit of mass, for which reason it has been extensively used to cushion the shocks of aircraft landings. Considerable work has been done on air springs, and a number of experimental installations have been made, but to the best of the author's knowledge, no such springs are being used in production models. An advantage of air springs is that, the same as with volute steel springs, the rate increases with the load, besides which it is comparatively easy to modify the rate of an existing installation with this type of spring.

Leaf Springs—The most widely used type of leaf spring is the semi-elliptic, illustrated in Fig. 1. It consists of one main leaf, which usually has its ends formed into eyes for connection to the spring horns or spring brackets by means of either bolts or shackles and bolts; and of a number of

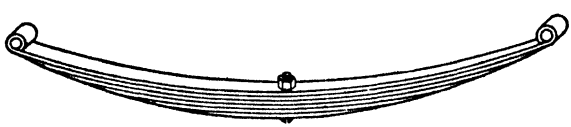


FIG. 1.—SEMI-ELLIPTIC LEAF SPRING.

shorter leaves, the lengths usually decreasing substantially uniformly with the distance from the main leaf. There is an exception to this rule in the case of some springs for heavy vehicles, with a large number of leaves, in which the second leaf, or that next to the main leaf, may be substantially as long as the latter, or even enwrap the eye of the main leaf. The various leaves of a leaf spring usually are held together by a center bolt.

Semi-elliptic springs are secured to the axle at their center, and have their ends pin-jointed or shackled to the frame, as already explained. Quarter-elliptic springs have been used for light cars. With these the "butt" is bolted to the frame, and the end of the main leaf connected to the axle in some way. Formerly a number of other spring types also were in use, including three-quarter elliptic, full-elliptic, and platform springs, the latter being made up of three semi-elliptic springs, two longitudinal and one transverse, connected together by shackles. It is now known that, provided the springs are correctly proportioned, with the same amount of steel in the springs and the steel equally stressed, the same riding qualities are obtained with semi-elliptic as with the more complicated spring types, and the latter therefore are no longer being used.

Spring Steels—The common spring steel, which has long been used for carriage and railway springs, is a carbon steel containing approximately one per cent of carbon, and is now known as S.A.E. steel No. 1095. For motor vehicles, however, alloy steels are used exclusively. Alloy spring steels include three principal types, viz., molybdenum steels of the 4000 series, chromium steels of the 5100 series, and silico-manganese steels of the 9200 series. Of these the latter probably are in most extensive use; at least they have been in use longest.

Molybdenum spring steels were developed by the Chrysler Corporation, and are used by it exclusively. They are also known under the name of Amola steels. There are two S.A.E. standard molybdenum spring steels, but they differ only with respect to the carbon content. Both contain 0.75-1.00 Mn, 0.20-0.35 Si, and 0.20-0.30 Mo. The No. 4063 has a carbon content of 0.60-0.67 per cent, while the No. 4068 contains from 0.63 to 0.70 per cent carbon. The latter is used chiefly for the heavier gauges (more than ¼ in. thickness), because of its better hardenability. Specifications of Chrysler Corporation's Amola spring steels differ slightly from the above with respect to the carbon, silicon and manganese ranges. Molybdenum spring steels are normalized before the leaves are worked to shape.

The 5150 chromium spring steel contains 0.48-0.53 C, 0.70-0.90 Mn, 0.20-0.35 Si, and 0.70-0.90 Cr, and the P and S are held to less than 0.04. A 5160 chromium spring steel, similar to the former but with roughly ten "points" more carbon (0.55-0.65) also has been used, as has a chrome-silicon steel with ten "points" less chromium (0.50-0.80) and a higher silicon content (1.30-1.60), the latter so far only for coil springs.

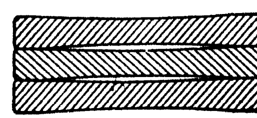
The principal silico-manganese steel is the 9260, which contains 0.55-0.65 C, 0.70-1.00 Mn, 1.80-2.20 Si, and not over 0.04 P and S each. The heat treatment for this steel consists in quenching in oil from approximately 1600 F, and drawing at a temperature which will give the desired hardness. Spring steels usually are drawn to show between 420 and 470 Brinell. In this connection it is of interest to recall that the tensile strength of a steel is substantially 500 times its Brinell hardness. A. A. Remington, who tested silico-manganese spring steels in England during the 1920s, found one particular grade to have a tensile strength of 215,000 psi, an elastic limit of 184,000 psi, an elongation of 6 per cent, and a reduction of area of 22 per cent. Somewhat better values are now being obtained, and it is common to figure with an elastic limit of about 200,000 psi for all types of alloy spring steels. During World War II two National Emergency (NE) steels of this type were developed which, in addition to all of the elements given above, contain some chromium, one between 0.10 and 0.25 per cent, the other between 0.25 and 0.40 per cent.

In the tempered condition the various alloy spring steels will show tensile strength of about 225,000 psi. The maximum permissible working stress, that is, the maximum stress under static load, varies inversely as the thickness of the leaves. This is due to the fact that the hardness decreases with increase in depth below the surface, and the mean hardness and mean strength therefore are less in plates of heavy gauge. With alloy steels, the permissible stress under static load varies from about 55,000 psi for the heaviest gauges, to about 90,000 for the lightest. However, in the case of automotive springs, where the deflection, and, therefore, the stress, is positively limited, it is preferable to base design calculations on the maximum stress (the stress at maximum deflection). With alloy steel this can be made from 110,000 to 140,000 psi, depending on the gauge. It is advantageous to work at comparatively high stresses, because the weight of steel required for the springs varies inversely as *the square* of the stress, and spring steel is rather expensive. On the other hand, the stress preferably should not be made so high that the endurance life of the spring will be less than that of the car as a whole.

Carbon steel of the 1095 grade is used for leaf springs in applications where weight is less and cost more important than in motor vehicles. It is hardened and drawn in the same way as alloy spring steel. This steel will show a tensile strength of 170,000-190,000 psi, and an elastic limit of 140,000-150,000 psi, depending on the drawing temperature. As the amount of energy which can be stored elastically in a unit mass of steel varies as the square of the permissible stress, which in turn varies in direct proportion to the elastic limit, it follows that for any given service, carbon-steel springs must weigh nearly twice as much as alloy-steel springs. In the design of automotive-type carbon-steel springs, maximum stresses under static load of from 45,000 to 75,000 psi were figured with, the higher stress being used with thin leaves. The maximum stress was held between 70,000 and 100,000 psi.

Leaf Sections—Spring leaves normally are rolled to a substantially rectangular section. The long sides of the rectangle may be either straight or slightly concave, while the short sides are made convex, with a radius equal to the thickness of the leaf. A pack of concave leaves, with the concavity considerably exaggerated, is shown in Fig. 2. The reason for making the leaves concave is somewhat obscure. Some say it is done to provide a space for lubricant between the leaves, while others say it is done to prevent "tethering" of the stacked leaves. Even a slight convexity will cause the leaves to tether, and to make sure that there will be no convexity, the leaves are made slightly concave. An S.A.E. standard for motor-vehicle springs sets limits on the concavity of the leaves, that is, the difference between thicknesses at the center and at the edges. The lower limit is always zero, and the surface of the leaf therefore can be absolutely flat, while the upper limit varies with the width and thickness, from 0.004

FIG. 2.—PACK OF CONCAVE SPRING LEAVES.



in. for leaves up to 3/8 in. thick and up to 2.5 in. wide, to 0.020 in. for leaves over 7/8 in. thick and over 5 in. wide.

Spring leaves for a long time were rolled to Birmingham Wire Gauge sizes, but these sizes are inconvenient, because there is no definite relation between successive numbers of the gauge. Leaves, therefore, were rolled also to fractional-inch thicknesses. A new gauge for spring leaves was originated by the Engineering Department of Chrysler Corporation during the thirties, and has since been incorporated in an S.A.E. standard for motor-vehicle springs. It is based on the principle of preferred numbers, the thicknesses being so chosen that there is a constant ratio between the stiffnesses of successive gauge numbers. As the stiffness varies as the cube of the thickness, cubes of the thicknesses form a geometrical series. The following table gives all of the thicknesses or gauge numbers which are likely to be used for motor-vehicle springs, together with the squares and cubes of the thicknesses, which powers occur in some of the spring formula. At present spring leaves conforming to this gauge are being rolled only in alloy steels.

Table I—S.A.E. Spring-Leaf Gauge

t	t^2	t^3
0.132	0.0174	0.00230
0.145	0.0210	0.00305
0.160	0.0256	0.00410
0.176	0.0310	0.00545
0.194	0.0376	0.00730
0.214	0.0458	0.00980
0.237	0.0561	0.0133
0.262	0.0686	0.0180
0.291	0.0847	0.0246
0.323	0.104	0.0337
0.360	0.130	0.0467
0.401	0.161	0.0645
0.447	0.200	0.0893
0.499	0.249	0.124
0.558	0.311	0.174
0.625	0.391	0.244
0.702	0.493	0.346
0.788	0.621	0.489

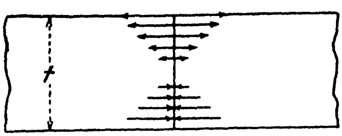
Similar data for fractional inch leaves are given in Table II.

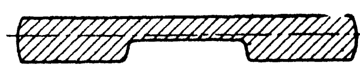
Table II—Fractional-Inch Spring-Leaf Thicknesses

Fraction	t	t^2	t^3
3/16	0.187	0.0350	0.00654
7/32	0.219	0.0480	0.01050
1/4	0.250	0.0625	0.01562
9/32	0.281	0.0790	0.02219
5/16	0.312	0.0973	0.03037
11/32	0.344	0.1183	0.04071
3/8	0.375	0.1406	0.0527
13/32	0.406	0.1648	0.0669
7/16	0.437	0.1910	0.0834
15/32	0.469	0.2200	0.1032
1/2	0.500	0.2500	0.1250

High-Efficiency Sections—The ordinary semi-elliptic spring is equivalent to two cantilever beams, each considered encastré at the center of the spring, and loaded at its end. In a loaded cantilever beam the material in the upper half of the section is in tension, and that in the lower half, in compression. As indicated in Fig. 3, tension and compression are a maximum at the upper and lower surfaces, respectively, and decrease uniformly toward the center, where the so-called neutral fiber is located, which is not stressed at all.

Spring-leaf failures practically always are fatigue failures. That is, a small crack or fissure starts at some point of the surface where, owing to some irregularity or imperfection, there is a concentration of stress, and it gradually extends inward until breakage occurs. It has been observed that such failures always start on the tension side of spring leaves, which is due to the fact that the tensile strength of the material is less than its compressive strength. This observation led to the design of the grooved type of spring leaf, illustrated in Fig. 4, which has been adopted by a number of manufacturers. A groove of substantially one-third the width of the leaf, and of a depth equal to one-half its thickness, is rolled in the compression side, that is, the under side of leaves of conventional semi-elliptic springs. For a given moment of inertia the grooved section must be made 7.4 per cent thicker than the "full" section. The maximum tensile stress in the grooved section then is only 97 per cent that in the full section, while the sectional area of the grooved leaf (and therefore its weight per unit of length) is 93 per cent that of the full section. The weight of spring required to carry a given load varies inversely as the square of the maximum stress, and for the same maximum tensile stress the grooved leaf needs to weigh only $0.97^2 \times 0.93 = 0.875$ (seven-eighths) as





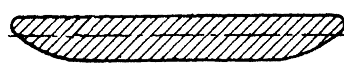


FIG. 3 (*left*).—DIAGRAM OF STRESSES IN CANTILEVER.
FIG. 4 (*upper right*).—GROOVED SPRING LEAF.
FIG. 5 (*lower right*).—SPRING LEAF OF TRAPEZOIDAL FORM.

much as the full-section leaf. This grooved spring leaf was patented by Wm. H. Wallace of Detroit.

Similar savings are effected by giving the plate a substantially trapezoidal form, as shown in Fig. 5. The section is not a true trapeze, as the sides, instead of being straight, are parabolic.

Increasing the Fatigue Strength—During World War I the British Army experienced an unusual amount of trouble from breakage of truck springs, and after the war a research program was put through with the object of improving the design and production processes of vehicle springs. The method of fatigue-testing springs, by subjecting them to a continuous series of similar stress cycles, was then introduced. Among other things it was discovered that if the plates were machined, the fatigue limit, that is, the maximum stress range through which a part can be put indefinitely without failure, was half as high again as it was with plain rolled plates. The reason for this is not hard to understand, for fatigue failures start as small cracks or depressions which gradually widen, and if the plates are smoothly machined there is, of course, much less chance of such defects being present.* Later it was found that the fatigue life of springs can be greatly increased by shot-blasting or shot-peening the tension side of the leaves. This process consists in throwing small, hard globules of chilled iron against the surface of the leaf by centrifugal force, and has the effect of compressing the surface metal, thereby putting it under negative stress. In addition to increasing the fatigue life, shot-peening has the effect of slightly reducing the curvature of the leaves, and this must be allowed for in forming them.

The properties of the spring can be improved also by cold-setting or bulldozing, which process consists in subjecting either the plates individually or the asembled spring to a stress exceeding the elastic limit of the steel. This produces a permanent set, and reduces settling of the spring in service. Cold-setting produces a tensile stress in the material on the compression side, and a compressive stress in the material on the tension side of the plate, the latter effect being similar to that produced by shot-peening the tension side. In service, these negative stresses must be overcome by the load before it can begin to build up positive stresses. Thus the maximum stress is reduced, and the fatigue life increased.

Theory of Leaf Springs—The simplest form of leaf spring is that consisting of a single leaf. Such a spring may be considered either as two cantilever beams loaded at their ends, or as a simple beam loaded at the middle. Fig. 6 is a sketch of such a spring. If we consider each half of the spring as a cantilever and denote the load on one end by P, the half-length of the spring by l, the width by b, the thickness by t, and the modulus of elasticity by E, we get for the deflection of the end of the spring,

$$f = \frac{1Pl}{3EI} = \frac{4Pl^3}{Ebt^3} \text{ in.}$$

(See Cantilever Beams in any textbook on Mechanics.) The bending moment at a distance x from the end of the spring is Px, and the stress in the material at that point is

$$S_x = \frac{6Px}{bt^2} \text{ psi.}$$

* Smoothening the surfaces of the leaves by machining and polishing has been discontinued, because it was found to promote scoring.

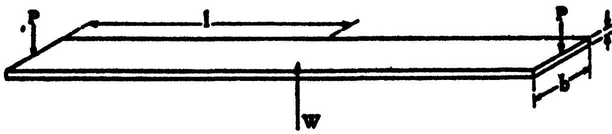


FIG. 6.—DIAGRAM OF SINGLE-LEAF SPRING.

This shows that with a single leaf of uniform section the stress due to the bending moment increases uniformly from nothing at the end to a maximum at the middle of the spring. Therefore, if a uniform-section leaf were used, the material would be very poorly utilized, and one of the objects in using multi-leaf or laminated springs is to make the stress substantially uniform throughout the length. Now suppose we took a number of leaves and assembled them as shown in Fig. 7. Then, if a load were applied to the top leaf, all of the leaves would be deflected equally. If there were n leaves, the deflection would be the same as in the case of a single leaf subjected to a load P/n. Therefore, the deflection of a spring like that shown in Fig. 7 would be

$$f = \frac{4Pl^3}{Enbt^3} \text{ in.}$$

When a spring leaf deflects there are two forces at work, namely, the force due to the load P carried and the force due to internal strains. In a multiple-leaf spring, moreover, as soon as deflection begins, there is sliding motion of the leaves over each other, and this introduces friction as a factor. Friction between the leaves opposes their deflection, and therefore assists the internal forces due to strains in the material. Hence the deflection will be reduced, but the reduction will be comparatively small, and will depend, moreover, on the number of leaves and on the camber of same. Experiments have shown that an eight-leaf spring in the dry condition has its vibration damped out in about three-and-one-half complete periods, or in fourteen quarter periods, each of which

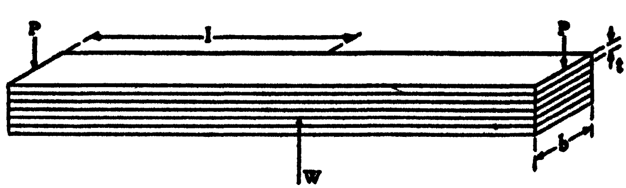


FIG. 7.—ASSEMBLY OF LEAVES OF EQUAL LENGTH.

latter corresponds to one complete deflection of the spring from its position of equilibrium. Hence, only about 7 per cent of the energy stored in the spring when fully deflected is absorbed by friction during the deflection, and in a well-lubricated spring the damping action would be even materially less. This indicates that in an average passenger-car front spring in good condition, the loss of energy through interleaf friction does not exceed 5 per cent during a quarter period, and in an average rear spring, 8 per cent.

Effect of Clamped Portion—The deflection is affected also by the fact that in the car a portion of the spring at the center is restrained by the retaining clips. This part of the spring is not entirely inactive, because no matter how strong the clips or studs may be, and how tightly they may be set up, they will yield elastically under the enormous forces active at the edge of the seat. Moreover, the edges of the seat and of the clamping plate must be beveled or rounded, in order to prevent localization of stress. The clamping, however, has some effect in reducing the deflection of the spring. Tests on a considerable number of springs showed that clamping reduces the deflection under a given load by from 10 to 17 per cent, and since the deflection varies as the cube of the length, the effective length is reduced by from 3.5 to 6.0, or an average of 4.75 per cent. With rubber pads above and below the spring within the clamp, the decrease in effective length is reduced to about one-half, and therefore is roughly 2.5 per cent of the total length. That the proportional decrease in effective length should vary considerably is only natural, because the proportional length of the clamped portion varies widely. Thus in a certain truck rear spring of which data are available, the length of the clamped portion is 15.4 per cent of the total spring length, while in a certain bus rear spring it is only 6.3 per cent.

Spring Deflection—In an actual vehicle spring the leaves are of gradually decreasing lengths, and since the outer end of any leaf is not supported by a leaf below it, the deflection under a given load is greater than in a spring of the form shown in Fig. 7. Reuleaux has shown mathematically that if the lengths of the leaves decrease uniformly, as in Fig. 8, the deflection will be increased 50 per cent, and for such a spring, therefore, the coefficient in the formula for the deflection should be 6 instead of 4. Experience has shown, however, that the actual deflection of the spring is somewhat smaller, which is undoubtedly due to interleaf friction, and that the coefficient should be 5.7 instead of 6. This gives for the deflection of a semi-elliptic spring with n leaves of equal thickness and uniformly stepped lengths, under a load P at each end,

$$f = \frac{5.7Pl^3}{Enbt^3} \text{ in.}$$

The total load on the spring is $2P$, and the rate of the spring (the load per inch of deflection) therefore is

$$R = \frac{2P}{f} = \frac{Enbt^3}{2.85l^3} \text{ lb per in.}$$

The bending moment at the middle of a semi-elliptic spring is Pl. In a spring having n leaves of equal thickness, this moment is equally divided, and the moment on each leaf at the

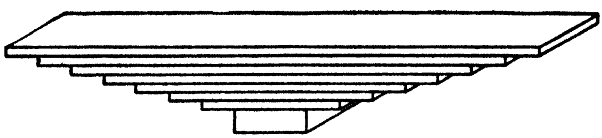


FIG. 8.—ASSEMBLY OF LEAVES OF UNIFORMLY STEPPED LENGTHS.

center is Pl/n. Since the section modulus of each leaf is $bt^2/6$, the unit stress is

$$S = \frac{\frac{Pl}{n}}{\frac{bt^2}{6}} = \frac{6Pl}{nbt^2} \text{ psi.}$$

The foregoing equations enable us to determine the characteristics of a spring already designed. When a new spring is to be designed a new set of equations is needed. The load $2P$ is known, and the spring characteristics, including the rate R, the maximum deflection f, and the maximum stress S are chosen in the light of past experience. The spring length l also is assumed, and it is then necessary to determine the thickness, width and number of leaves that will give the desired characteristics.

From the equations for the deflection and the stress we can derive two expressions for the load P, and by equating these we get the following equation for the stress in terms of the deflection:

$$S = 1.05 \frac{ftE}{l^2} \text{ psi.}$$

This equation can be transformed so as to give directly the correct leaf thickness for the maximum deflection f and the maximum stress S decided upon

$$t = 0.95 \frac{Sl^2}{fE} \text{ in.}$$

The product nb of the number and width of leaves can be found from a transformation of the equation for the spring rate R—

$$nb = \frac{2.85Rl^3}{Et^3}.$$

The width of spring is then chosen, and this immediately gives the number of leaves required.

Camber—In the foregoing we have considered springs of an elementary type, with the object of simplifying the mathematics. Actual springs differ from those discussed in several ways. In the first place, conventional chassis springs are not straight when in the free state, but cambered. Spring plates are made from rolled stock, and after they have been cut to the right length, are bent to form arcs of circles. At least the ordinary or conventional spring plate, which is said to have a true sweep, is thus curved. In a half-elliptic spring, the distance from a line joining the centers of the eyes to the inner side of the main leaf is known as the opening of the spring, or its camber.

Formerly motor-vehicle springs were given considerable camber, the practice probably being a heritage from the horse-vehicle industry, where full elliptic springs were used predominantly, which must have large camber because the maximum deflection depends on it. In motor-vehicle practice, however, it has been found that large camber is undesirable, and at present the tendency is to so design the springs that they are substantially straight when under their full static load. They will then have a moderate positive camber when free, and a corresponding negative camber when deflected to the maximum. One objection to large camber is that it increases the influence of the shackle angle on the rate and other characteristics of the spring.

Grading and Nipping—The equations given in the foregoing pages apply to springs with all leaves of the same thickness, and lengths equally stepped. However, practically all modern vehicle springs are "graded"; that is, they are built up of leaves of different thicknesses. Sometimes only the main leaf is thicker than the others, but in many cases leaves of three or more thicknesses are used, thicknesses decreasing in the direction away from the main leaf. The original reason for grading springs may have been that it permits of obtaining a spring of exactly the required rate with leaves of standard gauge. At present, however, the chief reason is that if the main leaf were no thicker than the others, the spring eye most likely would be too weak.

If all of the plates of a graded spring were given the same curvature (or, rather, if successive plates of a given spring were curved to radii differing only by the thickness of a leaf), they would fit together nicely in assembly, but it is evident that the different leaves of such a spring would not be equally stressed if the spring were loaded. The thin-

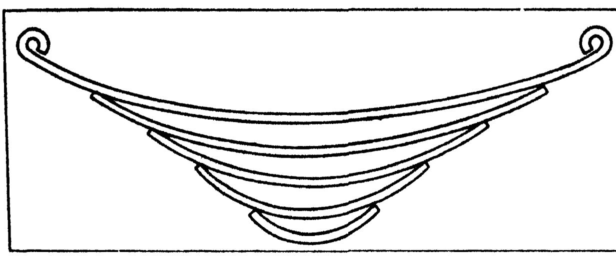


FIG. 9.—ILLUSTRATING "NIPPING" OF LEAF SPRINGS.

ner leaves, obviously, would be stressed less than the thicker ones, and as the load limit would be set by the stress in the thick leaves, the material in the spring would not be used to best advantage. This difficulty is overcome by "nipping" the leaves, that is, by forming the thinner ones to a smaller radius than the thicker ones, as illustrated in Fig. 9. Then, by assembling the leaves by means of a center bolt or center band, the thinner or shorter ones are deflected in the same direction as when load is applied to the spring as a whole, and they are therefore prestressed; while the longer leaf or leaves are deflected in the opposite direction, and they are therefore prestressed in the reverse direction, or negatively prestressed. When the assembled spring is loaded, any small load on it will add proportionately to the initial stress in the thin, short leaves, while in the case of the main leaf it will reduce the initial negative stress. In other words, as the spring is gradually being loaded, the stress in the small leaves builds up from the initial stress due to nipping, that in the long leaf starts to build up from a negative value, also due to nipping, while in intermediate leaves it may build up from zero. As the stresses increase with load less rapidly in the case of the thin leaves, but start from a considerably higher level, it is quite possible to have substantially the same stress in all leaves under maximum deflection. In calculating the stress in any leaf due to a given deflection, the initial positive or negative stress due to nipping must be taken into account. For instance, if the nipping causes a negative stress of, say, 30,000 psi in the main leaf, and if the spring is subjected to a deflection which if applied to the free, unassembled main leaf alone would stress it to 150,000 psi, the actual stress in the main leaf will be only 120,000 psi, for before a "positive" stress can begin to build up, the "negative" stress of 30,000 psi must be overcome. On the other hand, in the case of the thin leaves, which are "positively" stressed by the nip, the stress due to any given load adds to the initial stress.

Equations for Graded Springs—If the lengths of the different leaves of a graded spring vary uniformly, the equations developed in the foregoing for the properties or characteristics of springs with leaves of equal thickness can be readily adapted to the case of graded springs. Most of these equations contain the factors n and t^3, both either in the numerator or the denominator of the right-hand side. If the leaf thicknesses are unequal, their product, nt^3, is replaced by the summation Σt^3, that is, the sum of the cubes of thicknesses of individual leaves. This gives the following equations for the deflection and rate of graded springs, the symbols for which in this case have an inferior g applied to them:

$$f_g = \frac{5.7Pl^3}{Eb\Sigma t^3} \text{ in.}$$

$$R_g = \frac{Eb\Sigma t^3}{2.85l^3} \text{ lb per in.}$$

These equations for the deflection and rate give the values obtained when the spring is tested in a spring-testing machine. When it is installed on the car the rate is modified by the clamping effect of the clips or studs securing it to the axle, and also by the shackle angle. These effects will be discussed in the next chapter.

Stresses in Individual Leaves—To find the maximum stresses in individual leaves of graded springs, we replace the laminated spring by a solid one having the same rate, and made up of a number of sections of the same width and thicknesses as the individual leaves of the laminated spring. This is illustrated by the example of a three-leaf spring in Fig. 10. There the central rectangle represents the main leaf; on each side of it there is another rectangle, representing one-half of the second leaf, and on the outside there are two more rectangles, each representing one-half of the third leaf. All sections are of the same width (though the second and third are split in two for the sake of symmetry), and all are symmetrical with respect to the neutral axis yy. The central section of the solid spring, on line xx, has a moment of inertia which is equal to the sums of the moments of inertia of the center sections of the individual leaves.

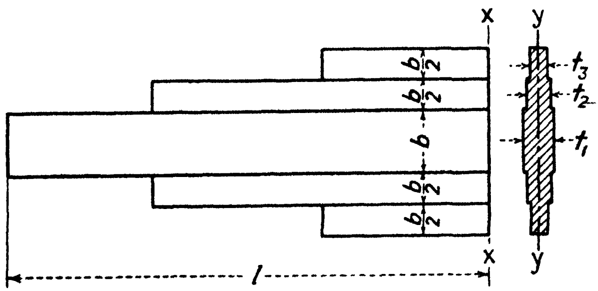


FIG. 10.—CANTILEVER OF VARYING SECTION CORRESPONDING TO ONE-HALF OF A SEMI-ELLIPTIC SPRING.

If now a load P is applied to the free end of the solid spring of length l, the maximum stress in the central section on line xx will be

$$S = \frac{6Pl}{bEt^2} \text{ psi,}$$

which is the same as that in the main leaf of the laminated spring under the same load. There is no stress in the material at the neutral axis yy, and the stress in each section increases from the neutral axis outward at the same rate. As the surfaces of the thinner leaves are closer to the neutral axis than the surfaces of the main leaf, it is obvious that the maximum stress will be less in the former, in direct proportion to their thicknesses. The thinner leaves therefore would not be used to full advantage if they were unstressed when the spring is without load. This, as already explained, is corrected by "nipping" the leaves.

Leaf Ends—In laminated springs the lengths of the leaves are graduated to ensure a more nearly uniform stress distribution and better use of the material. Of course, in a spring leaf the stress always varies from a maximum at the surfaces to zero at the neutral axis, and this cannot be helped, but efforts can be made to keep the stress uniform along the length. If the individual leaves have the form of circular arcs, and are cut off square, the material is not uniformly stressed over the whole length, because whereas the bending moment increases uniformly from the end to the middle of the spring, the resisting moment changes in definite steps at the ends of individual leaves. To ensure uniform stress distribution, the resisting moment must be made to also change uniformly along the length, and this result can be brought about in a simple manner by making the end of each leaf V-shaped, in such a way that the point of the V on one leaf comes where the leaf next above it begins to taper off. This

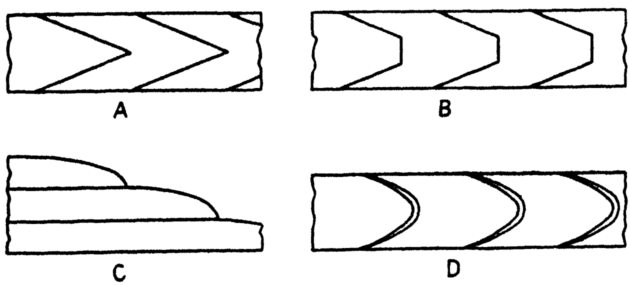


FIG. 11.—TYPES OF LEAF ENDS OR "SPRING POINTS."

gives a leaf end (or spring point) of the shape shown at *A* in Fig. 11. As it is not necessary to have absolutely uniform stress distribution, the tip of the V often is cut off, which gives a leaf end of the form shown at *B*. Instead of gradually decreasing the width of the leaf, the same end can be achieved by decreasing its thickness. However, whereas the stiffness of a spring leaf varies directly as its width, it varies as the cube of its thickness, and in order that the resisting moment of the leaf may decrease uniformly toward the tip, the vertical section of the overhanging portion must be what is known as a cubical parabola ($y^3 = ax$), as shown at *C*. The stiffness of the end of the leaf can be varied gradually also by decreasing both the width and the thickness, as shown at *D*. This design is known as the egg-shaped beveled leaf end. Another, somewhat similar form is known as the taper rolled end. That portion of the leaf overhanging the next shorter one is rolled to a taper, and may be either sheared off square or left rounded as rolled. To avoid high local pressures, the end must be so sheared that the edge in contact with the next-longer leaf is slightly rounded.

When leaves are sheared off square without tapering, the overhanging portion usually is given a slight negative nip, so that each leaf contacts the next longer one a short distance from the end. If this were not done, there would be a pronounced load concentration at the very end, and a consequent scraping action between leaves.

Sample Calculation—We will now illustrate the application of the foregoing equations by a practical example. We will assume that a front spring is to be designed for a light truck, which will have to support a maximum static load of 1500 lb; that the static deflection under full load is to be 2.5 in., and the clearance under this condition, 2.25 in. Symmetrical springs are to be used, 40 in. long (which makes $l = 20$ in.), and the width will be assumed to be 2 in. The springs are to be made of alloy steel and nipped. We will figure with an actual stress in the main leaf, when the clearance is fully taken up, of 120,000 psi, and if a nip negative stress of 25,000 psi is figured with, the calculated stress for 4.75 in. deflection will be 145,000 psi.

The first thing to do is to find the thickness of main leaf that will result in a stress of 145,000 psi under a deflection of 4.75 in. We have

$$t = 0.95 \frac{145{,}000 \times 20^2}{4.75 \times 29{,}000{,}000} = 0.400 \text{ in.}$$

We will therefore use a main leaf of 0.401 in. thickness, which is an S.A.E. gauge number.

Next we must determine the number of leaves required and the thicknesses of the various shorter leaves. In this connection the equation for the rate comes handy. As the spring deflects 2.5 in. under a load of 1500 lb, its rate is $1500/2.5 = 600$ lb per in. Inserting in the equation for the rate,

$$600 = \frac{29{,}000{,}000 \times 2 \times \Sigma t^3}{2.85 \times 20^3},$$

from which it follows that

$$\Sigma t^3 = \frac{600 \times 2.85 \times 20^3}{29{,}000{,}000 \times 2} = 0.2358 \text{ in.}^3$$

The cube of the thickness of the main leaf is 0.0645, which leaves 0.1714 for the remaining leaves. By referring to Table I we find (by trial and error) that the spring can be made up of the following six leaves:

No. of Leaves	*Thickness, t*	*Cube of Thickness, t^3*	*Total*
1	0.401	0.0645	0.0645
2	0.360	0.0467	0.0934
3	0.291	0.0246	0.0738
		Grand total	0.2317

This is within 2 per cent of the calculated value of 0.2359, and will meet the requirements. The lengths of the shorter leaves may be made 35, 30, 24, 18 and 12 in., respectively, measured between the centers of the tapered ends. These dimensions may be regarded as the equivalent lengths of the

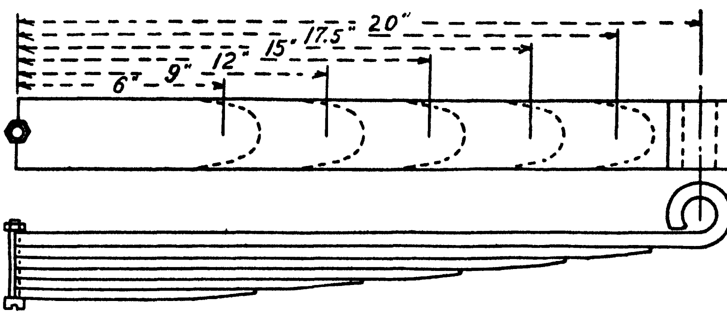


FIG. 12.—FRONT SPRING OF SAMPLE CALCULATION.

leaves, and used in the calculations. Two views of one-half of the spring (without camber) are shown in Fig. 12.

In the assembled spring the stresses in the surface fibers of the different leaves are proportional to the thicknesses of the leaves. As the main leaf is 0.401 in. thick and is subjected to a stress of 145,000 psi by a deflection of 4.75 in., the stresses in the thinner leaves, of 0.360 and 0.291 in. thickness, respectively, will be

$$\frac{0.360}{0.401}\,145{,}000 = 130{,}000 \text{ psi, and}$$

$$\frac{0.291}{0.401}\,145{,}000 = 105{,}000 \text{ psi.}$$

The second and third leaves may be made to the shape they will have in the assembled spring, that is, without nip, and the maximum stress to which they can be subjected therefore will be 130,000 psi. The stresses of the other leaves, however, will be added to by the initial stresses due to nipping. To determine these initial stresses we must first find the nip gaps required, that is, the spaces at the center between adjacent leaves when they are placed together without any pressure being applied.

The nip gap between the main leaf and the one next to it must be made such that when the two are drawn together by the center bolt, and only the main leaf is deflected, the deflection sets up in the latter a stress of 25,000 psi. Under the influence of center-bolt tension the main leaf acts as a double cantilever, each loaded at a distance of 17.5 in. from its support. The equation for the stress in a cantilever beam is

$$S = 1.5\frac{fEt}{l^2},$$

from which it follows that

$$f = \frac{Sl^2}{1.5Et},$$

and in our case

$$f = \frac{25{,}000 \times 17.5^2}{1.5 \times 29{,}000{,}000 \times 0.401} = 0.439 \text{ in.}$$

The equation for the deflection of a cantilever can be transformed to read

$$P = \frac{fEbt^3}{4l^3},$$

and in our case,

$$P = \frac{0.439 \times 29{,}000{,}000 \times 2 \times 0.401^3}{4 \times 17.5^3} = 76.6 \text{ lb.}$$

Of course, the force which must be applied at the middle of the main leaf to close up the nip gap is twice this, or 153.2 lb.

Now, if the second and third leaves are to remain undisturbed and not subjected to any nip stresses, it must require an equal force to close up the nip gaps of the three short leaves. We can so design the leaves that it takes one-third of this force, or 51 lb, to close up each of these gaps; that is, 51 lb on the spring center bolt, which is equal to 25.5 lb at each end of each leaf. It remains to determine what the nip gaps and the initial stresses in the various leaves will be.

We will first consider the case of the shortest leaf, which is 12 in. long and 0.291 in. thick. Under a load of 25.5 lb at each end, this leaf will deflect

$$\frac{4 \times 25.5 \times 6^3}{29{,}000{,}000 \times 2 \times 0.291^3} = 0.0154 \text{ in.}$$

Therefore, the nip gap between the shortest leaf and the one next to it must be 0.0154 in. The stress induced in the leaf by a load of 51 lb applied at its center is

$$S = \frac{6 \times 25.5 \times 6}{2 \times 0.291^2} = 5400 \text{ psi (roughly).}$$

We proceed in a similar manner with respect to the two other gaps. To close up nip gap 2 it is necessary to deflect both the shortest leaf and the one next to it, and the gap must be made such that a pressure of 51 lb applied at the center will close it. In this case, instead of applying the equations of the cantilever, we apply those of the laminated spring. The spring that must be compressed is 18 in. long and consists of two leaves, both of 0.291 in. gauge. For the deflection we have

$$f = \frac{5.7 \times 25.5 \times 9^3}{29{,}000{,}000 \times 2 \times 2 \times 0.291^3} = 0.037 \text{ in.}$$

The additional stress induced in these leaves by closing up this gap is

$$S = \frac{6 \times 25.5 \times 9}{2 \times 2 \times 0.291^2} = 4070 \text{ psi.}$$

In the case of the third nip gap we have for the deflection

$$f = \frac{5.7 \times 25.5 \times 12^3}{29{,}000{,}000 \times 3 \times 2 \times 0.291^3} = 0.0586 \text{ in.,}$$

and for the stress induced

$$S = \frac{6 \times 25.5 \times 12}{3 \times 2 \times 0.291^2} = 3600 \text{ psi.}$$

The total stress in the short leaf therefore is

$$105{,}000 + 5400 + 4070 + 3600 = 118{,}000 \text{ psi;}$$

that in the next leaf,

$$105{,}000 + 4070 + 3600 = 113{,}500 \text{ psi,}$$

and that in the third leaf,

$$105{,}000 + 3600 = 108{,}600 \text{ psi.}$$

The next two leaves, which are not subjected to nip stresses, but are of heavier gauge, are subjected to a maximum stress of 130,000 psi, while the main leaf, in which the load produces a stress of 145,000 psi, is subjected to only 120,000 psi because of the negative nip stress of 25,000 psi. Thus all of the leaves are fairly uniformly stressed under maximum load.

Passenger-Car Rear Spring—As another example we will calculate a rear spring for a medium-size passenger car. We will assume that the spring has to carry a maximum load of 880 lb, under which it deflects 8 in., so that its rate is 110 lb per in. As the wheelbase probably will be about 116 in., we can make the length of the spring 52 in., because lengths of rear springs on passenger cars average 45 per cent of the wheelbase. Springs for cars of this size usually are 2 in. wide, and we will assume this width. The clearance under full load can be made 4 in., so that the maximum deflection of the spring from the unloaded position is 12 in. We will assume a maximum stress in the main leaf of 120,000 psi, and a negative nip stress of 25,000 psi, so that the maximum deflection can produce a stress of 145,000 psi. We then get for the proper thickness of the main leaf

$$t = \frac{145{,}000 \times 26^2}{12 \times 29{,}000{,}000} = 0.267 \text{ in.}$$

We will choose a leaf of 0.262 in. thickness, which is an S.A.E. gauge number. Since it is slightly thinner than the calculated size, the stress in it will be somewhat smaller, and the error therefore is on the safe side. For the summation of cubes of leaf thicknesses we get

$$\Sigma t^3 = \frac{110 \times 2.85 \times 26^3}{2 \times 29{,}000{,}000} = 0.0950 \text{ in.}^3$$

We can build the spring (Fig. 13) up as follows:

No. of Leaves	*Thickness, t*	*Cube of Thickness, t^3*	*Total*
1	0.262	0.0180	0.0180
2	0.237	0.0133	0.0399
5	0.194	0.0073	0.0365
		Grand total	0.0944

This combination will give the desired rate. The nip spaces required and the stresses in individual leaves can be calculated in the same way as in the preceding example.

Springs with Irregularly Stepped Leaves—Springs for heavy trucks and buses sometimes have two or more leaves of full length, and then are stiffer than springs built up of leaves of the same gauges but uniformly stepped as to length, so that the same formulas cannot be applied to them. It is known that if all of the leaves are of full length, the stiffness or rate is substantially 1.5 times as great as if the leaves are uniformly stepped, and as each full-length leaf may be assumed to add equally to the stiffness, the multiplier $(1 + n/2n_t)$ is sometimes applied to the rate for such springs, n being the number of full-length leaves and n_t the total number of leaves. This expression, of course, is equal to 1.5 when all leaves are of full length.

The problem of irregularly stepped leaf springs may be attacked also by the "conjugate-beam" method, which was explained in considerable detail by William Samuels in *Automotive Industries* for April 20, 1935. The conjugate-beam

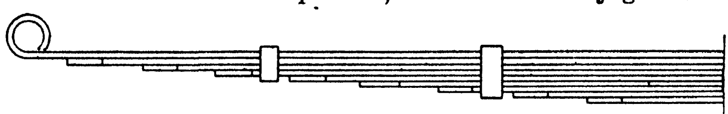


FIG. 13.—ONE-HALF OF REAR SPRING OF SAMPLE CALCULATION.

principle is applied to a cantilever equivalent to one-half of the actual semi-elliptic spring. According to this principle, if a "modified bending-moment diagram" of the actual beam is used as a load diagram for the conjugate beam, then the resulting bending-moment diagram for the latter is also a deflection diagram for the actual beam. The modified bending-moment diagram is a coordinate diagram in which the quotient of the bending moment at any point along the length of the beam by the product EI is plotted over the length of the beam as a base, E being the modulus of elasticity and I the moment of inertia of the beam at the particular point of the length. The conjugate beam is an imaginary beam introduced as an analytical expedient. It is of the same length as the actual beam, but reversed, the fixed end of the former being in line with the free end of the latter. The method is rather involved, and therefore will not be fully explained here; anyone specially interested in the subject is referred to the article mentioned.

Unsymmetrical Springs—Most semi-elliptic springs are secured to the axle at the middle of their length, but some are supported off center, as shown in Fig. 14, and are then known as unsymmetrical springs. If the axle is connected to the frame by the springs alone, the loads on the two spring eyes will be in inverse proportion to the lengths of the two sections. With equal loads on both ends, the deflections of the two sections would be in proportion to the cubes of the lengths, but as the loads are in inverse proportion to the lengths, the deflections are in proportion to the squares of the lengths. If the difference between the lengths of the two sections is relatively small, the spring can be calculated by means of the same equations as used for symmetrical springs, l being taken as equal to one-half the total length. On the other

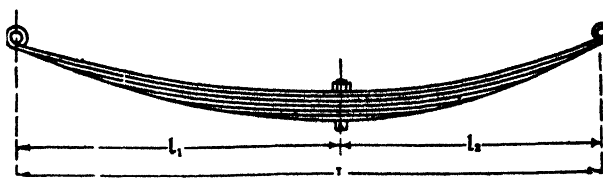


FIG. 14.—UNSYMMETRICAL SEMI-ELLIPTIC SPRING.

hand, if the lengths differ considerably, it is advisable to regard the spring as made up of two quarter-elliptics. The following relations then hold: The loads p_1 and p_2 on the short and the long sections, respectively, are

$$p_1 = \frac{l_2}{l_1 + l_2} P \text{ lb},$$

and

$$p_2 = \frac{l_1}{l_1 + l_2} P \text{ lb}.$$

The deflections of the two sections will be in proportion to the squares of their lengths, and the actual deflection will be

$$f = f_1 + \frac{l_1}{l_1 + l_2}(f_2 - f_1) \text{ in}.$$

The loads on both spring eyes create the same moment at the center of support, and from the load on either eye and the length of its lever arm the value of Σt^3 can be calculated, while the proper thickness of the main leaf can be calculated by means of the equation for the deflection.

Vibration Frequencies of Sprung Masses—When a spring-supported mass is moved from its position of rest by an impulse, it is set vibrating. Suppose that a mass supported by a chassis spring is subjected to a road shock, so that the spring is compressed further. This adds to the energy stored in the spring in the form of fiber stress. As soon as the impulse has ceased, the spring rebounds, and when it passes through the position of equilibrium, the fiber stress—except that due to the static load, which we disregard here—is removed. The energy, however, has not been dissipated, but has been converted into kinetic energy, for when the spring passes through the position of equilibrium, the mass carried by it is moving upward rapidly, and therefore possesses kinetic energy. Except for the small amount of energy dissipated by the damping action during the quarter cycle represented by the first half of the rebound and the potential energy due to elevation of the sprung mass, all of the energy originally stored in the spring by reason of its deflection under the impulse has been converted into kinetic energy.

After the spring has passed the position of equilibrium, its upward speed decreases, hence it gives up kinetic energy, which is converted mainly into potential energy or energy of position, the mass supported by the spring being raised to a higher level against the force of gravity. When the rebound is completed, all of the kinetic energy has disappeared and has been replaced by an equivalent amount of potential energy, except for the frictional loss. Next, as the spring compresses again, this potential energy is again converted into kinetic energy, and so on.

Let f be the deflection of the spring under a given impulse, in ft;

R, the force in lb required to deflect the spring 1 ft;

W, the weight supported by the spring;

n, the number of complete cycles of the spring per minute;

g, the constant of gravity (32.16 fps^2).

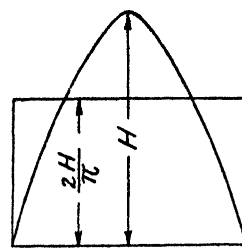
Since it takes R lb to deflect the spring 1 ft, and the actual deflection is f, it follows that the force of the impulse causing the deflection is fR. In deflecting the spring through a distance f, the resistance that must be overcome varies uniformly from nothing at the beginning to fR at the end. The energy expended (or stored up) therefore is $f^2R/2$ ft-lb.

Since the spring passes through n cycles per minute, the duration of one cycle is $1/n$ minute or $60/n$ second. The distance traversed during a complete cycle is $4f$, because f is one-half the motion in one direction. Hence the average velocity of the free end of the spring during the cycle is

$$\frac{4f}{\frac{60}{n}} = \frac{fn}{15}\text{ fps.}$$

The motion of the spring end is harmonic; that is to say, it varies in accordance with a sine curve, Fig. 15, where horizontal distances represent time, and vertical distances, displacements. But since the differential of the sine is equal to the cosine, the velocity of the spring end also can be represented by a sine curve. Now, the maximum ordinate of a sine curve is $\pi/2$ times its mean ordinate, from which it follows that the maximum velocity of the spring end—the velocity at which it passes through the position of equilibrium, where all of its energy is in the form of kinetic energy—is

FIG. 15.—SHOWING RELATION BETWEEN MEAN AND MAXIMUM ORDINATES OF SINE CURVE.



$\pi/2$ times its average velocity. The maximum velocity of the spring end therefore is

$$v = \frac{fn}{15} \times \frac{\pi}{2} = \frac{\pi fn}{30}.$$

Now, the kinetic energy of a spring-supported mass at the moment referred to is given by the well-known equation

$$E = \frac{Wv^2}{2g} \text{ ft-lb.}$$

Substituting the value of v found in the foregoing, we get

$$E = \frac{W\pi^2 f^2 n^2}{g1800} \text{ ft-lb.}$$

As pointed out, this is equal to the energy stored in the spring in the form of fiber stress at the end of the deflection, viz., $f^2R/2$. Equating the two expressions we get

$$\frac{W\pi^2 f^2 n^2}{1800g} = \frac{f^2 R}{2},$$

from which it follows that

$$W\pi^2 n^2 = 900Rg$$

and

$$n = \frac{30}{\pi}\sqrt{\frac{Rg}{W}} \text{ cycles per minute,}$$

where R is in lb per ft of deflection. If R is given in lb per in. of deflection, as is usually the case, the equation becomes

$$n = \frac{30}{\pi}\sqrt{\frac{12Rg}{W}} \text{ cycles per minute.}$$

It will thus be seen that the frequency of vibration of the sprung mass increases with the rate R of the spring and varies inversely as the weight W supported by it. It will also be noticed that the deflection f does not appear in the equation, showing that the rate of vibration is independent of the amplitude. This, however, is so only because we neglect the damping action. Owing to interleaf friction, the frequency of an actual spring varies slightly with the amplitude of vibration.

The above equation for the rate of vibration, or the periodicity, of a spring can be presented also in a simpler form. The rate R of the spring is equal to the quotient of the load on it by the deflection produced by this load under static conditions, viz., W/f, and substituting this for R and inserting the values of the constants, we get

$$n = \sqrt{\frac{35{,}000}{f}} \text{ cycles per minute,}$$

where f is given in inches. This equation gives the following rates of vibration for springs with different deflections under static load:

Defl. (in.)........	2	3	4	5	6	7	8	9	10
Freq. (m^{-1}).......	132.5	108	94	84	76	71	66	62.5	59

The vibratory motion of a motor-vehicle chassis and body, of course, is not a simple harmonic motion in the vertical direction; it is complicated by the fact that the body is supported on two, three or four springs, which receive impulses of different magnitude at different moments; yet, the deflection of the rear springs under static load and the periodicity of the rear end of the vehicle are fair indices of the riding quality of the rear seat.

Resilience of Steel Springs—The amount of energy which can be stored elastically in a unit mass of steel varies with the manner in which the steel is stressed. It evidently is a maximum if the deformation is such that all particles of the steel are equally stressed. This is the case when a steel bar is subjected to tension. Suppose a bar with a cross section of 1 sq in. to be fixed at its upper end and to have a weight applied to its lower end. A weight of one pound attached to the bar will stretch each inch of its length $1/E$ in., where E is the modulus of elasticity. If we assume that the material has a maximum safe working strength of S psi, then a load of S lb can be applied to the bar, and this evidently will stretch it S/E in. While it is being stretched, energy is being stored in the bar in the form of fiber stress, the amount so stored being equal to the product of the resistance overcome by the stretching force, by the distance through which it is overcome. The resistance starts with zero and ends with S; it increases uniformly with the extension, and its mean value therefore is $S/2$. Multiplying this by the extension, S/E, we get for the energy stored elastically, $S^2/2E$ in.-lb per cu in.

Now let us take an ordinary cantilever of uniform section and load it at the free end. From the equations for the deflection of and the stress in such a beam we reach the conclusion that the maximum energy which can be stored in it per unit of volume is $S^2/18\text{E}$ in.-lb per cu in., or only one-ninth as much as in the bar subjected to tension. Much better results are obtained if the cantilever is designed for uniform stress along its length, as by decreasing its width uniformly from the fixed to the free end. In that case the amount of energy that can be stored is $S^2/6E$. In a laminated spring, while the condition of uniform stress distribution along the length is approached, it is not quite attained, and our equations for deflection and stress give an energy-storing capacity of $S^2/6.3E$, the volume being taken as equal to $nlbt/2$.

Coil Springs—Coil springs have the advantage over leaf springs that they can store about twice as much energy per unit of volume, so that a coil spring needs to weigh only about half as much as a leaf spring for the same job. In practice this advantage is offset, however, by the fact that the leaf spring serves not only to cushion the chassis, but also to guide or control its cushioned motion, whereas with coil springs special members, such as radius rods and sway bars, must be provided for this purpose.

In a coil spring under compression, the wire or rod of the coil is subjected chiefly to torsion, and the material works in shear. It was mentioned in the foregoing that in a spring plate the stress varies uniformly from the surface to the neutral fiber, and the average stress is exactly one-half the maximum. When a round wire is subjected to torsion, the stress also varies uniformly from a maximum at the surface to zero at the center, but as there is more material near the surface than near the center, the average stress is equal to two-thirds the maximum, instead of only one-half. Therefore, in the coil spring the material is used to better advantage.

Coil-Spring Formula—The characteristics of coil springs are completely expressed by two equations for the deflection, one of which gives it in terms of the load and the spring dimensions, while the other gives it in terms of the stress and the dimensions. From these two equations others can be derived by mere transformation which give directly the load P in lb, the wire diameter d in in., the coil mean diameter D in in., the number of active coils n, and the stress S_s in the material. The two equations referred to are as follows:

$$f = \frac{8nPD^3}{Gd^4} \text{ in.}$$

$$f = \frac{\pi D^2 n S_s}{Gd} \text{ in.}$$

In these equations G is the modulus of rigidity, which for steels of the grades used in automotive springs is usually about 11,000,000 psi.

The energy stored in the spring is $Pf/2$ in.-lb, and the volume of steel in the spring is $\pi^2 d^2 Dn/4$ cu in. By substituting for P and f their equivalents in terms of stress and spring dimensions found from the above two equations, and dividing the result by the expression for the volume of steel in the spring, we get for the energy stored per unit of volume, $S_s^2/(4G)$ in.-lb per cu in. This, as already stated, is about twice as much as the amount stored in a cu in. of leaf-spring steel, viz., $S^2/(6E)$, if the maximum permissible values are used for S_s and S. The greater part of the gain is due to the fact that in the coil spring the average stress is considerably more than one-half the maximum (and the energy stored varies as the square of the stress). The low modulus of elasticity also helps, but it is offset to a considerable extent by the fact that the maximum permissible shearing stress S_s is substantially lower than the maximum permissible tensile stress S. It has long been customary to figure with an elastic limit in shear of about 0.6 the elastic limit in tension, but the shearing stresses allowed in automotive-type coil springs are about 0.7 times the tensile stresses figured with in leaf springs of the same material. This is permissible because the coil springs are made with great care. The rods from which the springs are wound are precision-rolled or ground in a centerless grinder, so they should be practically devoid of surface defects. Moreover, the surface material is prestressed by shot-peening and cold-setting. Stresses up to 70,000 psi are allowed in coil springs under static load, and up to 100,000, and even a little more, under extreme shock load.

The foregoing data enable us to make a comparison of the weights of steel required for springs of different types for a given service. Let us consider the rear springs of a passenger car which have to support, say, a maximum load of 1750 lb and withstand a maximum deflection of 12 in. The energy stored by the springs then will be

$$(1750 \times 12)/2 = 10{,}500 \text{ in.-lb.}$$

With carbon steel we can figure with a maximum tensile stress of 100,000 psi; with alloy steel with a maximum tensile stress of 140,000 psi and a maximum shearing stress of 100,000 psi. In calculating the weights of steel required we will use the theoretical formula, which neglects the inactive material. The energy stored per cu in. is $S^2/(6E)$ in the case of leaf springs, and $S_s^2/(4G)$ in the case of coil springs. As one cu in. of steel weighs 0.281 lb, the amounts of energy stored per pound become $S^2/(1.686E)$ and $S_s^2/(1.124G)$. As in our case the total amount of energy to be stored is 10,500 in.-lb, we have for the weights of steel required with the three different types of spring:

Carbon-steel leaf springs:

$$\frac{10{,}500}{\dfrac{100{,}000^2}{1.686 \times 29{,}000{,}000}} = 51 \text{ lb.}$$

Alloy-steel leaf springs:

$$\frac{10{,}500}{\dfrac{140{,}000^2}{1.686 \times 29{,}000{,}000}} = 26 \text{ lb.}$$

Alloy-steel coil springs:

$$\frac{10{,}500}{\dfrac{100{,}000^2}{1.124 \times 11{,}000{,}000}} = 13 \text{ lb.}$$

These are the weights of the active portions of the two rear springs combined. The actual weight of the springs would be from 10 to 20 per cent higher, due to the inactive material.

Fig. 16 shows a typical coil suspension spring. It is wound of rod of 0.527 in. finished diameter, and has $10\frac{1}{2}$ active coils of a mean diameter of $4\frac{7}{8}$ in. The spring is designed to carry a maximum static load of 800 lb, and to have a rate of 90 lb per in. According to the formula, the deflection under the maximum static load will be

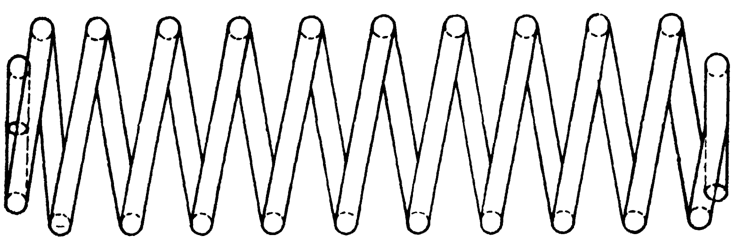


Fig. 16.—Coil-Type Vehicle-Suspension Spring.

$$f = \frac{8 \times 10.5 \times 800 \times (4\frac{7}{8})^3}{11{,}000{,}000 \times 0.527^4} = 9.2 \text{ in.}$$

As the free length of the spring is 20.5 in., its length under full static load will be

$$20.5 - 9.2 = 11.3 \text{ in.}$$

If a maximum shearing stress of 100,000 psi is allowable in the material, the maximum deflection can be

$$f = \frac{3.14 \times 10.5 \times (4\frac{7}{8})^2 \times 100{,}000}{11{,}000{,}000 \times 0.527} = 13.2 \text{ in.}$$

Hence we can allow a clearance of $13.2 - 9.2 = 4$ in.

At each end of the coil spring there is a dead coil, ground off flat, by which the spring is clamped to its seat, or rests on its seat. This coil is of reduced diameter and joins the active coils through an intermediate section having the normal pitch and a gradually increasing radius. The lower ends of coil springs usually are clamped to their seats by clamp bars and bolts.

Torsion-Bar Springs—The torsion-bar spring, as already explained, consists of a bar of spring steel which is anchored to the frame at one end and freely supported thereon at the other, where it connects to a lever arm that is pin-jointed or linked to the steering head or axle spindle. As a rule, the lever has a short shaft or spindle which is mounted in a bearing on the frame, and the torsion bar is connected to this shaft in such a manner that it is subjected to torsional stresses only.

Torsion bars can be used both with and without rigid axles. Fig. 17 is a sketch of the original torsion-bar suspension, due to Dr. Ferdinand Porsche. There the torsion bars of the front suspension extend through a tubular cross member of the

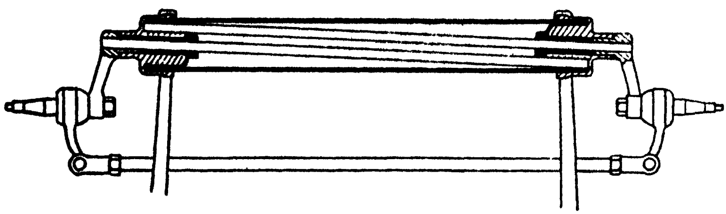


FIG. 17.—SKETCH OF PORSCHE TORSION-BAR SUSPENSION.

frame. Each bar is anchored in a bracket at one end of the cross tube, and splined in the hub of a lever arm mounted in needle bearings in the bracket at the other end. The free end of the lever arm connects to the steering spindle by means of a ball-and-socket joint. While it is not shown in the sketch, the steering spindle is supported also by another arm extending from the frame bracket parallel with the one shown.

Torsion-bar suspensions saw considerable use in Europe prior to the outbreak of World War II, especially on small, low-priced cars. Among other well-known makes, they were used on the Volkswagen or popular car in Germany, the front-driven Citroën and the Mathis in France, and on Lagonda and Vauxhall models in England. In most designs the torsion bars extended lengthwise of the frame, and at their free ends connected to lever arms which were pin-jointed to the steering heads. In this country a torsion-bar suspension for both passenger and commercial vehicles was developed by A. F. Hickman of the Truck Equipment Company, Buffalo, N. Y. As shown in Fig. 18, the torsion bars (for the front suspension) extend parallel with the frame side rails, immediately outside thereof, and at their forward ends carry lever arms which connect by short links to the ends of arms sweeping up from the ends of the axle center. The rear suspension is designed along similar lines, the arms on the torsion bars being linked to brackets on the axle housing. As the chassis is hung on four links, it has considerable lateral freedom.

Theory of Torsion-Bar Suspension—A diagram of a torsion-bar suspension in which the bar is supposed to extend

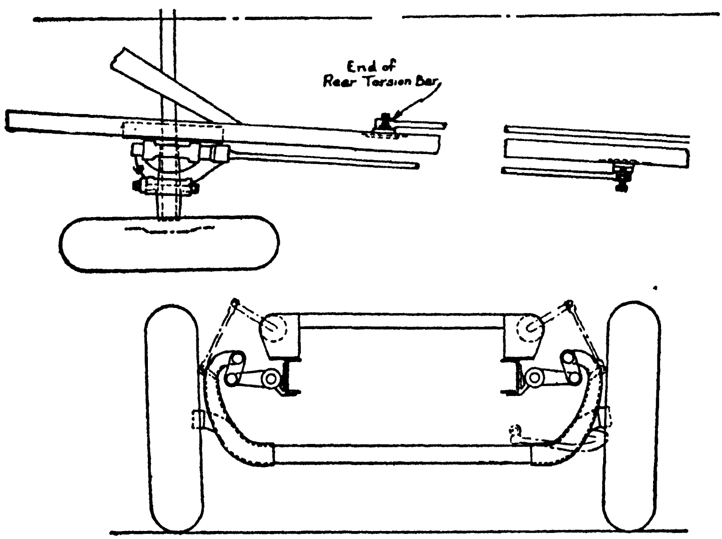


FIG. 18.—TWO VIEWS OF HICKMAN TORSION-BAR SUSPENSION.

in the direction of vehicle length is shown in Fig. 19. The equation for the deflection of a cylindrical shaft when subjected to a torsional moment is

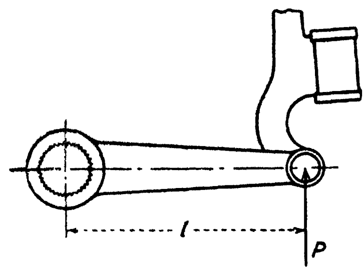
$$\theta = \frac{TL}{GJ} \text{ radians},$$

where T is the moment or torque in lb-in.; L, the length of the shaft or bar, in in.; G, the modulus of rigidity (11,000,000 for steel), and J, the polar moment of inertia of the section of the bar. With the lever arm on the bar extending in the horizontal direction, as in Fig. 19, the moment T evidently is equal to Pl. Remembering that the polar moment of inertia of a circular section is $\pi d^4/32$, we have

$$\theta = \frac{32TL}{\pi d^4 G} \text{ radians} = \frac{TL}{18{,}900 d^4} \text{ deg.}$$

When the arm on the torsion bar is horizontal, the up-and-down motion of the chassis is equal to the movement of the free end of the torsion-bar arm around its fulcrum, and the reaction of the steering head or wheel spindle on the arm then is at right angles to the center line of the latter. On the other hand, when the arm is inclined upwardly and makes a considerable angle with the horizontal, the increase in deflection with increase in load is materially smaller.

FIG. 19.—DIAGRAM OF TORSION-BAR SUSPENSION WITH LEVER ARM IN HORIZONTAL POSITION.



In the case of torsion-bar suspensions it is advisable to calculate the necessary dimensions from the maximum dynamic load (the bump load), and the stress which can be allowed under this load. In passenger cars with constant-rate springs the clearance under maximum static load is so proportioned, on the average, that the maximum bump load is about 165 per cent the maximum static load. With torsion-bar springs, since the load rate increases quite rapidly as the point of maximum deflection is being approached, we may well allow for a somewhat greater bump load, say 200 per cent the maximum static load.

The stress produced in a torsion bar by a moment or torque T is given by the equation

$$S = \frac{16T}{\pi d^3} \text{ psi,}$$

and if S_m and T_m are the stress and torque under maximum bump load, the bar diameter must be

$$d = \sqrt[3]{\frac{16T_m}{\pi S_m}} \text{ in.}$$

As the torsion bar is a simple member which can be hardened quite uniformly, and which also lends itself readily to methods of prestressing, relatively high maximum stresses can be allowed, viz., from 110,000 to 140,000 psi, the latter value being applicable where the material is suitably prestressed. Buick allows a maximum stress of 140,000 psi in torsion bars for heavy military vehicles, which are prestressed to 30,000 psi.

The stress in the torsion bar can be given also in terms of the deflection:

$$S = \frac{G\theta d}{2L} \text{ psi},$$

where θ is the deflection in radians, and L the length of the active part of the torsion bar, in in. If θ is given in degrees and G is replaced by its value, the equation becomes

$$S = \frac{11{,}000{,}000\theta d}{(2 \times 57.3)L} = \frac{96{,}000\theta d}{L} \text{ psi}.$$

From this it follows that the active length of the bar must be

$$L = \frac{96{,}000\theta_m d}{S_m} \text{ in.},$$

where θ_m is the maximum deflection in deg.

Variation of Load Rate—It has been pointed out already that the load rate of a torsion-bar suspension is variable, and this characteristic of the device will now be further discussed. Fig. 20 is a sketch of a torsion-bar lever extending upward at an angle α. The linear deflection x above the horizontal through the lever-arm fulcrum is equal to $l \sin \alpha$, and the effective lever arm y is equal to $l \cos \alpha$.

Two equations for the stress in the bar were given in the foregoing, one in terms of the torque and the other in terms of the arc of deflection. By equating the two expressions we get

$$\frac{16T}{\pi d^3} = \frac{G\theta d}{2L},$$

from which it follows that

$$T = \left(\frac{\pi d^4 G}{32L}\right) \theta \text{ lb-in.}$$

If we denote the expression in parentheses by c, $T = c\theta$. By now referring to Fig. 20, it can be seen that when the lever arm makes an angle with the horizontal, $T = Py$. The linear spring rate or load rate is equal to dP/dx. By differentiating the simple equation for T and transforming we get

$$c = \frac{dT}{d\theta} = y\frac{dP}{d\theta} + P\frac{dy}{d\theta}$$

$$= y\frac{dP}{dx} \cdot \frac{dx}{d\theta} + P\frac{dy}{d\theta}.$$

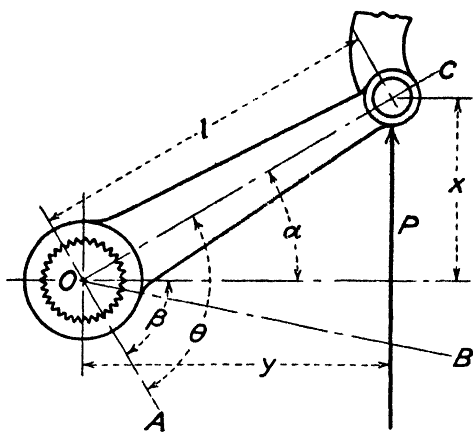


FIG. 20.—DIAGRAM OF TORSION-BAR SUSPENSION WITH LEVER ARM INCLINED.

Now, the differential or increment $d\theta$ of the angle of total deflection is equal to the differential or increment $d\alpha$ of the angle made by the lever arm with the horizontal, and we may therefore substitute the latter value for the former, which gives

$$c = y\frac{dP}{dx} \cdot \frac{dx}{d\alpha} + P\frac{dy}{d\alpha}.$$

Also,

$$dx/d\alpha = l \cos \alpha = y,$$

and

$$dy/d\alpha = -l \sin \alpha = -x.$$

Making these substitutions we get

$$c = y^2\frac{dP}{dx} - Px,$$

and transforming,

$$\frac{dP}{dx}(= R) = \frac{c + Px}{y^2} = \frac{c + Px}{l^2 - x^2}.$$

In the case of bars of rectangular or other non-circular sections, the value of c in the above equation can be determined experimentally. For circular sections, $c = \pi d^4G/32L$, and making this substitution and simplifying,

$$R = \frac{\pi d^4G + 32LPx}{32L(l^2 - x^2)} \text{ lb per in.}$$

Torsion-Bar Calculation—In the case of a torsion-bar suspension, what is really most important is the rate under normal static load, which should be approximately equal to the minimum rate of the spring. An expression for the minimum rate might be obtained by differentiating the above expression for the rate R and equating the first differential coefficient to zero. Unfortunately, this does not give a simple, usable expression, and in calculating torsion bars it is therefore necessary to assume a value for the maximum deflection of the bar, and from this calculate the necessary active length. After that has been done, the rate R at any load P and any deflection x can be calculated by means of the above equation. The load P is a function of x, and must first be calculated for a number of different values of x, preferably approximately equally spaced. In the equation for R, x has a positive value when the lever arm is inclined upward from the horizontal, and a negative value when inclined downward. After a series of values of P and R have thus been found, they can be plotted against the deflection x. If the load rate at normal static load thus arrived at does not appear to be suitable, the maximum deflection θ can be given a different value, and the calculation carried through once more. Other things being equal, the greater the maximum deflection assumed, the lower will the minimum rate. We will illustrate the method of procedure by working out an example.

We will assume that a torsion-bar suspension is to be designed to support a maximum static load of 800 lb at the end of a lever arm 10 in. long. By making the lever arm fairly long, the maximum angular deflection can be kept down, which is desirable because it will minimize the effect of spring action on the tread if the arms extend transversely, and on wheel accelerations and decelerations if the arms extend longitudinally. As already mentioned, the maximum bump load may be set down as equal to twice the maximum static load, so that in this case it is 1600 lb. We may further assume that the deflection of the lever above the horizontal is limited to 30°, and that the total angle of deflection then is 90°. With the torsion bar in the free state the lever arm will then extend downward at an angle of 60°. This is the zero position, and the torque and the stress increase uniformly with the deflection from this position.

Under maximum bump load, when the lever extends upward at 30° from the horizontal, the effective length y of the lever arm is

$$10 \times \cos 30° = 10 \times 0.866 = 8.66 \text{ in.}$$

As the load with the spring in this position has been assumed to be 1600 lb, the value of the torque is

$$T = 1600 \times 8.66 = 13{,}850 \text{ lb-in.}$$

As this torque deflects the bar 90°, the angular rate is 13,850/90 = 154 lb-in. per deg.

If we set down the maximum stress in the bar at 120,000 psi, the required diameter is

$$d = \sqrt[3]{\frac{16 \times 13{,}850}{3.14 \times 120{,}000}} = 0.838 \text{ in.}$$

By now inserting values in the equation for the required length we get

$$L = \frac{96{,}000 \times 90 \times 0.838}{120{,}000} = 60.3 \text{ in.}$$

Under maximum bump load the deflection above the horizontal is

$$x = 10 \times \sin 30° = 10 \times 0.5 = 5 \text{ in.}$$

We now have the values of the load P (1600 lb) and the deflection x (5 in.) for the position of maximum deflection, and we are therefore in position to insert in the equation for the rate R. But before doing this we can simplify that equation for this particular case:

$$\frac{3.14 \times 0.838^4 \times 11{,}000{,}000 + 32 \times 60.3Px}{32 \times 60.3(100 - x^2)} = \frac{8830 + Px}{l^2 - x^2}.$$

By now inserting 1600 for P and 5 for x, and carrying through the calculation, we get $R = 224$ lb per in. The load and rate for any other deflection can be calculated in the same way. Calculations were carried through for a number of deflections ranging from 5 in. to −5 in., and the values obtained in the individual steps are given in Table III. From the values of P and R thus found the chart Fig. 21 was drawn. In Fig. 20

the line OA represents the position of the lever in the free or unloaded state; OB, its position under maximum static load (800 lb), and OC, its position under maximum bump load (1600 lb).

Table III—Calculation of Loads and Rates of a Torsion-Bar Spring

x..........	5	3	1	0	-1	-3	-5
$\sin\alpha$.......	0.5	0.3	0.1	0	-0.1	-0.3	-0.5
α..........	30°	17.5°	5.75°	0	$-5.75°$	$-17.5°$	$-30°$
θ..........	90°	77.5°	65.75°	60°	54.25°	42.5°	30°
y..........	8.66	9.54	9.95	10	9.95	9.54	8.66
T..........	13,850	11,930	10,130	9,240	8,340	6,540	4,620
P..........	1,600	1,250	1,020	924	838	687	533
$(100 - x^2)$..	75	91	99	100	99	91	75
Px.........	8,000	3,750	1,020	0	-838	$-2,061$	$-2,665$
R..........	224	138	99.5	88.3	80.5	74.2	82

The various steps in calculating P and R for any deflection x may be summarized as follows:

The angle α corresponding to any value of x or $-x$ is

$$\alpha = \arcsin x/l \text{ deg.}$$

The angular rate is

$$R_a = P_m/\theta_m \text{ lb per deg.}$$

The wind-up angle

$$\theta = \alpha + \beta,$$

where β is the wind-up angle with the lever arm horizontal. The moment on the torsion bar

$$T = R_a\theta = R_a(\alpha + \beta) \text{ lb-in.}$$

The load

$$P = T/y = T/(l \cos \alpha) = R_a(\alpha + \beta)/(l \cos \alpha) \text{ lb,}$$

and the rate (as derived in the foregoing),

$$R = \frac{\pi d^4 G + 32LPx}{32L(l^2 - x^2)} \text{ lb per in.}$$

Prestressing Methods—Prestressing can be effected by shot-peening and by presetting under a load—applied in the same direction as the service load—which will stress the material beyond its elastic limit. The effects of these two processes are dissimilar, for whereas shot-peening increases the strength of the material under torsional loads regardless of the direction of the latter, the beneficial effects of presetting are limited to the direction in which the presetting load is applied, and a torsion bar so treated cannot be successfully used under loads in the opposite direction. If both processes are applied, the peening operation must precede the presetting.

Design of Ends—The ends of torsion bars are upset to an outside diameter about 35 per cent larger than that of the

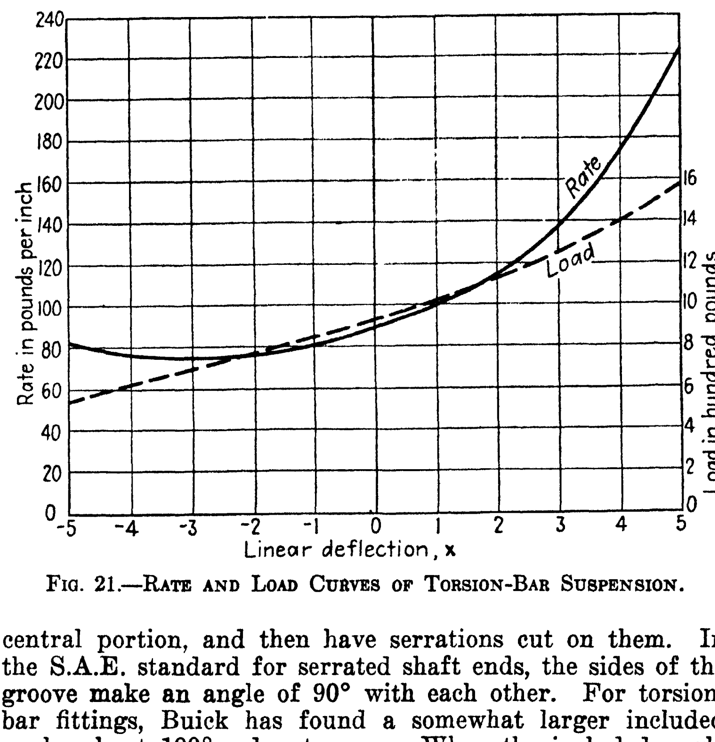


FIG. 21.—RATE AND LOAD CURVES OF TORSION-BAR SUSPENSION.

central portion, and then have serrations cut on them. In the S.A.E. standard for serrated shaft ends, the sides of the groove make an angle of 90° with each other. For torsion-bar fittings, Buick has found a somewhat larger included angle, about 100°, advantageous. When the included angle is too large there is danger of the female member of the joint being subjected to excessive disruptive forces, whereas if it is too small, the load will be rather unevenly distributed along the length of the serrations. A relatively large radius is used at the root of the serrations, to prevent excessive stress concentration. The upset ends connect to the central portions of the bar by tapering sections with about 30° included angle, and the transition from the conical to the cylindrical section is made with a large radius.

With a torsion-bar suspension it is comparatively easy to adjust the clearance of the spring under normal load. This can be done by providing the so-called fixed end of the bar with a lever, from the free end of which a link with a turnbuckle connects to a fixed point on the frame.

Torsion-bar springs are relatively new, and in the United States they have been used to only a limited extent so far, but there are good reasons for believing that they will come into wide use in the future. Each spring consists of a single unit of relatively simple form, hence the manufacturing costs should be low, except for the cutting of serrations on the ends and in the fittings. Simpler forms of attachment could be used, at the cost of some additional weight. The desirable feature of a rate increasing with the deflection is readily obtainable, and is, in fact, inherent in the design; and, finally, the modulus of resilience (the specific energy-storing capacity) of the torsion bar is greater than that of any spring type now in common use.

One would expect even higher "elastic efficiency" from torsion tubes than from torsion bars, for whereas in a solid bar the mean stress in the material, within the elastic range, is equal to two-thirds the maximum, with a tube having an inside diameter equal to, say, three-fourths the outside diameter, the mean stress would be equal to 88 per cent the maximum. It is questionable, however, whether in relatively long tubes the wall thickness could be held to sufficiently close tolerances to make it possible to realize this theoretical advantage, and so far experience with torsion tubes in suspension systems does not seem to have been entirely satisfactory.

Volute Springs—Volute springs have been used extensively as railroad buffer springs abroad, and in recent years as suspension springs on military vehicles that must travel off the highways. They have seen some use also as helper springs on heavy commercial vehicles. The conventional volute spring is wound of strip steel on a cylindrical mandrel to a constant pitch angle. Usually the width of the strip is at least ten times its thickness. Some volute springs are wound hot under pressure, in which case adjacent coils are in contact with each other and absorb energy by friction when there is spring action. For automotive suspensions, however, the springs are so wound that there is clearance between the coils. End portions of the strip equal in length to three-fourths of the end coils are cut off at an angle equal to the pitch angle, so that when the volute is wound they will be at right angles to the axis and form the seating portions. The portions of the blade which are cut off at an angle and rest on the seats are inactive. Fig. 22 is a developed view of a volute spring.

A volute spring naturally acts a good deal like a coil spring, but it differs from the latter in that its individual coils are of unequal stiffness. From the fact that in a coil spring the rate or stiffness varies inversely as the cube of the coil diameter, it is obvious that in a volute spring the coil at the large end is much softer than the one at the small end. Therefore, when load is applied, deflection at first occurs mainly in the large coils, and the material in these coils is stressed more than that in the coils at the small end. The stress in the largest coil is limited by "bottoming" of that coil. Up to the point where this coil "bottoms," the spring has a "straight-line" characteristic; that is, the deflection is directly proportional to the applied load. When the first coil bottoms, the spring becomes materially stiffer, its rate increases, and the stress in the remaining coils increases more rapidly with further increase in load. The different coils bottom successively, until finally the spring is compressed solid. If a volute spring is to be used as a suspension spring, it must be so designed that the static load will not quite "bottom" the largest coil, for as soon as any coil has "bottomed," it is useless for cushioning purposes.

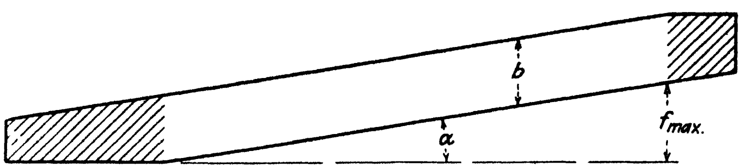


FIG. 22.—DEVELOPMENT OF VOLUTE SPRING.

The material of volute springs, the same as that of coil springs, is subjected principally to torsion. A rectangular section, however, is not so well adapted as a round section to sustain torsional stresses, for whereas with a round section the stress is uniform all around the circumference, with a rectanglar section it is a maximum at or near the middle of the sides, and a minimum at the edges. With sections such as used in volute springs, the stress is especially high near the middle of the long sides, and failures of volute springs usually originate in cracks at these positions, rather than at the edges.

In the manufacture of volute springs, the end of the strip which will form the inactive section of the small end is tapered down in thickness, partly because this facilitates coiling, and partly because it reduces the maximum stress. Volute springs sometimes are wound to a free height somewhat greater than that wanted in the completed spring, and the height is then reduced by cold-setting or bulldozing, in which process the spring is stressed beyond its elastic limit. As with other spring types, this has the effect of improving the mechanical properties by "cold-working," and of preventing or reducing "set" in service. Some difficulty is encountered in hardening volute springs, owing to the small space between coils. To obtain a high and uniform hardness, it is necessary that in the quenching operation the heat be carried off rapidly from the entire surface, and the springs therefore are quenched in an oil bath over a submerged jet.

Two Basic Types—The conventional volute spring is wound on a cylindrical mandrel, its seating edge being held at right angles to the mandrel, which gives a constant pitch angle or helix angle. In such a spring, however, the material is not very efficiently utilized, for when it is compressed solid, and therefore contains its greatest store of elastic energy, the material is stressed non-uniformly, the stress being much greater at the small than at the large end. To use the material to better advantage, the spring must be so designed that when it is compressed solid, the stress will be uniform throughout the active length of the blade. This necessitates varying the pitch angle in such a manner that it is directly proportional to the coil radius. Springs of this type are said to have a constant bottoming stress. The variable pitch angle can be produced either by winding the coil on a taper mandrel, or by winding it on a cylindrical mandrel and then subjecting it to a presetting or bulldozing operation. In this operation the spring is subjected to an axial load that sets up stresses exceeding the elastic limit of the material. The smallest coil reaches the elastic limit first, and the bulldozing or presetting operation is continued until all of the coils are affected, and the pitch angle varies uniformly from the large to the small end. Formerly volute springs were preset against a plane surface, the operation being repeated until proportionality between pitch angle and coil diameter was achieved, or at least closely approached. In this operation the maximum stress in the material was limited to that produced in service when the spring is closed up solid. At present these springs are preset in a bowl, and the presetting stress—which is always greater than the maximum which can be reached in service—can be controlled by varying the depth of the bowl.

Volute-Spring Formulae—Performance characteristics, of course, are different for the two principal types of volute—the one with constant pitch angle, and that with uniform bottoming stress. The following formulae for the two types of volute springs are taken from a *Manual on the Design and Manufacture of Volute Springs*, by the S.A.E. War Engineering Board, published by the Society of Automotive Engineers. In these equations P_t is the transition load (the load at which bottoming begins); P_s, the load closing the spring up solid; f_t, the deflection at which bottoming begins; h, the total deflection; b, the blade width; t, the blade thickness; p, the radial pitch; D, the outside diameter; d, the inside diameter; R, the largest free-coil radius (usually $\frac{1}{2}D - \frac{1}{2}p$); r, the smallest free-coil radius (usually $\frac{1}{2}d + \frac{3}{4}t$); S, the maximum stress in the spring aside from secondary and residual stresses.

Taking first the volute with constant pitch angle, we have for the rate within the "straight-line" range of deflections,

$$R = \frac{P_t}{f_t} = \frac{2{,}150{,}000pbt^3}{R^4 - r^4} \text{ lb per in.}$$

The maximum bottoming stress is

$$S = \frac{3{,}500{,}000tph}{rR^2 - r^3} \text{ psi.}$$

The load which will close the spring up solid,

$$P_s = \frac{0.32bt^2S}{r} \text{ lb,}$$

and the load at the transition point (where bottoming begins),

$$P_t = \frac{0.31brt^2S}{R^2} \text{ lb.}$$

For volute springs with uniform bottoming stress the rate within the straight-line range of deflections is given by the same equation as that for the constant pitch-angle spring. The maximum bottoming stress for this type of spring is

$$S = \frac{5{,}250{,}000tph}{R^3 - r^3} \text{ psi.}$$

The load which will close the spring up solid is given by the same equation as that for the constant pitch-angle spring, and the load at the transition point is

$$P_t = \frac{0.31bt^2S}{R} \text{ lb.}$$

The above equations apply to springs made of ordinary spring steel having a modulus of rigidity of approximately 11,000,000 psi.

Properties of the two principal types of volute springs are compared graphically in Fig. 23, which was redrawn from an illustration in the S.A.E. War Engineering Board's *Manual on the Design and Manufacture of Volute Springs.* Load-deflection curves are given for three different springs, all containing the same amount of active steel in the same space, and all subjected to the same maximum stress. In each of

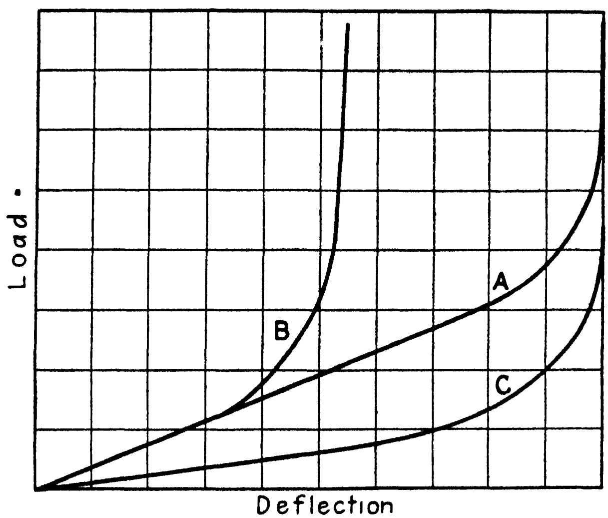


FIG. 23.—LOAD-DEFLECTION CURVES OF THREE VOLUTE SPRINGS.

these springs the radius r at the beginning of the smallest active coil is equal to $0.4R$, R being the radius at the end of the largest active coil. Spring A is of the type in which the bottoming stress is constant throughout, while springs B and C have constant pitch angles. Springs A and B have the same rate within their respective "straight-line" ranges, and the same load is required to compress them solid. Springs A and C have the same total deflection. When the springs are compressed solid, the amount of energy stored in them—which is measured by the area included between the base line and the load-deflection curve—is 2.8 times as great in spring A as in springs B and C.

Fig. 24 shows a volute spring applied to a motor bus as a helper spring. It is installed in an inverted position, the small end resting on the pressure plate of the leaf spring, on which it is centered by means of a dowel, while the large end bears against a seat formed on a frame bracket. This particular spring is wound of strip of tapering width, so that the small coils are of greater depth than the large ones. This would have the effect of increasing the rate with deflection even more rapidly than in a volute wound of strip of uniform width, but probably is intended chiefly to reduce the stress in the small coils.

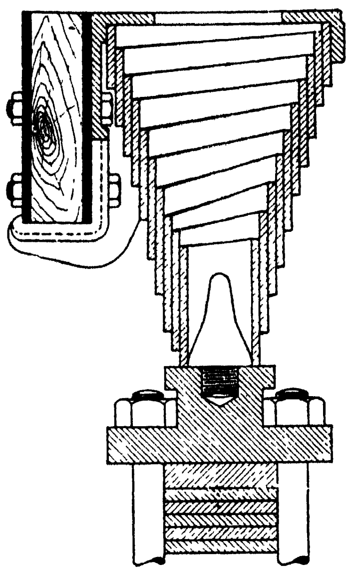


FIG. 24.—VOLUTE HELPER SPRING FOR MOTOR BUS.

Volute springs possess the feature of a variable rate, which is an advantage in certain applications. They also permit of storing a large amount of energy in a restricted space, which may specially suit them to certain designs. On the other hand, owing to the non-uniform stress distribution over the cross section of the blade, and in some designs also over its length, the elastic efficiency of the volute is not very high. From figures published by the S.A.E. War Engineering Board regarding the energy-storing capacity of different types of springs, it would appear that for a given service a volute spring would have to weigh about 20 per cent more than a coil spring, and about 67 per cent more than a torsion-bar spring. The comparison is based on total weights, including inactive material.

CHAPTER XVII

Conventional Suspension Systems

Factors Affecting Passenger Comfort—That vibration causes discomfort is well known, but there has been some uncertainty as to the degree in which each of the two elements of vibratory motion, the frequency and the amplitude, contribute to it. Steady or uniform motion does not cause discomfort, nor does gentle acceleration, but severe acceleration is decidedly unpleasant. One is inclined to assume that the acceleration is a measure of the discomfort, and that, since in vibratory motion the maximum acceleration is directly proportional to both the amplitude and the frequency, discomfort is equally dependent on both of these factors.

Experimental investigation, however, has shown that the frequency is a much more potent factor than the amplitude. The problem was studied at General Motors Proving Grounds, by means of a device known as a bouncing table. It comprises a seat which can have a simple harmonic, vertical motion imparted to it, of any desired frequency and amplitude. The "subject" is seated on this device, the stroke or amplitude is set at any given value, and the frequency is gradually increased until the subject begins to feel uncomfortable. Tests were conducted with numerous individuals, and the results are said to have agreed remarkably well. From these results the so-called "comfort curve," Fig. 1, was drawn, which shows the combinations of amplitude and frequency at which discomfort begins. The "comfort curve" corresponds approximately to the equation

$$a\omega^{2.7} = 324{,}000,$$

where a is the amplitude or displacement in in., and ω the frequency of vibration in cycles per minute.

Important pioneer work on the factors affecting riding comfort was done at Purdue University by Prof. H. M. Jacklin and G. J. Liddell. In Research Bulletin No. 44 of the Purdue Engineering Experiment Station they published a chart containing (among others) a curve of frequency-maximum acceleration combinations at which vertical or up-and-down vibration becomes "disturbing." This curve shows that if the frequency is increased from 3 to 6 cycles per second, the maximum acceleration must be decreased from 5 to 0.8 ft per sec^2, thus confirming the preponderance of frequency as a factor in causing discomfort.

Nature of Chassis Vibration—Up to the time when independent suspension was introduced, in the early thirties, the front springs of motor vehicles always were made much stiffer than the rear springs. In the course of automotive development, rear springs were gradually made more flexible, to

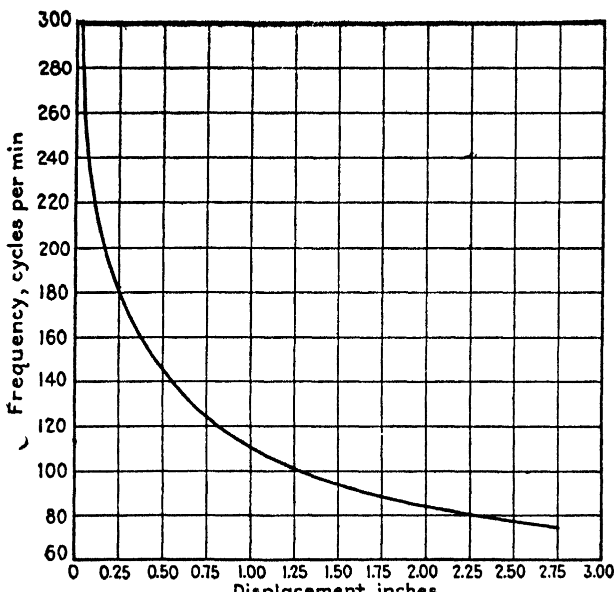


FIG. 1.—"COMFORT CURVE."

improve the ride. Softening of the front springs would have had a similar effect, but front springs had to be made relatively stiff, because they served not only to absorb shocks, but also to control the motion of the front axle and wheels relative to the car frame, and if they were made unduly soft, the steering became erratic. As long as the front springs were the only axle-control members, their static deflection could not well be made more than about one-third that of the rear springs. With springs so proportioned, any increase in spring load due to road shock naturally results in a much smaller change in the level of the front than in that of the rear end of the chassis, and the front end, moreover, has a higher rate of natural vibration. It is quite plain that with a vehicle so sprung, both ends of the chassis cannot rise and fall in synchronism, through equal distances, which would give what is now often referred to as a "level ride." Analysis shows that under these conditions the actual motion of the chassis is the resultant of two angular oscillatory motions around conjugate transverse axes, one within the wheelbase, the other some distance forward of the front axle. The two oscillatory motions have different amplitudes and different frequencies.

When a vehicle is being driven over the road, the oscillations of its sprung mass have frequencies which are dependent, not on the frequency at which road impulses or bumps are encountered, but on the relation between the spring stiffness and the mass of the sprung part of the vehicle. It was shown in a previous chapter that when a mass suspended by a spring is set vibrating, its frequency of vibration varies directly as the square root of the spring rate or stiffness, and inversely as the square root of the mass. This applies to linear vibration. In the case of angular vibration, or oscillation around an axis, the frequency varies directly as the square root of the angular rate, that is, the moment of the spring force around the axis which produces a unit angular deflection, and inversely as the square root of the moment of inertia of the mass around the axis.

Location of Axes of Oscillation—The exact locations of the conjugate axes around which the sprung mass oscillates depend on the ratio between the static deflections of the front and rear springs, and on the moment of inertia of the vibrating mass around a transverse axis through its center of gravity. The moment of inertia is the summation of the products of all mass particles by the squares of their distances from the axis through the center of gravity. It is equal also to the product of the mass of the vibrating body by the square of its radius of gyration, k. The radius of gyration, therefore, is that distance from the axis which, if all of the mass were located there, would give the same moment of inertia as that of the actual body.

Provided the front springs are stiffer than the rear, the axis of oscillation within the wheelbase, which may be called the axis of pitch, is always located between the center of gravity and the rear axle, and if the front springs are much stiffer, it is relatively close to the rear axle. Under the same conditions, the axis outside the wheelbase, which may be called the axis of bounce, is always forward of the front axle. It is relatively close to the axle when the front springs are much stiffer than the rear, and it moves forward as the stiffness of the front springs decreases and that of the rear ones increases. When both pairs have the same static deflection, it is located at an infinite distance. The pitch axis is then at the center of gravity, and both ends of the chassis have a parallel motion.

However, as already pointed out, the locations of the axes depend also on the moment of inertia of the sprung mass. If k^2, the square of the radius of gyration, is equal to the product ab, where a and b are the distances from the center of gravity of the sprung mass to the axes of the front and rear axles, then the axis of pitch is directly over the rear-axle axis, and that of bounce, directly over the front-axle axis. It is obvious that in that case, if an impulse were given to the sprung mass through either the front or the rear axle, it would not produce motion around the axis located there. Therefore, if the axis of pitch is directly over the rear axle, an impulse to the rear wheels will produce pure harmonic motion around the axis of bounce, over the front axle, and an impulse to the front wheels will produce pure harmonic motion around the axis of pitch, over the rear axle. In practice k^2 generally is somewhat less than ab, because it is difficult to locate enough of the sprung mass ahead of the front and back of the rear axle to make k^2 equal to ab. The axis of pitch is then a short distance ahead of the rear axle, and impulses to the rear wheels will set up oscillations around both axes, but mainly around the axis of bounce, near the front axle; similarly, impulses to the front wheels also will set up oscillations around both axes, but chiefly around the axis of pitch, near the rear axle. How the axes of pitch and bounce move with changes in the ratio between the rates of front and rear springs, and with the relative length of the radius of gyration, is shown in Fig. 2, which is reproduced from an S.A.E. paper by Maurice Olley, who initiated the independent front springing system of the Cadillac and other General Motors cars.

The opinion is generally held that, other things being equal, the riding qualities of a car are best when $k^2 = ab$, that is, when the axes of pitch and bounce are directly over the rear and front axles, respectively. The sprung mass then has a relatively large moment of inertia and a relatively long radius of gyration, hence its frequencies around both axes will be low. But in order to get this relatively long radius of gyration it is necessary to "shove" a considerable portion of the mass ahead of the front and back of the rear axle, and this involves difficulties of design, besides tending to impair the appearance of the car, due to excessive overhangs of the body over the axles. In practically all of the latest prewar American passenger-car models, k^2/ab lay between 0.80 and 1.00, closely approaching the later value in some of the smaller cars.

Pitch and Bounce Frequencies—As long as k^2/ab is less than 1.00, the frequency of pitch is higher than the frequency of bounce, which may be explained on the grounds that the

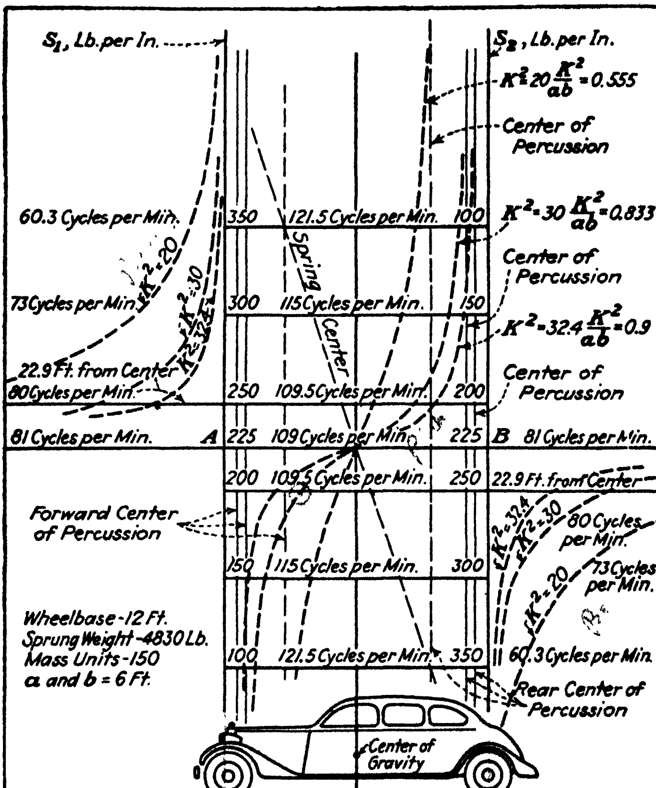


FIG. 2.—SHOWING HOW CONJUGATE AXES SHIFT WITH VARIATIONS IN SPRING RATES AND RADIUS OF GYRATION.

moment of inertia of the vibrating mass around the axis of pitch (within the wheelbase) is less than its moment of inertia around the axis of bounce (outside the wheelbase). It has been found also that the frequency of pitch is always somewhat higher than the front-end frequency calculated from the rate of the front springs and the mass supported by them, and the frequency of bounce is higher than the rear-end frequency calculated from the rate of the rear springs and the mass supported by them. When there is a considerable difference between the static deflections at front and rear, the frequencies of bounce and pitch will be quite different, and whenever these frequencies differ there is, of course, a constant shifting in the phase relation between the two oscillations.

In Fig. 3 the dotted curve of smallest amplitude and shortest period (highest frequency) represents the pitching motion, while the dashed curve, which has twice the amplitude and five-sixths the frequency, represents the bouncing motion. It must be remembered that both of these are angular motions. Now, while it is not entirely logical to add an angular motion around one axis to a second around another axis, the full-line curve, which is the resultant of the two dotted curves, may

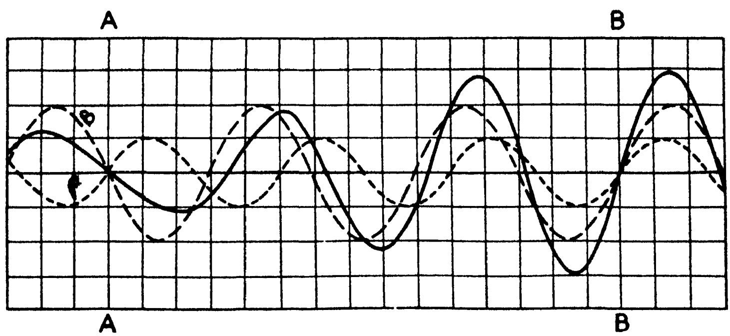


Fig. 3.—Showing How Unequal Front and Rear Spring Frequencies Cause Pitching.

be regarded as representing angular motion around an axis intermediate between the conjugate axes of pitch and bounce, which would be somewhere near the feet of the driver. At *A-A* the two basic oscillations are in phase opposition, and the resultant therefore is a minimum. After three periods of the pitching, or 2.5 periods of the bouncing motion, there is phase equality, at *B-B*, and the resultant angular motion around the intermediate axis is then a maximum. Since the driver's head is a considerable distance above the axis, it is subjected to a powerful horizontal acceleration, which is known as the "neck-snapping" effect. In this particular case, one cycle of the resultant motion occupies the same time as six cycles of the pitching motion. The more nearly equal the frequencies of pitch and bounce, the longer the period of the resultant motion. The oscillating motions, of course, are rapidly damped out by the shock absorbers, interleaf friction, etc., and when the period of resultant motion is quite long, they may be completely damped out before phase equality is reached.

Frequency Ratios—There are two cases for which the ratio of frequencies of pitch and bounce can be expressed in very simple terms. One is that when $k^2 = ab$, when the conjugate axes are directly over the axles, for which

$$\frac{\text{pitch frequency}}{\text{bounce frequency}} = \sqrt{f_r/f_f},$$

where f_r is the static deflection of the rear, and f_f that of the front springs. In this connection it must be remembered that the value of f_r/f_f varies with the passenger load. The other case is that when the static deflections of both pairs of springs are equal, for which

$$\frac{\text{pitch frequency}}{\text{bounce frequency}} = \sqrt{ab/k^2}.$$

From the foregoing discussion the conclusion may be drawn that two conditions essential to a comfortable ride are that both the front and rear springs be relatively soft (have a high deflection under static load), and that a good deal of the sprung mass be located well out in front and rear, far from the center of gravity. In setting values for the deflections under static load, care must be taken that enough clearance is provided to prevent excessive "striking through" or bumping, and that there will not be too much side sway, because when the springs are made softer, the clearance is lessened and the side sway increases.

Static Deflection and Clearance—One of the most important specifications of a chassis spring is the deflection under maximum static load. It is this factor which determines the relative softness of the spring and its ability to cushion or absorb shocks. The greater the static deflection of its springs, the better the riding qualities of the vehicle are likely to be. There are, of course, certain limitations to the static deflection which can be allowed. In the case of front springs which also serve as control members for the axle and wheels, it is limited by the requirement that spring action must not affect the steering perceptibly. This usually puts a limit of about 3 in. on the static deflection of such front springs. In one case where, by way of experiment, the static deflection was increased from 3 to 4.5 in., while riding comfort was materially increased, the steering became very erratic, and the car did not feel safe. Full brake application resulted in an excessive change in the caster angle, and, besides, the clearance was insufficient, so that the springs "bottomed" on the least provocation. The effect of the spring action on the steering can be lessened by the provision of auxiliary control members, such as links extending parallel with and below the rear halves of the springs, in which case the brake torque puts the links in compression and the rear half of the spring in tension. With such an arrangement static deflections as large as 6.5 in. have been found practical. For the so-called "level ride" the static deflection of the front springs must be even slightly greater than that of the rears, apparently because the front wheels hit the bumps first. Such large static deflections at the front call for accurate control of the wheel motion by separate control members, and an accurate solution of the problem of steering geometry, which are usually provided in independent front-suspension systems.

Static deflections are limited also by considerations of road clearance required. If the springs are softened, the road clearance will be reduced, even if steps are taken to keep the center of gravity under static conditions constant, for the greater static deflection calls for greater spring clearance, and this reduces the minimum road clearance. Finally, the item of cost enters into the problem, because with a given sprung mass, a given spring type, and a given maximum stress in the material, the amount of spring steel required varies in direct proportion to the maximum deflection, which is equal to the sum of the deflection under maximum static load and the clearance.

Static deflections of chassis springs usually come within the ranges given on page 520.

The clearance required is somewhat dependent on the deflection under maximum static load, but the optimum ratio of the former to the latter varies with the type of vehicle, and can be determined only by experience. It is smallest in the case of passenger cars, and largest in that of trucks, no doubt because the latter are more likely to be severely overloaded and to see more service on rough roads. The clearance can be made one-half the static deflection in passenger cars, three-fourths the static deflection in buses, and equal to the static deflection in trucks. As a rule, however, with any particular type of vehicle, the ratio of clearance to static deflection is made somewhat larger if the deflection is near the

FULL-LOAD STATIC DEFLECTIONS OF CHASSIS SPRINGS

Front:
- Standard passenger car, springs acting as control members, 2-3.5 in.
- Standard passenger car with auxiliary control members, 3.5-6.5 in.
- Standard passenger car with separate control members, 6.5-11 in.
- Midget cars, 2.5-3.5 in.
- Buses, 4-7 in.
- Trucks, 2-4 in.

Rear:
- Standard passenger car, 7-10 in.
- Midget cars, 3.5-4.5 in.
- Buses, 5-7.5 in.
- Trucks, 3.5-6 in.

lower limit of the range. Clearance always is figured on the "metal-to-metal" basis, the effect of the rubber bumpers in limiting spring deflection being neglected.

Number of Leaves—By examining the equation for the rate R (p. 477), we notice that the latter is directly proportional to the number of leaves, and inversely proportional to the cube of the length of the spring. Therefore, for a given rate, the number of leaves and the cube of the length of the spring change equally. As the number of leaves is decreased, the steps or overhangs become longer, with the result that stress distribution is less uniform, and the material is not used to such good advantage. This consideration makes six the smallest practical number for vehicle suspension springs. With increase in the number of leaves, the stress distribution improves, but the cost of the spring increases, and 16 is about the largest number ever used. Passenger-car springs rarely have more than ten leaves, except in the case of cross springs, the length of which is limited, and in which the required deflection therefore must be obtained by using leaves of light gauge, of which a large number are required to obtain sufficient load capacity.

The above considerations should be of help in selecting the proper number of leaves for both the front and rear springs, and from this, together with the summation of the thicknesses of individual leaves and the rate, the necessary length can be found, by means of the equations given in Chapter XVI. In passenger cars the front springs usually have a length equal approximately to one-third of the wheelbase, and the rear springs, to 45 per cent of the wheelbase. Lengths of cross springs are limited to about 80 per cent of the tread. As in the case of cross springs all of the load of one end of the vehicle is carried on a single spring, for the same static deflection, and therefore the same riding quality, its rate must be twice that of each of two longitudinal springs.

Leaf-Spring Dimensions—The dimensions required by the spring maker include the length under load (measured from center to center of spring eyes), the number of leaves, the width and thickness or gauge of individual leaves, the seat angle, the opening in the case of springs mounted below the axle, and the overall height in the case of those mounted above the axle. The seat angle is the angle between a line through the spring-eye centers and a tangent to the outline curve of the spring at the center of the seat (or at the center bolt). It is considered positive when the two lines describing the angle intersect forward of the axle, and negative in the opposite case. The seat, of course, is on the main leaf in springs mounted below the axle, and on the short leaf on those mounted on top of the axle. The opening is the distance from a line through the eye centers to the surface of the main leaf at the center bolt, while the overall height is the distance from the line through the eye centers to the exposed surface of the short leaf at the center bolt. Opening and overall height are considered positive when the spring has positive camber, that is, when the main leaf is convex toward the short leaf, and negative in the opposite case.

Spring makers also use the terms "drop," "drop gap," and "drop height," which are applicable where the line through the spring-eye centers is not parallel with a tangent to the spring outline at the center of the seat. All three are measured from a horizontal "datum" line through the center of the eye farthest from the tangent. The drop is the distance from the datum line to the center of the other spring eye; the drop gap is the distance from the datum line to the exposed surface of the main leaf at the center bolt, and the drop height is the distance from the datum line to the exposed surface of the short leaf at the center bolt. The drop gap and drop height have positive values when the camber is positive, and negative values in the opposite case.

Deflection and Stress Due to Torque—Where the Hotchkiss drive is used, and the reactions to driving and braking torque are taken on the springs, additional stresses are imposed on the latter. We found for the deflection of a semi-elliptic spring under a load P on each spring eye—

$$= \frac{5.7Pl^3}{Enbt^3}.$$

The load on each spring eye due to a torque T on the spring is $T/(2l)$, hence under a torque T the linear deflection of the spring will be

$$f = \frac{2.85Tl^2}{Enbt^3}.$$

If we designate the angular deflection in radians by δ, $f = l \tan \delta$, and as in the case of small angles the tangent is substantially equal to the length of arc, we may write $f = l\delta$. Substituting this in the foregoing equation and transposing,

$$\delta = \frac{2.85Tl}{Enbt^3}.$$

As the rate of the spring (p. 477),

$$R = \frac{Enbt^3}{2.85l^3},$$

by substituting we get

$$\delta = \frac{T}{Rl^2} \text{ radians} \quad \text{or} \quad \frac{57.3T}{Rl^2} \text{ deg.}$$

It was shown on page 477 that the stress in plate springs in terms of the deflection is given by

$$S = 1.05\frac{ftE}{l^2},$$

and substituting for f its equi valent, $l\delta$, we get

$$S = 1.05\frac{tE\delta}{l} \text{ psi.}$$

The maximum torque on the spring may be taken to be $0.6Wh$, where W is the maximum weight carried by the wheel next to the spring, and h the height of the spring eyes above the ground.

Sometimes, in order to provide for the stresses due to brake torque, an extra half leaf is included in the rear half of the spring, on top of the main leaf. It is secured at the center by the clips which hold the spring to the axle, while near the outer end and at an intermediate point it is tied to the other leaves by rebound clips.

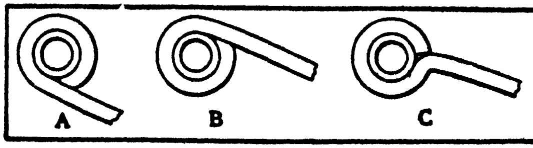


Fig. 4.—Conventional Types of Spring Eye.

Spring Eyes—There are three conventional types of spring eyes, known respectively as the up-turned, the down-turned, and the Berlin eye. The first-mentioned, shown at *A* in Fig. 4, is in most extensive use. It has the advantage over the down-turned type that there is less tendency for it to open up under load, and over the Berlin eye, that it is easier to produce. The advantage of the Berlin eye is that with it both longitudinal and transverse forces pass through the center of the main leaf, which reduces the resulting stresses. The only thing to recommend the down-turned eye is that the "geometry" of a spring with such eyes is different from that of springs with other eyes; the center around which the spring eye turns when the spring flexes is differently located, and this sometimes works out to better advantage when the chassis lay-out is made.

With multiple-leaf springs used on heavy commercial vehicles it is often difficult to get sufficient strength in the spring eyes. It is, of course, impractical to completely enwrap the eye of the main leaf by the second leaf, if both leaves are continuous or in a single piece each, as that would prevent the combination from flexing properly. In some cases the second leaf is wrapped halfway around the eyes of the main leaf, which gives what is known as a reinforced eye (*A*, Fig. 5). To make it possible to completely enwrap the eyes of the main leaf by the second leaf, the latter must be made in two parts, with a section at the center omitted and replaced by a filler block, so that each part can slide relatively freely between the leaves above and below. The end of such a spring is shown at *B*, Fig. 5. Some use has been made in springs for commercial vehicles of what is known as the military wrapper

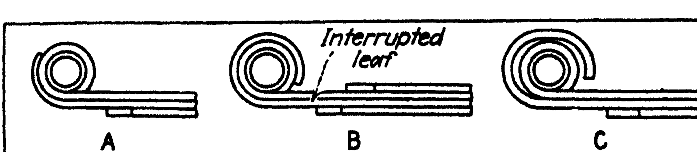


Fig. 5.—Special Types of Spring Eye.

(*C*, Fig. 5). There is sufficient clearance between the eye and the wrapper so that normal spring action is not interfered with, and in normal service the wrapper does not relieve the eye of load. It relieves the eye in extreme rebound, however, and it enables a vehicle to continue its journey if the main leaf should break. A simple method of strengthening the eye of the main leaf consists in forming it with a central circumferential rib, rolled or pressed in.

Leaf-Spring Geometry—When steering or mechanical-brake connections have to be made to parts carried by the axles, it is necessary to know the exact path of the axle under spring action, relative to the axis of the fixed spring eye. The following treatment of the subject of leaf-spring geometry is based on the section on Design Calculations of the *Manual on Design and Application of Leaf Springs*, published by the

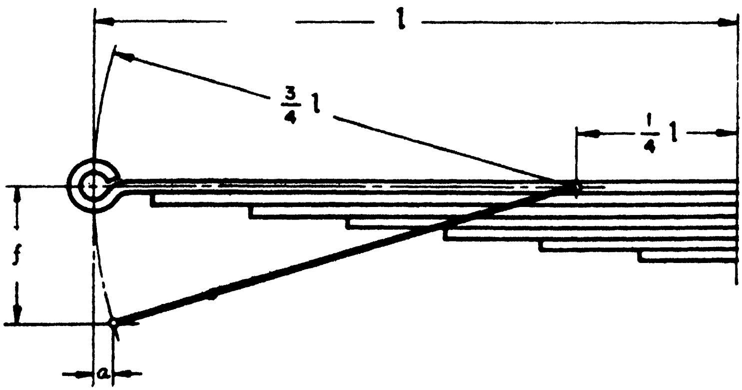


FIG. 6.—LINK EQUIVALENT OF CANTILEVER SPRING.

Society of Automotive Engineers and credited to notes of Maurice Olley.

A leaf spring having a circular-arc shape when in the free state, and designed to have uniform stress distribution along its length, will have a circular-arc shape at all loads. The following relations were arrived at analytically, and were checked on a number of actual springs: The center of a Berlin-type eye describes a circular arc of radius $0.75l$ around a center located on the main-leaf center line (Fig. 6). If the spring has eyes—up-turned or down-turned—whose center is at a distance e from the center line of the main leaf, the center of the arc described by the eye center will be at a distance $e/2$ from the main-leaf center line, in the opposite direction. Using this construction, the change in the length of chord can be determined to within 1 per cent, up to deflections equal to $0.6l$.

A semi-elliptic spring can be replaced by two cantilevers, and then may be represented by the three-link mechanism shown in Fig. 7. Similar diagrams can be drawn for unsymmetrical semi-elliptic springs, and for springs with eyes offset either in the same direction or in opposite directions. The most convenient way to draw the diagram is to start from the position in which the main leaf has no camber. In that position the linkage is not flat; it is flat when the distance between eye centers is a minimum, when the main leaf has a camber of $+1.5e$ with down-turned and $-1.5e$ with up-turned eyes.

The motion of any part rigidly secured to the spring seat can be plotted by making use of the two characteristics of motion of the center member of the three-link mechanism indicated in Fig. 7. This member tilts in such a way that its center line produced always passes through point O, at a distance A from the center of the main leaf when without camber. The tilt angle per inch of deflection is equal to $(l_1 - l_2)/l_1l_2$ radians, and this together with the fact that the action radius is approximately $0.75(l_1 + l_2)$ determines the angular motion of the spring center in a Hotchkiss-drive layout. All points rigidly secured to the spring seat tilt the same as the central link. The center point of the spring seat describes an arc having a radius equal, approximately, to $0.375(l_1 + l_2)$, around a center located on line 1-2. All other points on parts rigidly secured to the spring seat describe arcs of different radii, around centers differently located, both of which can be determined by means of a layout on the basis of the available data.

Spring Bushings, Shackles, and Shackle Bolts—The eyes of commercial-vehicle springs usually have bronze bushings pressed into them, made of any of the commercial bearing bronzes. These bushings are made with a wall thickness of ⅛ in. Such bushings formerly were used also in passenger-car springs, but there they have been largely discarded, on account of their requirements for service, and their tendency to develop squeaks and rattles. The bearing load on these bronze bushings should not exceed 1000 psi, and preferably should be considerably less. The so-called spring bolts frequently are plain cylindrical pins which are clamped in the spring brackets or shackles by means of clamp screws. The clamp screw passes slightly below the surface of the pin, which has a groove turned in it. (Fig. 8.) Axial and radial holes are drilled in the pin for lubricating purposes, and a high-pressure lubrication fitting is screwed into its end. An

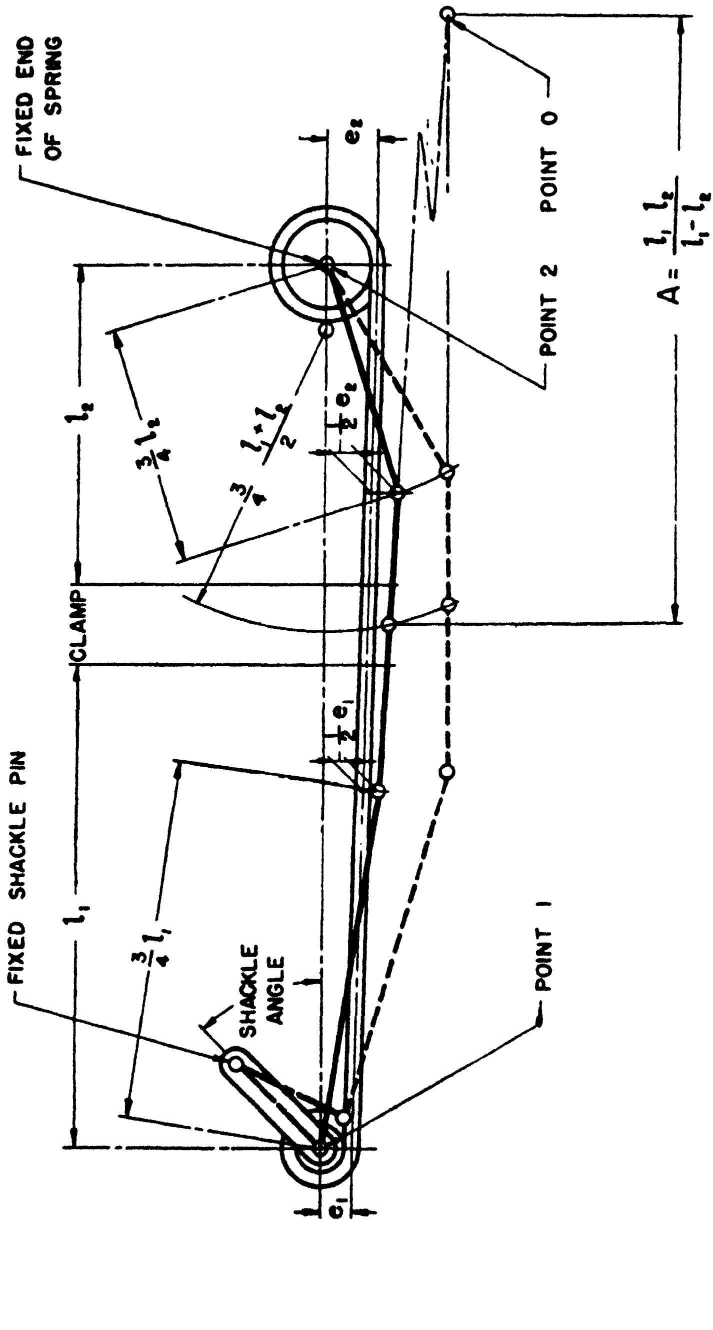


FIG. 7.—THREE-LINK EQUIVALENT OF SEMI-ELLIPTIC SPRING.

alternate scheme is to use a hollow floating pin, together with a retaining bolt passing through it, as shown in Fig. 9. There are bushings in both the spring eye and the spring bracket, those in the bracket having flanges on them to take thrust loads. A reservoir is formed in the tubular pin, from which lubricant can pass to the bearing surface through holes drilled in the wall of the pin.

At present considerable use is made of threaded spring bolts (Fig. 10), especially in passenger cars. A threaded bushing is pressed or screwed into the spring eye, which latter fits into the spring bracket or shackle with considerable side clearance. The spring pin has a corresponding thread cut on it, and is a free fit in the threaded bushing. Advantages of the threaded shackle are that the radial loads—because of the V threads—prevent side play; that large areas are provided for thrust loads, and are effectively lubricated

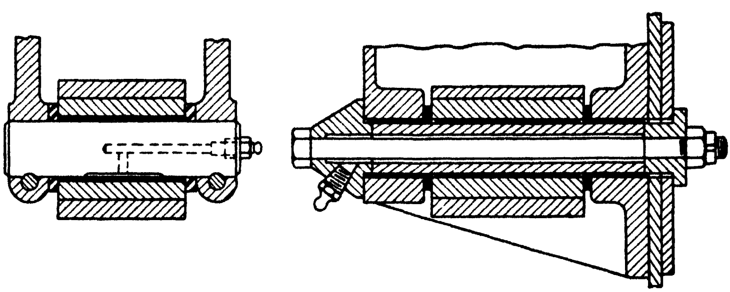
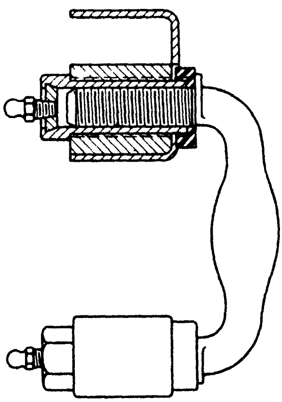


FIG. 8 (*left*).—BRONZE-BUSHED SPRING EYE AND PLAIN SPRING BOLT.
FIG. 9 (*right*).—FLOATING SPRING BOLT.

because they are more or less sealed by grease; and that wear on the pin and bushing does not increase the lateral freedom or play.

The shackle shown in Fig. 10 is of the C-type, in which the two spring bolts or pins are forged integral with the single shackle. With this type, the pins are not so rigidly held in parallelism as when separate shackles are used, and the torsional stresses on the main leaf undoubtedly are somewhat reduced. Fig. 11 shows another type of shackle, also with threaded pins, the peculiarity of this design being that the ends of the pins are tapered and seated in correspondingly tapered holes in the shackles, which latter are drawn together and into firm contact with the pins by means of a central bolt. Sliding motion is thus restricted to the threaded bearing surface.

Two other shackle types are illustrated in Figs. 12 and 13. The first is known as the Y type. The bosses of its two prongs are drilled to take the ends of the pin passing through the spring eye, while the other, long boss takes the pin of the frame bracket. Fig. 13 shows an H-type shackle, which consists essentially of two plain shackles connected by an integral cross bar. The two one-piece shackles illustrated undoubtedly provide greater lateral stiffness than a pair of ordinary shackles, especially after the vehicle has seen considerable service. It seems to be the consensus of spring engineers that while a certain amount of lateral flexibility, as provided by the C-type shackle, for instance, is desirable, any construction with which there is a tendency to rapid increase of lateral freedom in service is undesirable.



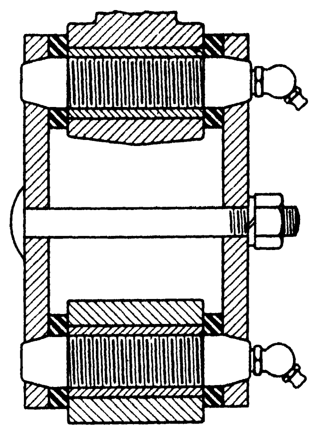


FIG. 10 (*left*).—C-SHACKLE WITH INTEGRAL THREADED PINS.
FIG. 11 (*right*).—PLAIN SHACKLES WITH TAPERED SEATS FOR PINS.

Rubber-Bushed Spring Eyes—In the passenger-car field the trend in recent years has been toward the use of rubber bushings in spring eyes and brackets. Two different applications of such bushings are shown in Figs. 14 and 15. In the first, two bushings are pressed directly into the spring bracket and the spring eye, each extending half-way through and having a flange at its outer end, on which thrust loads are taken. Both ends of the spring bolt are reduced in diameter, one end being riveted to its shackle, the other clamped in its shackle by the nut. The length of the central portion of the pin, which is surrounded by the rubber bushings, must be held to close limits, as this dimension determines the pressure exerted on the bushings when the nut is tight-

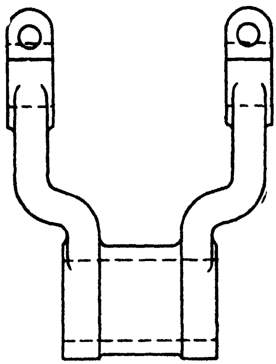
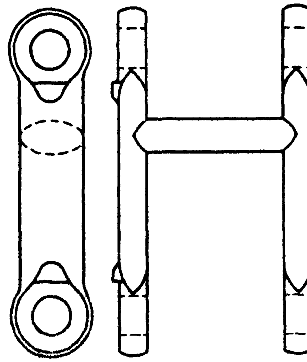



FIG. 12 (*left*).—Y-SHACKLE.
FIG. 13 (*right*).—H-SHACKLE.

ened. In the bushing shown in Fig. 15, the rubber is confined between concentric metal tubes, of which the inner tube serves as a spacer for the arms of the bracket, while the ends of the outer tube are flanged inward. These flanges, together with a central enlargement of the inner tube, enable the rubber bushings to take lateral loads. A bushing of this general type, known as the Silent-bloc, has long been in use in European countries. It consists of concentric metal tubes, with the space in between filled with rubber under pressure. The rubber varies in thickness from 0.050 to 0.200 in. One of the tubes can be turned 60° relative to the other, and will return

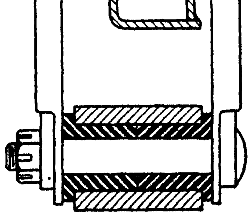
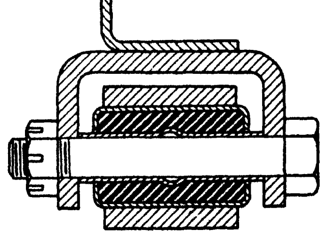



FIG. 14 (*left*).—RUBBER BUSHINGS ON SHACKLE PINS.
FIG. 15 (*right*).—METAL-CASED RUBBER BUSHING ON SPRING BOLT.

to its original position when the torsional moment is removed.

Effect of Shackle on Spring Rate—The deflection of a spring under a given load can be determined in a spring-testing machine, but what counts in service is the vertical drop of the vehicle frame under a given increase in load, and this depends on both the spring and the shackle. The shackle affects the "deflection" (or the rise and fall of the chassis) in two distinct ways. In the first place, as the load is increased, the distance between spring eyes increases, and this results in a change in the angular position of the shackle. With every change in the angular position of the shackle there is a change in the height of the frame. The shackle angle,

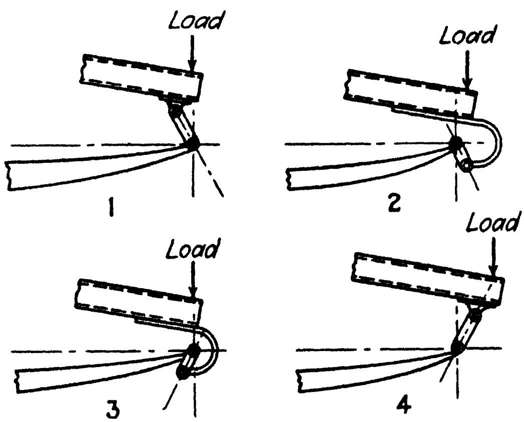


FIG. 16.—SPRING SHACKLE IN FOUR DIFFERENT POSITIONS.

however, also has a direct effect on the spring action, for except when the shackle stands perpendicular to the line connecting the centers of the spring eyes, the load on it has a component parallel with that line, and except when the camber is zero, any change in the deflection of the spring causes the eye to execute a movement which has a component along the line connecting the spring-eye centers. The eye, therefore, moves either with or against a component of the load imposed on the spring by the shackle. In the former case the shackle subjects the main leaf to tension; in the latter to compression. When the shackle subjects the main leaf to tension, the load rate and the frequency (oscillations per minute) increase, whereas when the main leaf is subjected to compression, the shackle effect lowers the load rate and the frequency.

There are four possible arrangements of spring shackles, all of which are illustrated in Fig. 16. In the drawing 1 is a compression shackle subjecting the main leaf to tension; 2, a tension shackle subjecting the main leaf to tension; 3, a tension shackle subjecting the main leaf to compression, and 4, a compression shackle subjecting the main leaf to compression.

Arrangement 4 is rarely used, because with it, in cases of extreme rebound, the shackle is likely to pass the "dead-center" position and become a tension shackle. This can be prevented, however, by using a so-called non-reversible shackle, which is similar to the one shown in Fig. 13, except that the cross bar is located above the upper eyes.

The effect of the shackle on the spring rate and the frequency depends on the length and position of the shackle, on the camber of the spring, and on the spring load. Charts for determining the ratio in which the calculated or nominal

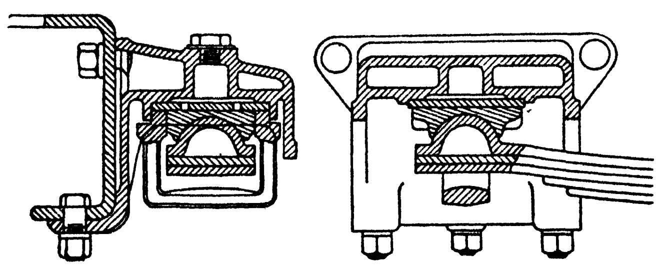


FIG. 17.—SPRING WITH SLIDING SHOES AND WEAR PLATES.

spring rate is increased or decreased by the shackle effect are given in *Manual on the Design and Application of Leaf Springs* published by the Society of Automotive Engineers.

Springs with Flat Bearing Surfaces—Instead of the main leaf being formed with eyes, its ends may be left flat and made to bear against wear plates. Such bearings, of course, are self-adjusting for wear. An embodiment of this principle, which has been used on a foreign truck, is illustrated in Fig. 17. The main leaf, which is supported by the second and third leaves over its entire length, has a cup or hemisphere formed on it at each end, lodged in a socket on a shoe that slides over a wear plate on the spring bracket. A yoke spanning the end of the spring prevents the cup from leaving the socket during severe rebounds. A lubricant reservoir is provided directly over the wear plate.

Rubber Shock Insulators—Mack Motor Truck Company has been using so-called rubber shock insulators in the con-

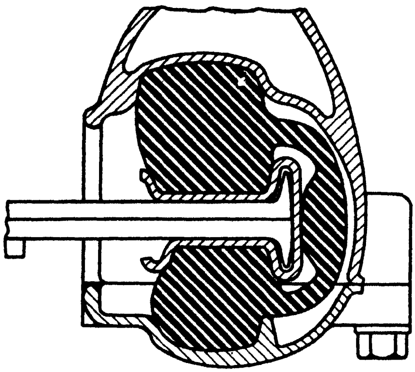


FIG. 18.—MACK RUBBER SHOCK INSULATOR.

nections between the springs and the chassis frame of its motor trucks and buses for a long time. As shown in Fig. 18, the two longest leaves of the spring have their ends turned at right angles and placed inside a metal-lined, molded rubber block, which is compressed in a housing riveted to the frame. A bolted bottom plate holds the assembly of spring end and rubber block together. The design permits of only limited motion of the main leaves in the direction of their length, and the springs therefore are designed so that they are substantially straight under normal load.

Center Bolts—Spring center bolts are made in four sizes, viz., 5/16-24 (5/16 in. diameter, 24-pitch thread), 3/8-24, 7/16-20 and 1/2-20. These bolts have round, slotted heads. It is recommended by the S.A.E. that 5/16-in. bolts be used for passenger-car springs of 1 1/2, 1 3/4 and 2 in. width, and for truck springs of 2 in. width; 3/8-in. bolts for passenger-car springs of 2 1/4 and 2 1/2 in. width, and truck springs of 2 and 2 1/4 in. width; 7/16-in. bolts for truck springs of 2 1/2 and 3 in. width, and 1/2-in. bolts for truck springs of 3 1/2 in. width and over. The general form of the bolt is shown in Fig. 19. A fillet of from 1/64 to 1/32 in. is provided below the head.

The center bolt, or, rather, the hole for it, weakens the spring, and many attempts have been made to eliminate it. Sometimes the leaves are provided with transverse humps at the center, while in other cases they are made with circular humps that nest in depressions in the leaves below. The leaves are then held togther by means of clips during transportation and assembly.

Spring Clips—Clips are used on laminated springs for two purposes—to distribute rebound stresses over a number of leaves and to hold the leaves in alignment. Most widely

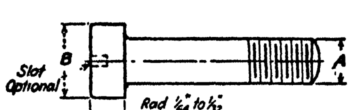


FIG. 19.—CENTER BOLT.

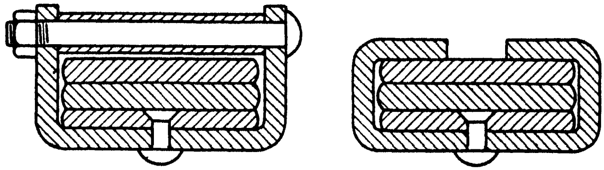


FIG. 20 (*left*).—BOLT-TYPE REBOUND CLIP.
FIG. 21 (*right*).—CLINCHER-TYPE REBOUND CLIP.

used, especially on the heavier springs, is the bolt-type clip illustrated in Fig. 20. It consists of a length of strip steel bent to U-shape and riveted to the end of the shortest leaf surrounded by it. The ends of the clip are drawn together by a bolt surrounded by a tubular spacer. The inside width of the clip must be such that the total clearance between it and the spring plates is at least 1/16 in., to prevent localization of stress at the outer end of the main leaf when the spring is subjected to a torsional moment in a transverse plane. In Fig. 21 is shown a clinch clip. Rebound clips are made of 5/32 by 5/8-in. or 3/16 by 3/4-in. stock for passenger cars, and of 3/16 by 1-in. or 1¼-in. stock for truck springs. There is still another type, known as a box clip, which is made of lighter (3/32-in.) stock and used chiefly together with spring covers, its ends being interlocked. Instead of anchoring it to the shortest leaf embraced, by means of a rivet, the overhanging, tapered portion of that leaf may be spread to project beyond the edges of the remaining leaves, and slots cut in the sides, in which the sides of the clip are located. When spring covers are used, rebound clips are generally considered unnecessary.

Rebound Leaves—Rebound can be limited also by placing a couple of leaves with reverse camber on top of the main leaf, as shown in Fig. 22. These reinforce the main leaf during the rebound and keep down the stress in it. While such rebound leaves have been used to a certain extent in the past, they are now generally regarded with disfavor, because they stiffen the spring. Addition of leaves to an existing complete spring increases the spring rate equally, whether such leaves assist or oppose the other leaves.

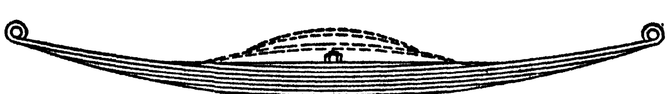


FIG. 22.—SPRING WITH REBOUND LEAVES.

Variable-Rate Springs—The great majority of all chassis springs are of the constant-rate type. One of the shortcomings of these springs is that if they are so designed that they will not "bottom" when subjected to severe shock while carrying their maximum load, they will be hard under light load, while if they are designed to give a good "boulevard ride," they will "strike through" all too easily under adverse conditions. A good boulevard ride and freedom from bottoming can be obtained with a spring of such design that its rate increases with the deflection. Such variable-rate springs are particularly needed for commercial vehicles, in which the spring load often varies greatly with the degree of loading of the vehicle. However, they have been used also on passenger cars. Following are two examples of the wide variation of spring loads in commercial vehicles: In a 30-passenger bus with rear-mounted powerplant, the load on each front spring may increase from 1500 lb with the driver alone, to 4200 lb with a full complement of "standees." In a 3-ton truck the

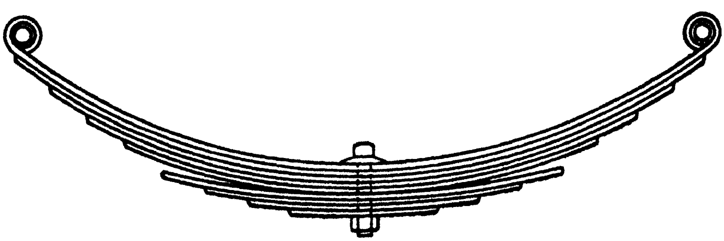


FIG. 23.—TWO-UNIT PROGRESSIVE VARIABLE-RATE SPRING.

load on each rear spring may increase from 1400 lb without load to 7000 lb with a 100 per cent overload. Assuming the maximum static deflection of the bus front springs to be 6 in., with constant-rate springs the natural frequency of the front end would increase from 77 cycles per minute at full load, to 128 at no load, while in the case of the truck, if the maximum static deflection were 5 in., the frequency would increase from 84 at full load to 188 at no load. Both of these no-load frequencies are too high for personal comfort and for the good of the mechanism.

One type of variable-rate spring consists of two or three units, placed one above the other. A two-unit spring is illustrated in Fig. 23. The first unit comprises the main leaf and two or three others, while the second unit also consists of a number of leaves, the longest of these being shorter than the shortest of the first unit. Leaves of the first unit are of thinner gauge than those of the second, and in the free state the first unit has considerably more camber than the second. As the load on the spring is increased, contact between the two units gradually extends outward, and the rate of the combination increases. The maximum deflection of the second unit, measured at the spring eye, is considerably less than that of the first, for which reason its leaves may be of heavier gauge. However, since the stress in the second unit under static load is quite small, if both units were designed for the same maximum stress, the stress range would be greater in the second unit, which would be subjected to greater fatigue effects in consequence. For this reason the second unit is designed for a lower maximum stress.

In a certain passenger-car three-unit spring the first unit had a maximum static deflection of 12 in., the second of 8 in., and the third of 5 in. In service, the first and second units never broke contact, as the rebound was limited by the shock absorbers, but there was a space of ⅛ in between the second

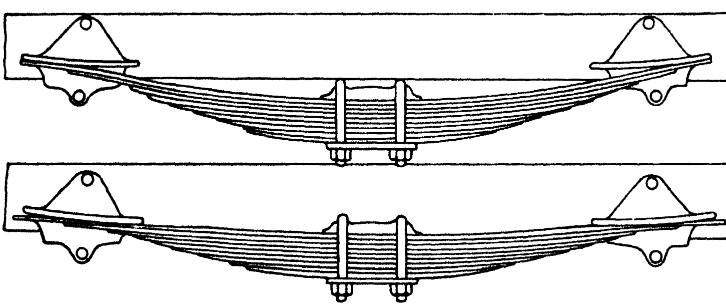


FIG. 24.—SINGLE-UNIT PROGRESSIVE VARIABLE-RATE SPRING.

and third units under maximum static load. Under road shock, of course, the entire spring closed up completely.

In another type of spring, illustrated in Fig. 24, the variable rate is obtained by making the flat or slightly rounded ends of the main leaf contact convex wear plates. Under light load the spring plate contacts the wear plate near its outer edge, as shown in the upper view, and as the load increases the line of contact moves inward, until at full load it is near the inner edge of the wear plate, as shown in the lower view. Thus, as the load increases the effective length decreases, and the spring becomes stiffer.

With the type of spring just described, the change in rate between minimum and maximum spring loads is rather limited, although it is greater than the change in the reciprocal of the effective spring length. A greater change in rate can be obtained by means of the Mather Magnaflect suspension system, in which the spring load is transferred to the axle through the spring and a lever or radius rod in tandem with it. When the vehicle is lightly loaded, the spring contacts the lever a considerable distance ahead of the axle, as shown in the upper view of Fig. 25, and the spring load then is greater than the wheel load in the proportion of the horizontal distance from the lever fulcrum to the axle, to the distance from the fulcrum to the point of contact between spring and lever. The lever thus multiplies the load on the spring, and causes it to deflect more under a given increase in wheel

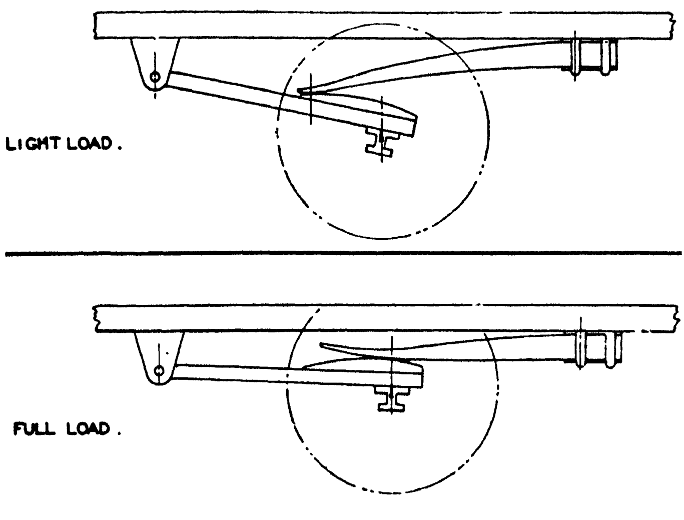


Fig. 25.—Mather Magnaflect Spring in Extreme Positions.

load. When the vehicle carries a full load, the spring contacts the lever directly over the axle, as shown in the lower view, and the spring load then is equal to the wheel load.

Helper Springs—To take care of the large variations in spring load in commercial vehicles, helper springs have been widely used. They usually are mounted directly above the main springs, as shown in Fig. 26. As long as the vehicle is lightly loaded, only the main spring is active, but when the load reaches a certain predetermined value, the helper spring contacts brackets on the frame, and thereafter the load is shared by the two. Helper springs often are made excessively stiff, with the result that the riding qualities are not nearly as good at maximum as at light load, whereas the reverse holds true with single springs. If we take the ratio of spring load to spring rate to be an index of the riding quality, it is obvious that the latter improves with increase in load up to the point where the helper spring becomes active. There it suddenly deteriorates, owing to the increase in spring rate, but on further increase in load it improves again. It would seem logical to have the helper spring become active at a load midway between minimum and maximum, and to have the riding-quality index—the ratio of spring load to spring rate—the same at the midpoints of both spring ranges. Then, if we designate the ratio of maximum to minimum load by a, the helper spring must have a rate equal to $(2a - 2)/(3 + a)$ times that of the main spring, and must become active when the spring load is equal to $(1 + a)/2$ times the minimum.

Spring Seats and Clips—Semi-elliptic springs are secured to the spring seats of the axles by means of studs, box clips, or U-clips. These holding-down members are preferably made

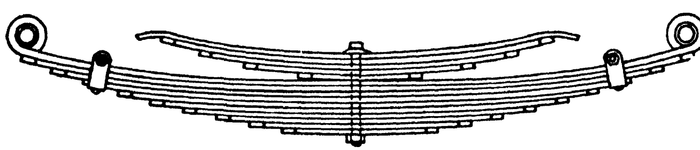


FIG. 26—TRUCK SPRING WITH HELPER SPRING.

of alloy steel of low carbon content, which is highly ductile and therefore does not break easily. The shank of the clip is made of a diameter given by the expression $0.22\sqrt{bh}$, where b is the width of the leaves and h the height of the spring section. It has S.A.E. threads cut on it. Hexagon nuts are used on these clips, of extra length (about twice standard), and they are locked by either a heavy-type spring washer, a lock nut, or a cotter pin. The distance between clips is made as small as the design of the spring seats or the diameter of the axle permits, in order to minimize the restraint on the spring action of the leaves. Efforts in this direction are apparent in the design of the spring mounting shown in Fig. 27 where the clips, instead of being parallel, are inclined toward each other, and therefore are closer together where they bear down on the spring. Moreover, the pressure plate, instead of having a flat or very slightly convex under surface, contacts the spring only directly below the clips.

A pad of soft material is often placed on the seat under

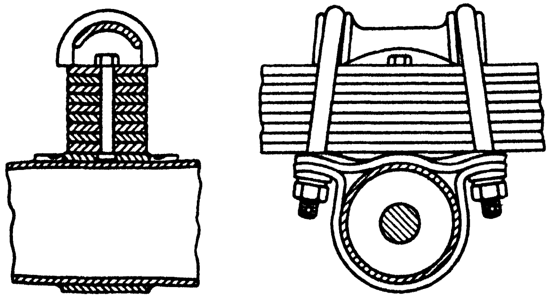


FIG. 27.—MOUNTING DESIGNED TO MINIMIZE INACTIVE SPRING MATERIAL.

the spring, to prevent concentration of stress, and good results have been obtained with two layers of 8-oz. duck soaked in white lead. In the case of front springs, a thin metallic wedge usually is placed between the seat and the spring in addition, to provide the desired wheel caster.

Fig. 28 shows a box clip in place. Originally these clips were made with a semi-elliptic section, the width being equal to one and one-half, and the thickness to two-thirds the shank diameter. This, however, makes them rather weak at the corners, and the proportions shown in the drawing are better. In the case of front springs, the clips usually serve also to hold the rubber bumpers in place. The bumper is provided with a sheet-steel retainer with flaps that are held by the clips.

Rear springs are either made rigid with the axle housing or free to turn thereon. The former arrangement is used in connection with theHotchkiss drive, in which all torque reaction on the axle housing has to be taken up on the springs, while the latter can be used where a torque arm or torque tube is provided. Front springs are mounted on top of the axle,

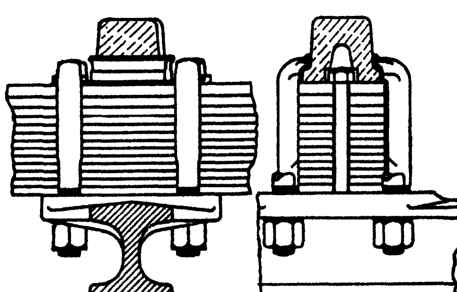


FIG. 28.—SPRING MOUNTING WITH BOX CLIPS.

which latter is "dropped" to place the frame and powerplant as low as ground-clearance requirements permit. The rear springs of trucks usually are mounted on top of the axle, while in practically all passenger cars and buses they are underslung, to lower the center of gravity of the vehicle.

In axles with pressed steel or tubular steel housings the spring seat is welded to the housing. The spring is then secured to the housing by means of U-clips which fit the axle and recesses at the corners of the seat snugly, and pass through holes in the spring block or clamping plate.

Two designs of spring seats for underslung rear springs are shown in Fig. 29. The one on the left consists of a drop forging machined to fit the axle tube and welded to it. It is recessed to reduce its weight and to provide additional welding area. This type of seat is used on cars having Hotchkiss

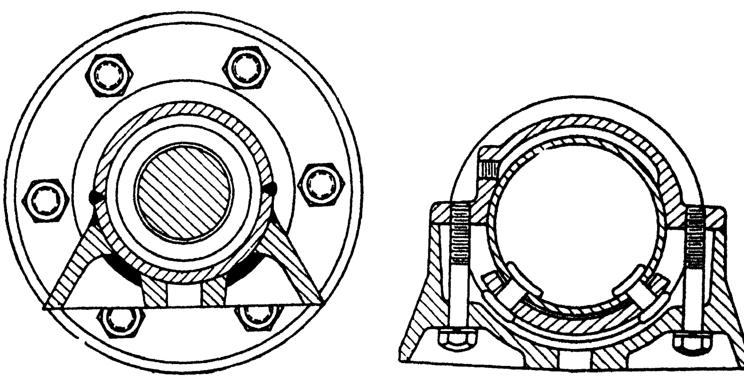


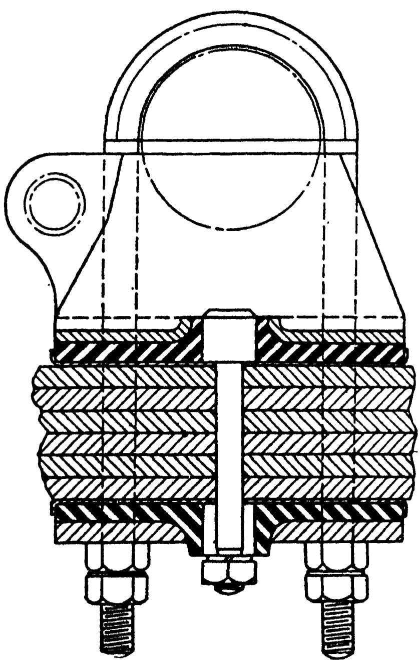
FIG. 29.—MOUNTINGS FOR UNDERSLUNG REAR SPRINGS.

drive, while the one on the right is used on cars with torque-tube drive. The latter completely surrounds the axle tube, but is split horizontally through the center, the two parts being held together by cap screws. The upper part is recessed to form a lubricant reservoir, while the lower part is recessed to accommodate a stop limiting the angular motion of the axle relative to the spring.

On passenger cars it is now common practice to place rubber pads, vulcanized to sheet-metal retainers, both above and below the spring, to prevent the transmission of noise from the axle to the frame and body. Such a construction is shown in Fig. 30. The surface of the rubber pad where it contacts the spring is sometimes grooved, to increase the flexibility or yieldability of the insulator.

Pressure Block—Instead of having a box clip bear directly on the spring leaves or on a sheet-metal spacer, a pressure block is now generally provided, and is designed to be held down by U-clips. The one shown in Fig. 31 is made with grooves for the clips, with a hole or socket for the center bolt, ribs to ensure lengthwise rigidity, and a curved bearing surface intended to prevent localization of stress and to reduce the amount of inactive spring material to a minimum.

Interleaf Friction—With ordinary leaf springs the spring action is controlled to a considerable extent by the interleaf friction, which is relatively high because of the roughness of



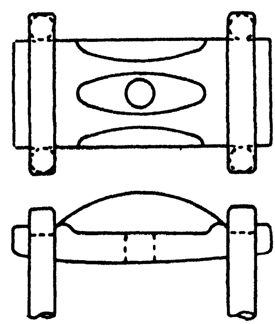


FIG. 31 (*above*).—PRESSURE BLOCK.

FIG. 30 (*left*).—RUBBER-INSULATED SPRING MOUNTING.

the rolled surfaces. Some experimental results on interleaf friction, obtained at the Detroit Steel Products Company plant, were given in an S.A.E. paper by Tore Franzen. The coefficient of static friction between dry spring leaves was found to be 0.35; the coefficient of sliding friction, 0.22. With lubricant on the leaves the friction increased with the load, which is the exact opposite of conditions in bearings with complete-film lubrication. The explanation of the apparent discrepancy undoubtedly is that whereas the bearing surface is smooth, the surface of the spring leaf is relatively rough. Friction between spring leaves dropped to its lowest values when the springs were lubricated with a mixture consisting of equal volumes of cylinder oil and powdered graphite. With such a lubricant the coefficient of sliding friction varied between 0.04 and 0.06 within the practical load range, and the static friction was only slightly higher, the coefficient reaching a maximum value of 0.08 at the upper end of the load range. When the plates were lubricated with light oil, the coefficient of sliding friction rose from about 0.04 at light to 0.10 at heavy loads, while the coefficient of static friction ranged between 0.20 and 0.28 over most of the load range. With the excess oil wiped off, the coefficient of sliding friction was 0.18 over most of the range, while the coefficient of static friction was about 0.36.

Spring Covers—In practice it is difficult to keep lubrication conditions of leaf springs uniform, unless they are enclosed. When exposed to the weather, the lubricant dries up, and the leaves may even become rusty, in which case the

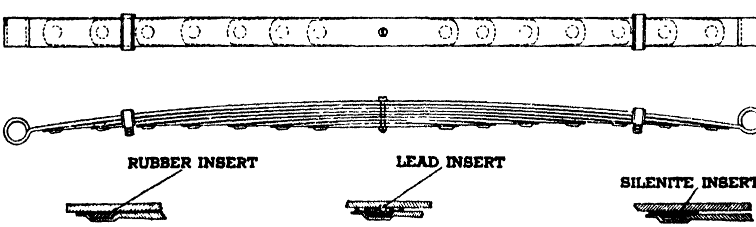


FIG. 32.—PACKARD REAR SPRING WITH INSERTS.

friction will be greatly increased, and the springs will squeak in action. Spring covers therefore have come into extensive use. These covers are made either of metal, of fabric, or of both metal and fabric. As a rule, each cover protects one-half of a semi-elliptic spring, extending from near the seat to near the eye. The covers are filled with lubricant. Car manufacturers usually advise against the use of lubricants containing inert materials or fillers, such as asbestos fiber and silica, but recommend lubricants containing a rust inhibitor.

Interliners and Inserts—Interleaf friction often is controlled by the use of either interliners or inserts. An interliner is a thin strip of some material other than spring steel, placed between adjacent spring leaves, while an insert may be defined as a block or pellet of some material having special frictional qualities, placed in a depression formed in that portion of the spring leaf which overhangs the leaf next below it. Fig. 32 shows a Packard rear spring in which inserts of three or four different materials are used. The insert projects slightly above the top of the leaf, so that the parts of adjacent leaves at which there is relative motion contact each other only through the inserts. The general object of the inserts is to reduce the interleaf friction and make it as nearly uniform as possible. Some insert materials are chosen on account of their low coefficient of static friction, while others are chosen for their low coefficient of sliding friction. Among the insert materials that have been used are wax-treated fiber board, self-lubricating bronze, a self-lubricating plastic material, rubber, and lead-antimony alloys.

Suspension on Coil Springs—While coil springs have been used most extensively in independent suspension systems for the front of passenger cars, they can be used also together with rigid axles, and rear suspension on coil springs has been a feature of two or three American passenger-car makes for some years. The advantage offered by coil over leaf springs when so used is that the amount of spring steel required is substantially cut in half. For instance, the rear coil spring of a moderate-sized passenger car, which supports a normal load of 850 lb, weighs only about 12 lb. From the user's standpoint, perhaps the greatest virtue of rear coil springs is that they operate uniformly and noiselessly throughout their life, or that of the vehicle.

When coil springs are to be used at the rear, one of the problems that arise is to accommodate a spring which under normal load must be from 12 to 14 in. high, without raising the frame above its normal level in a car with conventional suspension. The problem can be solved by placing the spring in a housing or retainer of inverted-cup shape, set into openings in the frame and projecting an inch or two above the top of the frame, into the rear-seat box. The springs support the frame at the junction of the side rails with the rear cross member, the latter being formed with integral gussets large enough to encompass the spring housing.

Fig. 33 shows the rear spring installation on Buick cars. It will be seen that pressed-steel construction is used extensively. Pressed-steel diagonal brace rods or torque bars extend from the forward end of the torque tube or third member of the axle, to the axle housing at points just inside the frame side rails, and are clamped to the axle tube together with pressings which form the seats or supports for the springs. These axle braces, together with the torque tube, guide or steady the axle in the fore-and-aft direction, while a radius rod pivoted to a frame bracket on one side of the car,

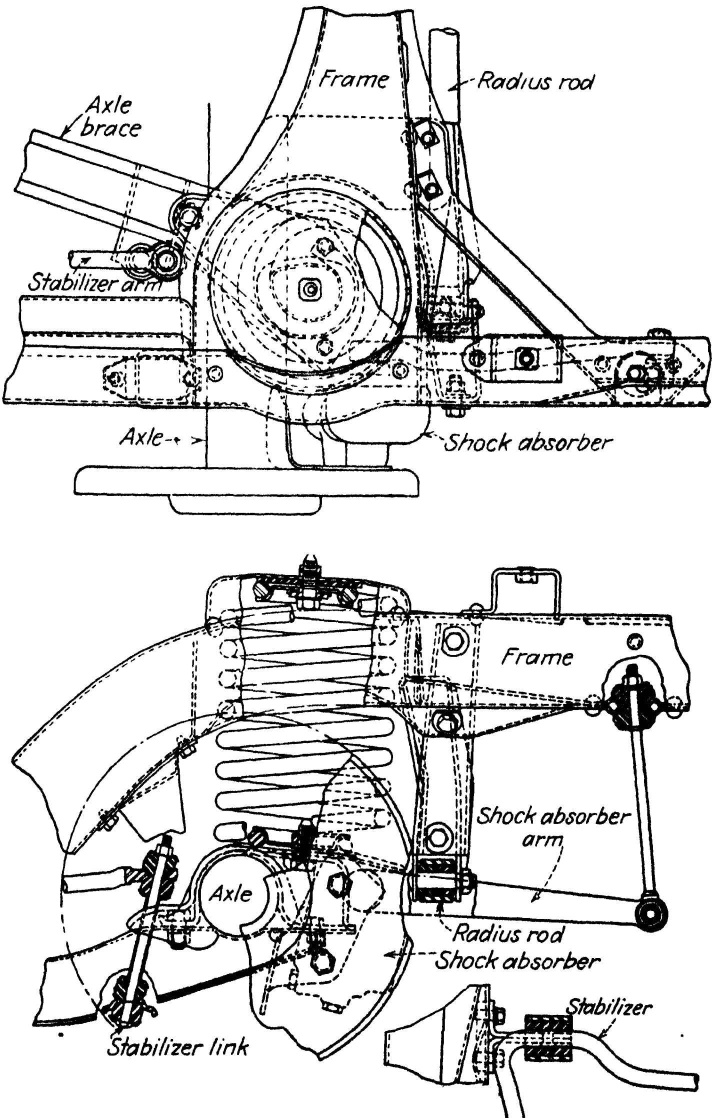


Fig. 33.—Buick Rear Suspension on Coil Springs.

and to an axle bracket on the opposite side, guides it in the transverse direction. A detail view of the radius-rod installation is shown in Fig. 34. The shock absorber is mounted on the brake backing plate, its arm extending rearward and connecting by a link with rubber-insulated joints to the frame side rail. A stabilizer bar carried in rubber-bushed bearing brackets extending from the frame side rails and connecting by links with rubber-insulated joints to brackets on the axle brace bars is located in front of the axle.

Spring Dampers or Shock Absorbers—When a spring-supported mass, such as that of a motor-vehicle chassis, is given an impulse, it is set vibrating, and it keeps on vibrating until the energy of the impulse is completely frittered away

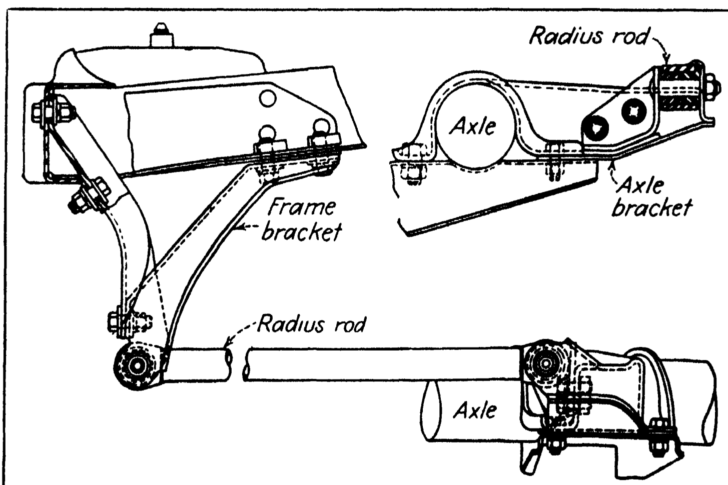


FIG. 34.—TRANSVERSE RADIUS ROD USED WITH COIL SPRINGS.

in overcoming damping forces. There are certain natural damping forces in all vibratory systems. In vehicle suspensions the natural damping force is a minimum if coil springs or torsion bars are used, when it is due chiefly to friction in the joints or bearings of the guiding mechanism. With leaf springs the interleaf friction is the chief source of damping force. But in no case is this damping force sufficient to ensure a comfortable ride under all conditions, and on passenger cars and buses special damping devices or shock absorbers are commonly fitted. The term "shock absorber," by the way, is a misnomer, as it is the spring that absorbs the shock. "Spring damper" would be a suitable name, but the term shock absorber is in such wide popular use that it would be futile to try to replace it.

The first shock absorbers used on motor vehicles were of the dry-friction type. They consisted of two sets of circular metal discs held in frictional contact with each other by a spider-type flat-metal spring on a bolt passing through central holes in the discs. Discs of one set alternated with those of the other set. Each set of discs was provided with a radial arm, and the two arms were pivotally connected to the axle and the vehicle frame, respectively. Under spring action there would be relative motion between the two sets of discs, and the friction would damp out the vibrations of the chassis. With these "solid-friction" spring dampers the damping force is constant, that is, independent of the velocity of motion or the amplitude of the vibration, and it is equally effective during both compression and rebound of the springs. When the damping force is active during both compression and rebound, it damps out vibration more rapidly, but, on the other hand, a damping force active during compression makes the springs that much stiffer, and therefore tends to transmit road shocks from the axles to the vehicle frame.

Spring dampers based on the solid-friction principle have been abandoned in favor of hydraulic devices. Hydraulic shock absorbers consist of one or more cylinders, with pistons adapted to move axially therein, or of a chamber of cylindrical-segment form in which a vane is adapted to move angularly. The cylinders or chamber compartments are filled with shock-absorber fluid, and motion of the pistons or vane forces the fluid through a restricted passage. This results in a back-pressure on the piston or vane, which constitutes the damping force. In this case, too, the energy absorbed by the damper is converted into heat.

Hydraulic shock absorbers are either single- or double-acting; that is to say, they exert a damping force only during spring rebound, or during both compression and rebound. They can be divided also into direct-acting and arm (or indirect-acting) types. With the direct-acting type the cylinder is pivotally connected to the axle or wheel control link, while the piston is similarly connected to the chassis frame. These connections now are always made through rubber grommets. With the arm type, the part setting the fluid in motion is either formed integral with or actuated from a shaft, and the shaft is provided with an arm that connects to the axle or forms one of the control links of the independently-sprung wheels, while the housing or body of the device is mounted on the chassis frame. In a single-acting shock absorber there is no positive connection between the shaft and the piston. During the rebound period the latter is forced into the cylinder by a crank arm on the shaft, and forces fluid through an orifice or a valve. Then, during the following compression period, the piston is returned by a coil spring in the cylinder.

Double-Acting Shock Absorber—Fig. 35 is a section on the cylinder axis of a Delco-Lovejoy double-acting hydraulic shock absorber of the passenger-car type. The housing is an iron casting and is bored out to receive the pistons *A* and *B*, which are connected by two screws *C* to form a rigid unit. The forged-steel crank arm *D* has a serrated joint with the shaft, which latter has a bearing in the housing. In operation,

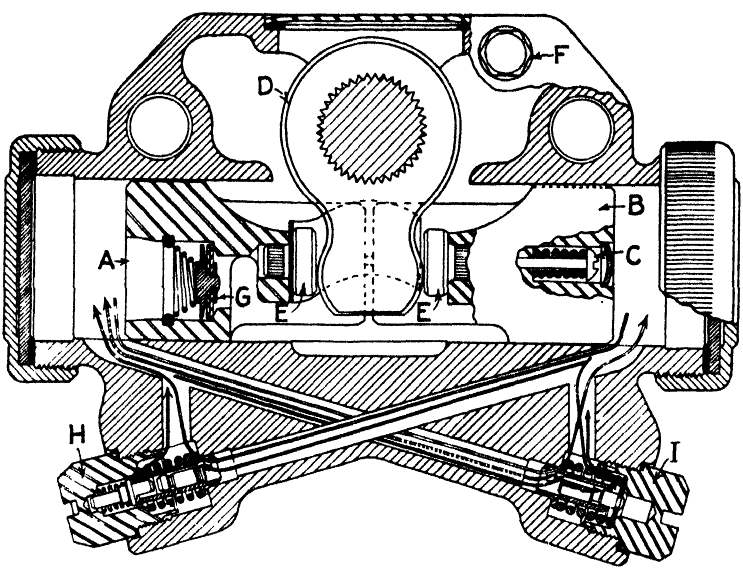
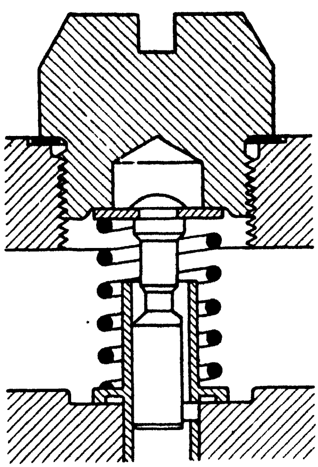


FIG. 35.—DELCO-LOVEJOY DOUBLE-ACTING HYDRAULIC SHOCK ABSORBER.

the interior of the housing is almost completely filled with fluid, and packing glands prevent leakage through the bearing. The crank arm has hardened cam surfaces, which contact hardened, mushroom-shaped members *E* pressed into drill holes in the pistons. Fluid is introduced into the reservoir through a filler opening *F* near the top. It enters the cylinders through intake valves *G* in the pistons, and both ends of the cylinder and the communicating passages between them are normally filled. There is a valve in each of the communicating passages. Valve *H*, below the rebound cylinder, controls the flow from the "compression cylinder," while valve *I* controls the flow from the "rebound cylinder."

When a fluid flows through an orifice, the pressure causing the flow varies as the square of the velocity. In a shock absorber with an orifice of constant size, the velocity of flow varies substantially as the length of stroke, and therefore as the amplitude of the spring motion. From this it may be concluded that the damping force varies as the square of the amplitude. If the orifice were made of such size that a suitable damping force would result at high amplitudes, the damping action at small amplitudes would be insignificant. This would be of no great importance if the springs had considerable interleaf friction, and furnished sufficient damping



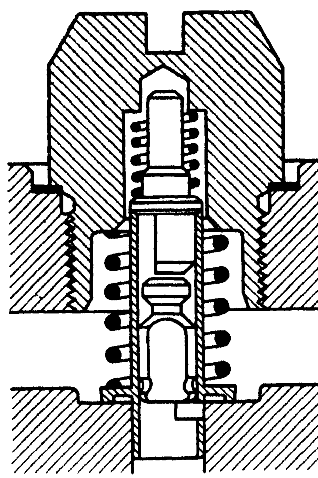


FIG. 36 (*left*).—RELIEF VALVE WITH OPEN ORIFICE.

FIG. 37 (*right*).—RELIEF VALVE WITH ORIFICE CLOSED BY STATIC VALVE.

action for this condition of operation, but chassis springs enclosed in covers and well oiled have only moderate interleaf friction, so that additional damping is required. This can be provided by the shock absorber if the orifice through which the fluid is forced is closed by a valve which opens only after the pressure in the fluid (and therefore the damping effect) has attained a certain value. This valve, referred to as a static valve, eliminates the "free center" feature, by which the damping action is held down to a minimum within a narrow range on both sides of the central position.

To prevent the damping effect from becoming excessive at high amplitudes, the area of the passage for the fluid must be increased, and this is accomplished by providing what is known as a relief valve, which remains closed at small amplitudes (and small pressures), and opens when a predetermined pressure is reached. Usually the relief valve and the static valve form a single assembly.

The two valves *H* and *I* shown in Fig. 35 are of the same general type, but rebound valve *I* has a free orifice, while compression valve *H* has an orifice controlled by a static valve. Enlarged sectional views of the two valves are shown in Figs. 36 and 37. The relief valves are of the poppet type, with hollow stems, and are held to their seats by coil springs. Within the hollow stem there is a stepped or double-diameter plug. One section of the plug fits tightly into the stem, but is flattened off on one side to provide a passage for the fluid. The other section, separated from the first by a "necked" portion, is of smaller diameter, and provides a calibrated annular orifice. In valve *I*, as already stated, this orifice is free, and there is therefore very little resistance to flow under rebound at small amplitudes. In valve *H*, on the other hand, the orifice is closed by a small static valve, also of the poppet type, located in the valve plug. Static valves sometimes are made of disc form, comprising a spring-steel disc which provides its own spring force.

In some shock-absorber models means are provided for adjusting the pressure at which the relief valve opens. This consists merely of a screw supporting the seat for the spring of the relief valve, which can be turned farther in or out by means of a thumb wheel.

In other models there are two flow passages in parallel between the compression and rebound cylinders. One of these has a free orifice, while the other is controlled by a check valve. During the compression period the check valve is open, and fluid then passes through both the orifice and the valve, so that the damping action is light. During rebound the check valve is closed; fluid then can pass only through the orifice, and the damping effect is proportionally stronger.

While in the model illustrated in Fig. 35 the relief valves are located in passages drilled in the shock-absorber body, in other models they are located in the pistons.

Direct-Acting Shock Absorber—With direct-acting shock absorbers the cylinder connects directly to the axle or wheel-control arm, and the piston to the car frame. This eliminates the shock-absorber bearing (which is subjected to heavy loads when there is severe spring play), and the high-pressure contact between the crank arm and the piston. Another advantage claimed for the direct-acting type is that it has considerably more piston displacement, so that for a given damping action the pressure is much lower.

A Monroe direct-acting shock absorber is shown in Fig. 38. It consists of three concentric steel tubes. The innermost of these is the pressure tube, which contains the piston. The second, which holds a supply of fluid, is known as the reservoir tube, while the outer one is a dust shield. Screwed into the upper end of the reservoir tube is the rod-guide-and-seal assembly, through which the piston rod passes, and secured to the upper end of the piston rod are a cap for the dust shield and a ring or eye by means of which the shock absorber is connected to the frame. A similar eye is welded to the closed bottom of the reservoir tube, for connection to the axle or wheel-control arm.

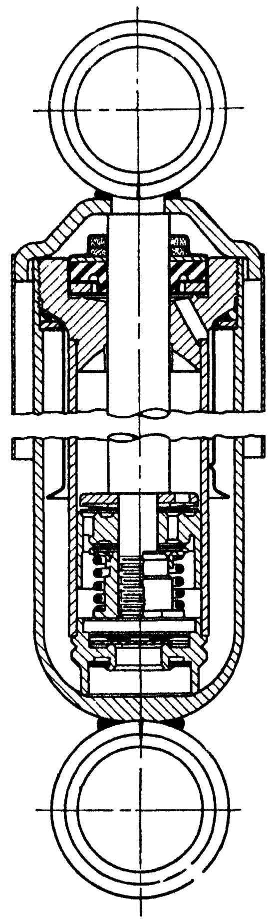


Fig. 38.—Monroe Direct-Acting Shock Absorber.

During the spring-compression period, the piston moves down in the cylinder or pressure tube, and forces fluid through valve-controlled passages in the piston from the lower to the upper end of the cylinder. However, owing to the presence of the piston rod, the space above the piston does not increase as rapidly as the space below decreases, and some of the fluid displaced by the downward motion of the piston is forced into the reservoir tube through an intake-and-compression valve at its lower end. This fluid returns again during the following rebound stroke. In this shock absorber the orifices through which the fluid is forced are closed by valves at the beginning of both strokes, hence there is a damping force from the very beginning of each stroke, and no "free center."

The relation between the spring amplitude and the damping force of a hydraulic shock absorber can be modified by causing the fluid to flow through long passages of relatively small diameter. This results in streamline (instead of turbulent) flow, with which the damping effect is directly proportional to the velocity. However, it is proportional also to the viscosity of the fluid, and a thermostatic control device must be provided to make the damping action fairly independent of atmospheric conditions.

Determining Location of Center of Gravity—In developing suspension systems, as well as in other connections, it sometimes is desirable to know the location of the center of gravity of the vehicle. To locate the center of gravity completely, it is necessary to determine its height above floor level and its horizontal distances from the longitudinal center plane and from either the front or rear axle. If a scale of the type used by highway officials to determine wheel loads is available, the load on each wheel can be readily determined, and from these values the location of the center of gravity in the horizontal plane can be found. Suppose, for instance, that the load on the two front wheels is W_1 and that on the two rear wheels W_2; then, if the wheelbase is a in., the center of gravity will be $aW_2/(W_1 + W_2)$ in. from the center of the front axle. Similarly, if the load on the two left-hand wheels is W_3, that on the right-hand wheels W_4, and the wheel tread at both front and rear b in., then the distance of the center of gravity from the center line of ground contact of the right-hand wheels will be $bW_3/(W_3 + W_4)$ in. The distance from the center line of ground contact to the longitudinal axis of the car is $b/2$, hence the distance of the center of gravity from the longitudinal axis of the car is

$$\frac{b}{2} - \frac{bW_3}{W_3 + W_4} = \frac{b}{2}\left(\frac{W_4 - W_3}{W_4 + W_3}\right) \text{ in.}$$

In the past the height of the center of gravity above floor level generally has been determined by tilting the vehicle either sideways or lengthwise. If the vehicle is tilted sideways until it balances on two wheels, its center of gravity evidently is directly above the line connecting the center points of ground contact of the two tires, and its height can then be calculated from the angle of inclination, the tread, and the rolling radius of the tire cross section. However, no very accurate results can be expected from this method, because of the difficulty of measuring the balancing angle, and the fact that when the car is tilted sideways the tire section distorts. Another method consists in raising one end so that the car as a whole is inclined lengthwise at a certain angle, and then determining the weight on the axle which has been raised. This method also is likely to yield results that leave something to be desired from the standpoint of accuracy Before a car is tilted either sideways or lengthwise, the springs must be blocked and the frame strapped to the axle or lower control bar, so there can be no relative movement of the sprung and unsprung masses.

A pendulum method of determining the height of the center of gravity has been worked out at General Motors Proving Grounds, and is claimed to give quite accurate results

FIG 39—DETERMINING HEIGHT OF CENTER OF GRAVITY IN A SWING

As shown in Fig. 39, the car is placed in a swing comprising a floor made of two 15-in. channels rigidly held together by I-beam cross bracing and draw rods, which is suspended on knife edges by means of four steel straps clamped to a cross shaft, on the ends of which the knife edges are formed. The knife edges are supported by saddles clamped to a horizontal I-beam at the top of the structure supporting the swing. Two such swings, of different length, are required, and the distance between floor and axis of support is made about twice as long in one as in the other.

The car is placed on the channels and adjusted until the combination is level. Then the brakes are set and the "pendulum" is set in motion. The time of a complete oscillation is determined by taking the time of from 500 to 1000 complete swings with a stop watch. As the equation for the period of the pendulum applies only if the oscillation is so small that the difference between the sine and the tangent of one-half the oscillating angle is negligible, the amplitude of oscillation is limited to 2°.

The swing with the car on it forms a compound pendulum to which the following equation applies:

$$I = \frac{t^2 Wl}{4\pi^2} \text{ lb-ft-sec}^2 = \frac{gt^2 Wl}{4\pi^2} \text{ lb-ft}^2$$

where I is the moment of inertia around the axis of the swing; t, the time of a complete oscillation, in seconds; W, the weight of the swinging mass, in lb, and l, the length of the pendulum, or the distance between the axis of oscillation and the center of gravity of the swinging mass, in ft. The above equation applies strictly only to a pendulum swinging in vacuum without damping, and if the conditions are different, suitable corrections must be made. While there is a slight damping effect when the car is being swung on knife edges as described, it has been found that its effect on the results of the test is negligible. On the other hand, the effects of the atmosphere cannot be neglected. These effects are of a threefold nature. In the first place, owing to the buoyancy effect, the true weight of the swinging mass is greater than that shown by the scales. Secondly, the air trapped in the car swings with it, and, thirdly, the swinging car sets some of the ambient air in motion, thereby producing an additional mass effect. In determining the moment of inertia of the mass around its axis of oscillation, use is made of the "virtual weight" W, as shown by the scales. When the car is being swung, it is completely filled with air at atmospheric density. The weight of this air is $V_s\rho$, where V_s is the interior volume of the car, in cu ft, and ρ, the density of the air, in lb per cu ft. If the total (outside) volume of the car is denoted by V, the volume subjected to the buoyancy effect is $V - V_s$, and the true weight of the swinging mass is

$$W_s = W + V_s\rho + (V - V_s)\rho = W + V\rho.$$

The "additional mass" effect produced by the action of the car on the outside air has been investigated by the National Advisory Committee for Aeronautics, which recommends the use of an additional mass term given by the equation

$$M_A = \frac{K}{4}\rho\pi c^2 b,$$

where K is a constant depending on the dimensions of the swinging structure; c, the depth of the structure, and b, its width. This additional mass term is small in the case of an automobile, and can be neglected without producing a material error in the final result.

In determining the height of the center of gravity by this method, the car is swung in each swing, and the periods of the complete oscillations are noted Then an equation is set up for I, the moment of inertia of the car around its center of oscillation, in terms of the measurements of each swing and the time of oscillation observed with it. We thus get two expressions for the same entity, both containing the unknown l, the distance between the axis of oscillation and the center of gravity of the car. Since the two expressions must have equal values, we can equate them and then solve for l.

Now let W_3 be the weight of the car and swing; W_2, the weight of the swing; W_1, the corrected weight of the car $(W + V\rho + (K/4)\rho\pi c^2 b)$; l_3, the distance from the axis of oscillation to the center of gravity of car and swing; l_2, the distance from the axis of oscillation to the center of gravity of the swing; l_1, the distance from the axis of oscillation to the center of gravity of the car; t_3, the time of a complete oscillation of car and swing, and t_2, the time of a complete oscillation of the swing alone. We then get for the moment of inertia of the car around its axis of oscillation,

$$I = \frac{g}{4\pi^2} W_3 l_3 t_3^2 - \frac{g}{4\pi^2} W_2 l_2 t_2^2 - W_1 l_1^2.$$

By now replacing $g/4\pi^2$ by K and $V\rho + (K/4)\rho\pi c^2 b$ by A, this equation can be transformed to read

$$I = KW_2 l_2(t_3^2 - t_2^2) + Wl_1(Kt_3^2 - l_1) - Al_1^2.$$

We may now replace l_1 by $B - h$, where B is the distance from the axis of oscillation to the floor of the swing, and h, the height of the center of gravity above floor level. The equation then becomes

$$I = KW_2 l_2(t_3^2 - t_2^2) + W(B - h)(Kt_3^2 - B + h) - A(B - h)^2.$$

All of the values of the right-hand side of the above equation, with the exception of h, can be measured. Values for the short swing and for the long swing are inserted in this equation for I, the two expressions are equated, and the equation is solved for h, the height of the center of gravity of the car above floor level.

This "pendulum" method also permits of determining the radius of gyration of the car around a transverse axis through its center of gravity. After h has been determined as explained in the foregoing, its value is inserted in one of the equations for I, and the moment of inertia of the car around its axis of oscillation thus determined. The moment of inertia I_0 of the car around a transverse axis through its center of gravity is equal to the difference between the value thus found and the product $W_s l_1^2$, where W_s is the true weight of the car, that is, the sum of its scale weight and the weight of the entrapped and displaced air.

$$I_0 = I - W_s l_1^2.$$

After the moment of inertia of the car around the transverse axis has thus been found, the square of the radius of gyration is obtained by dividing it by the weight of the car.

It may be pointed out that the radius of gyration thus determined is that of the complete car, not that of the sprung mass, which latter is frequently referred to in discussions on riding qualities.

It is claimed that by this method the moment of inertia of an automobile can be determined within ± 1 per cent, and the height of the center of gravity within ± 0.25 in. To ensure this degree of accuracy, the knife edges must be supported quite rigidly, and their saddles must be fixed in such a manner that there is no measurable motion.

Determining Radius of Gyration—A simple method of determining the radius of gyration of the sprung mass experimentally has been described by Arthur R. Parilla. It is based on the principle that the moment of inertia of a body around an axis parallel to an axis through its center of gravity is equal to the sum of its moment of inertia around the latter axis and the product of its mass by the square of the distance between the two axes; and on the fact that when k^2, the square of the radius of gyration, is equal to the product ab, where a and b are the distances between the center of gravity and the axes of the front and rear axle, respectively, the conjugate axes around which the sprung mass oscillates are directly over the axles.

Referring to Fig. 40, outriggers are secured to the frame side rails at front and rear, and a cross bar is secured to each pair of outriggers, both of the cross bars at the same distance from the center of gravity of the sprung mass, which must be determined in advance. Equal numbers of cylindrical discs are then secured to both cross bars. Since both of the cross bars are at the same distance from the center of gravity, the addition of the discs will not change the location of the latter, but it will increase the radius of gyration. Discs are added until the square of the radius of gyration is equal to the product *ab*. Since the axis of pitch is then directly over the rear-axle axis, if the front wheels are raised off the floor and then dropped, there will be no motion of the sprung mass toward or away from the rear axle. A chronograph is used to determine when this condition exists. The square of the radius of gyration of the sprung mass of the vehicle then is given by the equation

$$k^2 = ab + (2m/M)(ab - d^2) \text{ in.}^2,$$

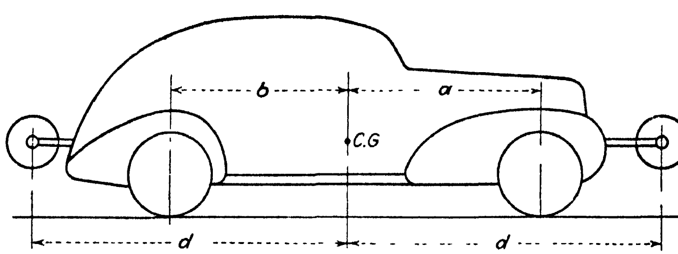


FIG. 40.—CAR PROVIDED WITH FRONT AND REAR OUTRIGGERS TO DETERMINE RADIUS OF GYRATION OF SPRUNG MASS.

where M is the sprung mass of the vehicle (W/g); m, the mass of the weights added at one end of the vehicle, and d, the distance of the center of the discs at either front or rear from the center of gravity.

It is recommended that when the determination is made, the spring damping forces be reduced to a minimum by disconnecting the shock absorbers.

The method of determining the height of the center of gravity by tilting the car sideways, already referred to, is illustrated in Fig. 41. It is assumed that the distance AB of the center of gravity from the center plane of the right-hand wheel has been determined in advance, by means of weighing scales. Then, if θ is the angle of tilt at which the car balances, the height of the center of gravity is $(AB/\tan\,\theta) + c$ in., where c is the distance from the ground to the axis of the tire section around which the car rolls when being tilted.

Progress in Spring Engineering—The following figures, relating to Cadillac practice, give a good idea of the progress made in passenger-car springs since the problem of a more comfortable ride was first seriously tackled, about 1930. Between 1930 and 1940 the weight of the rear springs was reduced from 1.25 lb to 0.75 lb per 100 lb of curb weight. During the same period the amount of energy elastically stored per pound of spring steel rose from 85 to 205 in.-lb. Coil springs for the front were adopted in 1934. The amount of energy stored by these springs rose from 265 in.-lb per pound in 1934 to 380 in.-lb in 1940. During this period the

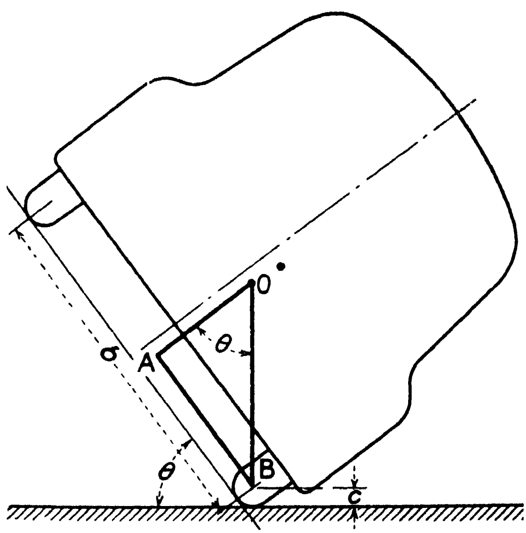


Fig. 41.—Determining Height of Center of Gravity by Tilting Car Sideways.

curb weight of the car remained close to 4000 lb, and the weight of the springs close to 13 lb, hence the greater amount of energy stored in 1940 resulted almost entirely from increased deflections. The steps taken to achieve these improvements included the adoption of grooved-section spring plates, a larger number of thinner plates, heat treatment in controlled-atmosphere furnaces, shot-peening of spring leaves, rubber pads at spring seats, grease-filled spring covers, and interliners.

CHAPTER XVIII

Independent Suspensions

The term "independent suspension" describes a method of elastically supporting the chassis on the road wheels without the intermediary of rigid axles When a pair of wheels are mounted on a rigid axle, and one of them passes over a road obstacle, the axle executes a small angular movement in the vertical plane, and both of the wheels simultaneously perform angular movements of exactly the same magnitude. With independent suspension the movements of the two wheels are not thus interdependent; any movement of one wheel may —and most likely will— be accompanied by a similar movement of the other, but the two will not be equal.

Various advantages have been claimed for independent suspension, but some of these are realizable only with certain special designs, and therefore are not inherent in the principle. Since all types eliminate the rigid axle, probably all reduce the unsprung mass, and it was undoubtedly the desire to secure the advantages inherent in low unsprung weight that chiefly motivated most of the early pioneers of the system, who generally provided independent suspension at both front and rear. When the system was taken up by large manufacturers in this country, during the early thirties, front-end instability (shimmy and tramp) had become a serious problem, and independent suspension seemed to offer the means for a satisfactory solution. It had been established that tramp and shimmy are essentially gyroscopic effects. The rotating front wheels of a moving car form gyroscopes, and if their planes of rotation are forcibly changed, as by road irregularities, wheel unbalance, etc., a forced gyroscopic precession occurs; that is, the wheels will swing, or tend to swing, around the knuckle pins. With independent suspension, if one wheel passes over an obstacle, the two wheels do not simultaneously change their planes of rotation equally, as they do when they are connected by a rigid axle. With certain types of independent suspension the up-and-down motion is even parallel, or nearly so, so that there is practically no change in the plane of rotation, and gyroscopic effects therefore are absent, or at least greatly reduced. Thus independent front suspension has a definite tendency to suppress front-end vibration. It is, however, not an unfailing cure or preventative. Certain makes of car with front independent suspension have been known to develop a pronounced tendency to shimmy after some years of service, probably because the front end of the chassis-body assembly lacked adequate stiffness.

The introduction of front independent suspension on passenger cars had a very favorable effect on riding comfort, in that it made possible the use of much softer front springs (greater static deflection), as compared with cars with conventional springing, in which the springs serve also as guiding or control members for the front axle. Owing to their flexibility, the springs cannot be regarded as positive guiding members. To ensure fairly positive steering, variations in the paths of the steering knuckles relative to the frame must be held within narrow limits, and this can be done only if the springs are relatively stiff. But stiff springs mean poor riding qualities. With independently sprung front wheels, special guiding members, in the form of rigid radius rods or links, are usually provided, and it is then permissible to use softer springs, thus ensuring a more comfortable ride. Equally soft front springs could be used on cars with rigid front axles, if separate, positive guiding means were provided, and a certain amount of development has taken place along that line. The fact remains that positive guiding of front wheels and flexible front springs were introduced together with independent suspension, and are generally considered characteristic of it. Independent springing does not call for any particular type of spring, and both conventionally sprung and independently sprung vehicles may be equipped with leaf, coil, or torsion-bar springs, or with the relatively rare rubber or air springs.

Classification—There are quite a number of different types or systems of independent springing. In one system (*A*, Fig. 1), the steering knuckles are arranged to slide on or in vertical or nearly vertical guides secured to the chassis frame. In another (*B*, Fig. 1) the steering heads or knuckle supports are carried at the ends of transverse semi-elliptic or quarter-elliptic springs, without other connection to the frame. Instead of springs alone, one spring and one pair of links parallel with it may be used to support the steering head (*D*, Fig. 1), the links in that case being made of the "wishbone" type and pivoted to the frame at points a considerable distance apart, to increase their ability to take driving thrust and brake torque. In the "parallel-link" system (*C*, Fig. 1), which is the one now in most general use, the steering heads are carried on superposed parallel links or control arms, usually of "wishbone" form, the spring being interposed between the frame and the lower link.

In systems *B*, *C*, and *D* the steering heads under spring action move in a transverse, substantially vertical plane. It is possible also to mount them directly on the chassis frame. As shown at *E*, Fig. 1, the knuckle then is provided with an arm in the form of a housing extending to the rear, and in the outer end of this arm or housing is journaled a crank, the

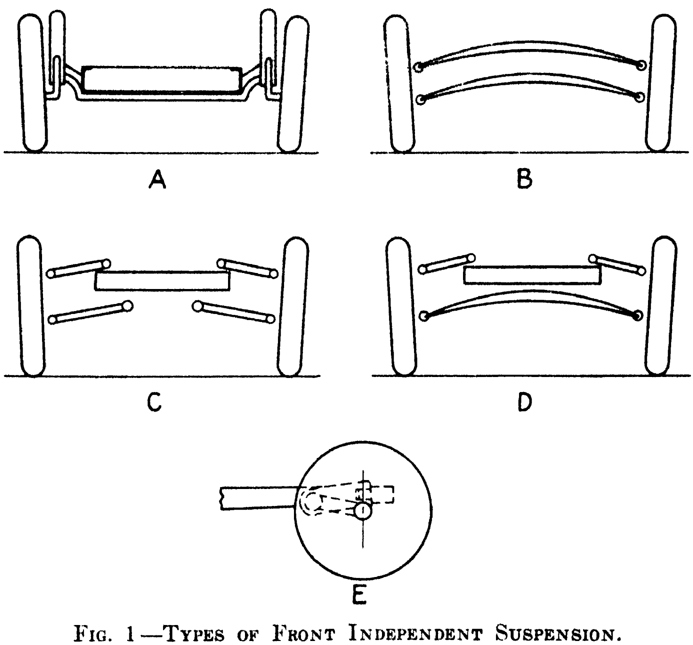


FIG. 1—TYPES OF FRONT INDEPENDENT SUSPENSION.

arm of which extends forward, and the pin of which forms the wheel spindle. A spring is interposed between the crank and the housing, and transfers the torque imposed on the crank by the reaction to the wheel load, to the housing, whence it is transferred to the frame by the knuckle pin. Instead of coil springs, torsion bars may be used in systems *C* and *E*. In the former system the bars would extend lengthwise of the frame and have the inner ends of the lower links rigidly secured to them, while in the latter they would extend across the frame and carry the trailing arms at their free ends.

One end of each torsion bar would be anchored to the frame.

The foregoing covers all of the principal types of independent springing for front wheels (steering wheels). Independent suspension is used less extensively for rear wheels. Of the three possible advantages of independent springing mentioned in the foregoing, only the first, reduction of unsprung weight, would apply in the case of rear-springing. As the driving-gear housing would be mounted on the frame, the system would make it possible to slightly reduce car height. If rigid half-axles are to be used with independently sprung wheels, they must oscillate around longitudinal axes, which means that they either must be driven independently in such a way that the driven member of the drive gear can oscillate around the axis of the driving member, or they must be provided with universal joints. As the universals would have to take the full axle torque, they would have to be of much larger size than those in the propeller shaft of a conventional car. It is questionable whether the possible reduction in unsprung weight warrants these complications, and independent suspension of rear wheels so far has not found a place in American passenger-car practice. It has been extensively used in Europe, however.

Rear-Wheel Independent Suspension—A number of possible arrangements of rear-wheel independent suspension are shown diagrammatically in Fig. 2. Where the driving axles pivot around the longitudinal axis of the chassis, as shown in sketch *A*, the differential gear sometimes is mounted on the propeller shaft instead of on the axle, and its two side gears drive bevel pinions which mesh with bevel gears on the two axle shafts, respectively. To avoid interference, the pinions have to be made of different diameters, and the gears likewise, but the ratios of both sets, of course, must be the same. Each axle tube at its inner end is provided with a cylindrical segment of less than 180 deg., lodged in an annular space between a bore of the axle housing and a bearing mount seated therein, which allows the axle to oscillate freely. The axles, of course, must pass through openings in the driving gear housing, of sufficient size to permit their oscillation under spring play, and the problem of closing these housings so that no lubricant can escape is a rather difficult one.

With short driving axles (*B*, Fig. 2) the differential and driving gear are enclosed in a central housing in the usual way, this housing being supported on the frame. The axle tubes are then pivotally mounted on opposite sides of the housing, and the axle shafts provided with universal joints concentric with the pivotal mountings. From this point out-

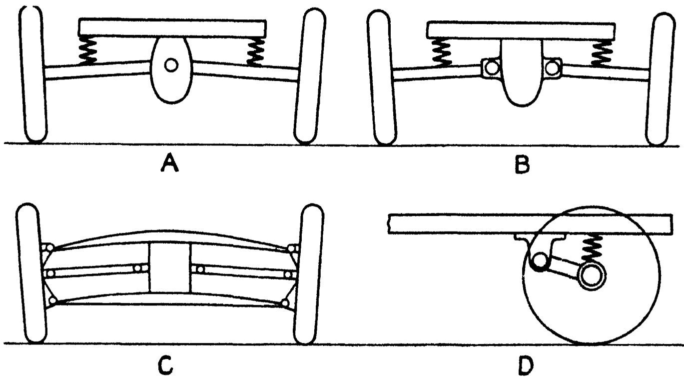


FIG. 2.—TYPES OF REAR INDEPENDENT SUSPENSION.

ward the axles may be rigid, in which case the wheels oscillate with the axles, and the width of the tread varies with spring play. On the other hand, the wheels may be mounted on stub axles extending through bearing shells that are supported from the frame by superposed transverse springs, parallel links, etc. (*C*, Fig. 2). In that case, axle tubes are dispensed with, the axle shafts being exposed and having universal joints at both ends. This, however, is a rather primitive design that has little appeal at the present time. Finally, bearing shells for the stub shafts of the driving wheels may be supported by crank arms journaled on the frame (*D*, Fig. 2). While the sketch shows a coil spring, torsion-bar springs are commonly used in system *D*.

An objection sometimes raised against oscillating axles is that, as shown in Fig. 3, the track or wheel tread changes with spring deflection, which tends to cause scuffing and

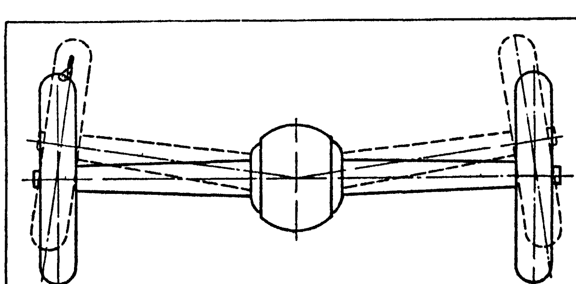


FIG. 3.—SHOWING CHANGES OF TREAD WITH SWINGING AXLES.

increased tire wear. But it has been shown by de Lavaud and others that the slip angle between the rolling tire and the ground is relatively small, and well within the capacity of the tire to take care of elastically at speeds above 15 mph. With oscillating axles, increased tire wear due to scuffing occurs chiefly under conditions of light wheel load, as in unloaded trucks, when the large positive camber causes "oversteering."

An important advantage of independent suspension for driving wheels—front or rear—is that a car equipped with it can negotiate deeply rutted dirt roads on which the final-drive housing drags on the central ridge. On such roads a car with a rigid driving axle would be "hung up" on its center housing.

History—The origin of independent suspension dates back to the early years of the automobile industry. As far as is known, the first car to have it was the French Sizaire-Naudin of 1905. In this model a part similar in shape to an I-section straight front axle formed the front cross member of the frame. It had vertical bosses at both ends, in which the vertical parts of L-shaped steering knuckles were adapted to slide. Chassis weight was transferred to the knuckles through a transverse, inverted, semi-elliptic spring, the ends of which rested on top of the sliding knuckles. The Sizaire-Naudin was a very simple, low-priced car (it had only 420 separate parts), and it is quite possible that the only object the designer had in view when laying out this suspension system was to lower the manufacturing cost, as the single front spring and the simple steering-knuckle design certainly were conducive to economical production.

What first really drew attention to the merits of independent front suspension was its use on the Italian Lancia car, brought out in 1922. In this the steering knuckles also slid up and down in bosses or guides secured to the chassis frame, but enclosed coil springs were used, together with hydraulic shock absorbers, and the whole design (being of much later date) was much more refined than that of the Sizaire-Naudin, all sliding surfaces being enclosed. It is quite possible that this design combined all three of the advantages of front independent springing, and the excellent riding qualities of the car were widely commented upon.

A sectional view of an improved design of Lancia front independent suspension is shown in Fig. 4. The weight of the chassis is transferred to the steering knuckle by a coil spring on each side, enclosed in a cylindrical guide made in two parts and closed at top and bottom. The lower part is

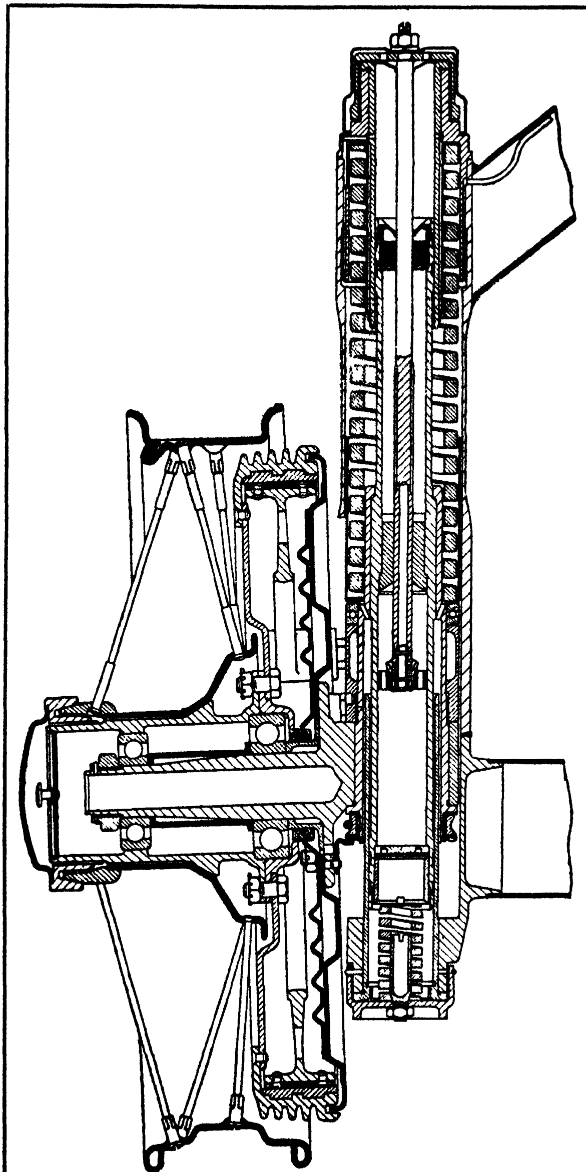


Fig. 4.—Lancia Front Independent Suspension.

brazed or welded to the end of a tube forming a frame cross member, while the upper part is secured to the radiator shell by a tubular brace. The steering knuckle swings around a tubular knuckle pin, which also forms the cylinder of a hydraulic shock absorber. The piston rod of the shock absorber is secured to the cap of the guide, and the piston, therefore, is stationary, while the cylinder moves up and down with the knuckle. In the lower part of the guide there is an additional coil spring, which cushions the rebound.

With a full complement of passengers the Lancia weighed 5750 lb, of which 2270 lb was on the front wheels. Each of the front main springs weighed only 3.12 lb. The maximum deflection was about 4 in., and this subjected the spring material to a stress of about 125,000 psi. The ratio of sprung to unsprung weight at the front was 6.25 without load, and 7.12 with a full load, these ratios being substantially twice as great as with conventional springing. Experience with this model showed that great frame rigidity is required in a car with front independent suspension, if excessive front-end shake and body squeaks and rattles are to be prevented. When the necessary stiffness was provided, by channel inserts in the side members and X-type and tubular cross members, the car rode very smoothly.

The first attempts to introduce independent suspension in this country, immediately after World War I, proved abortive, the companies sponsoring the move not reaching the manufacturing stage. Both General Motors and Chrysler Corporation adopted front independent suspension during the early thirties, some of the General Motors and all of the Chrysler lines employing the so-called parallel-link type, with coil springs. Two General Motors lines adopted the Dubonnet type, in which the wheel spindle is secured to an arm which under spring action swings in a plane parallel with the center plane of the wheel. Some years later this type was discontinued, and the parallel-link type adopted for these G.M. lines also.

Drawings of the parallel-link type front suspension as used on Buick cars are shown in Fig. 5. A box-section frame front cross member is used, which is convex toward the front. Mounted on top of this cross member near its ends are hydraulic shock absorbers, whose double arms form the upper links for the steering heads. The wishbone-type lower link, which is approximately twice as long as the upper one, is swiveled on a shaft that is clamped and doweled to the under side of the box-section frame cross member. Pressed-steel spring rests for the coil springs are riveted to the lower

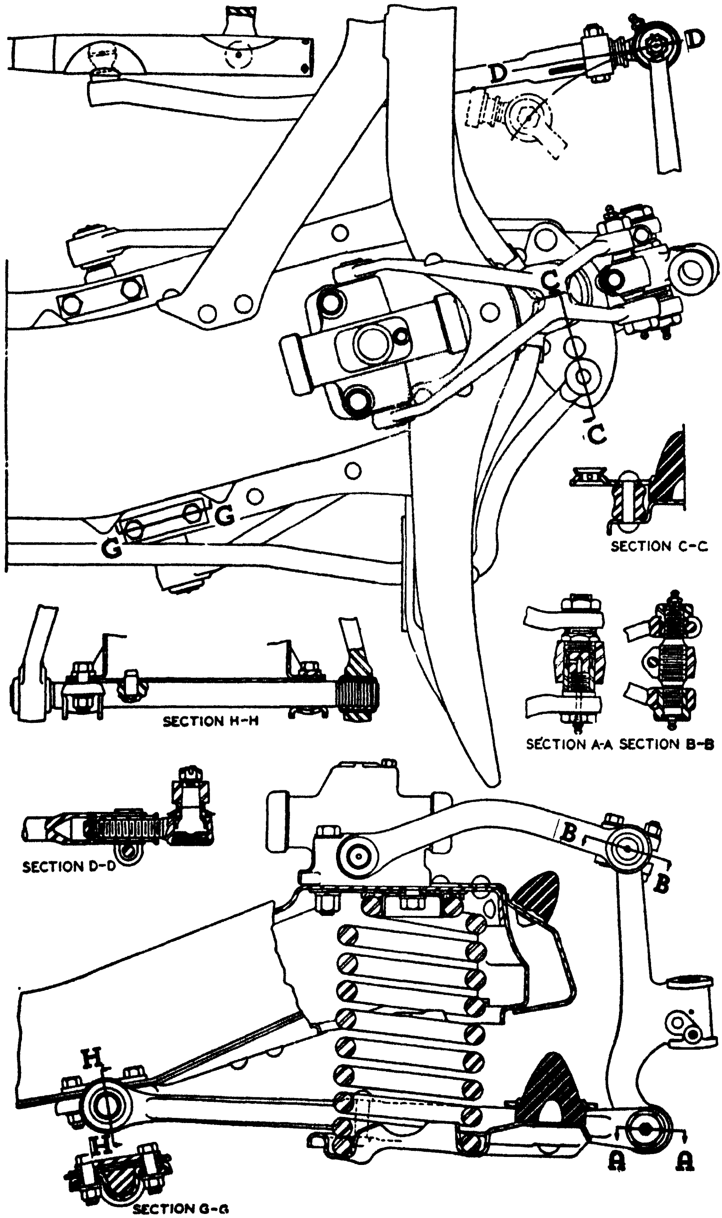


FIG. 5.—BUICK PARALLEL-LINK TYPE OF FRONT INDEPENDENT SUSPENSION.

links. There is a circular opening in the under side of the box-section cross member near each end, through which the coil springs are inserted, the springs seating against the top wall of the cross member, which is reinforced at these points by the side rails and the shock-absorber housings.

It will be noticed that the axes of the control arms make a considerable angle with the longitudinal axis of the car, and that the "wishbones" have a backward sweep in the plan view. This is merely a design convenience, which makes for a better layout of engine, frame, radiator and steering linkage, and particularly facilitates placing the powerplant

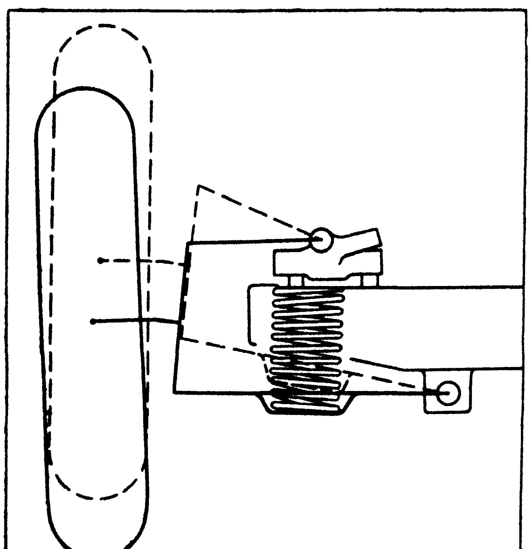


Fig. 6.—Diagram Showing That with Parallel-Link Type the Tread Remains Constant.

well forward, to increase the moment of inertia of the spring-supported mass around a transverse axis through its center of gravity. The upper links or control arms are of about one-half the length of the lower ones, which makes it possible to place the radiator in a low position between the front shock absorbers, and also reduces the tilt of the outer wheel on curves, as compared with a true parallel-link suspension. The arrangement also obviates the need for pivotally mounting the lower supports for the coil springs, fits in well with the general design of the frame side rail and cross member, and saves weight and cost. As may be seen from the lower view in Fig. 5, the two control arms are not parallel, their center lines approaching each other in the direction toward the center of the chassis, which is another design convenience.

The Constant-Tread Feature—The chief reason for making the lower links substantially twice as long as the upper ones is that this keeps the tread—the distance between center points of tire-ground contact on opposite sides—constant, thus avoiding a scuffing or scrubbing action on the tire treads. This is made clear by the diagram Fig. 6. If both links were of the same length, the tread would be a maximum when the links were in the horizontal position. As the springs deflected more, and the links assumed a position inclined toward the horizontal, the wheels would move inward parallel with their axes, and the tread would be reduced. With the lower links

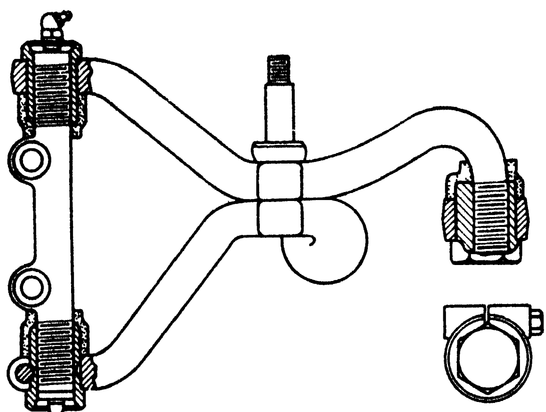


FIG. 7.—MEANS FOR ADJUSTING CAMBER AND CASTER (CHRYSLER).

longer than the upper ones, as shown in the diagram, the motion of the wheels consequent upon a departure of the links from the horizontal position is not parallel to their axes, but of a pivotal nature, around the center point of ground contact. If the wheels are vertical when the links are horizontal, the former will be leaning inward at the top when the links are either above or below the horizontal position.

Camber and Caster Adjustment—While usually almost no camber and very little caster is provided in modern cars, it is generally considered necessary to provide means for adjusting both, as the original factory setting may be disturbed by accidents. The means of adjustment naturally must be different from those in cars with front axles. Fig. 7 shows the upper control arm of a Chrysler car with front suspension of the parallel-link type. The arm is Y-shaped, having threaded bearings on a shaft bolted to the frame. The stud extending up (in the drawing) from the middle of the control arm is for attachment of a shock absorber of the direct-acting type. At its outer end the control arm is bent at right angles and threaded; an eccentric, internally-threaded bushing is screwed over it, and is clamped in a boss at the upper end of the steering head. The bushing is provided with a hexagon head, and can be readily turned by means of a wrench. Caster adjustment is made by turning the bushing so as to move it axially on the threaded part of the control rod in one direction or the other. If the caster is to be adjusted and the camber left as it is, the bushing must be given a full number of turns. If the camber is to be adjusted, the bushing is turned through whatever angle is required, one-half a turn corresponding to the full range of adjustment. After adjustment has been made, it is locked by the clamp bolt in the knuckle support.

Modified Dubonnet Front Suspension—Vauxhall Motors, Ltd., in England have been equipping their cars with a modified version of the Dubonnet independent front suspension, which in an earlier form at one time was used on Chevrolet and Pontiac cars in this country, and on Vauxhall and Opel cars (built by General Motors foreign subsidiaries) abroad. The more recent design is illustrated in Figs. 8 and 9. The frame of the car has a tubular front cross member, the ends of which are upwardly and backwardly inclined, and terminate in the steering heads. Pivoted on each steering head is a cylindrical housing extending toward the rear. A crank mounted in needle bearings near the outer end of the housing extends forward substantially parallel with the housing, its crankpin forming the wheel spindle. Spring action is provided by a torsion bar and a concentric torsion tube, the bar being secured to the crank and the tube by serrated joints, and the tube to the housing by a flanged-and-bolted joint. That portion of the crank "main journal" between the needle bearings also is serrated, and has a three-armed spider secured to it. Two arms of the spider actuate the plungers of the hydraulic shock absorber, while the third acts on a coil spring, through the intermediary of a button-shaped spring rest. With the car under normal load, the coil spring is under its maximum load of about 2100 lb. The coil spring and spider arm together produce a toggle effect, softening the torsion spring within the range of deflections corresponding to a boulevard ride. For larger deflections in either compression or rebound, the coil spring stiffens the suspension, the same as bump and rebound stops. Both softening of the suspension in the mid-position and stiffening in the outward positions are effected by the same spring. One advantage of this

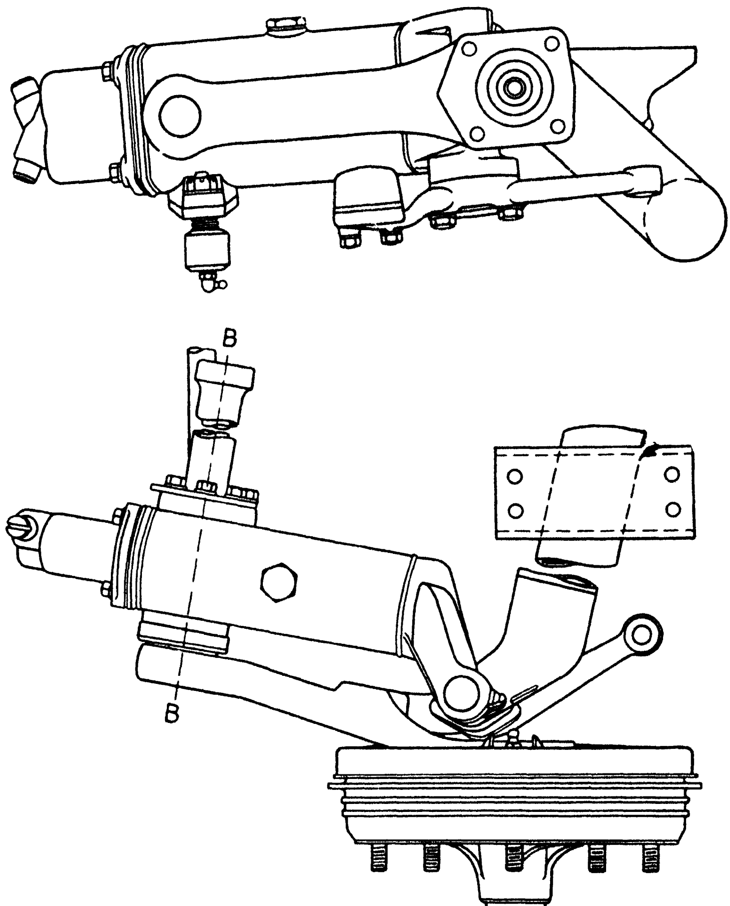


FIG. 8.—VAUXHALL FRONT INDEPENDENT SUSPENSION, SIDE VIEW AND PLAN.

system is that, as the steering head is carried on the frame, spring action does not change the relative levels of opposite ends of the tie rods, hence the steering geometry can be worked out quite accurately.

In the earlier Dubonnet, as formerly fitted to Chevrolet and Pontiac cars, a coil spring was used instead of the torsion bar and tube. It was enclosed in the pivoted housing and acted against an arm of the crankshaft, which compressed it under the influence of chassis weight. A separate link extend-

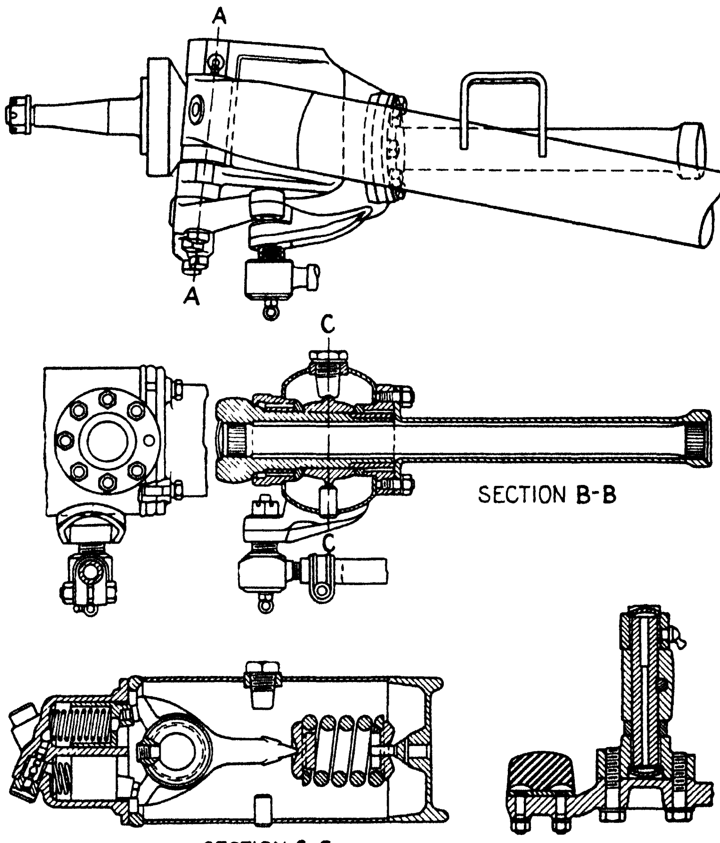


FIG. 9.—VAUXHALL FRONT INDEPENDENT SUSPENSION, FRONT VIEW AND DETAILS.

ing from the outer end of the spring housing to the brake backing plate took care of braking torque.

Armstrong-Siddeley Front Suspension—Following the end of World War II there was a strong move toward front independent suspensions on torsion bars in Europe, and particularly in Great Britain. In 1950 twelve old-established European makes, including eight British, had such suspensions. A good example of British practice is furnished by the Armstrong-Siddeley, illustrated in Fig. 10. In this design the usual steering head or knuckle support is dispensed with, the knuckle being supported directly in spherical bearings at the outer ends of the parallel links. The upper link forms the arm of a hydraulic shock absorber bolted to the top of the frame forward cross member, while the lower link is

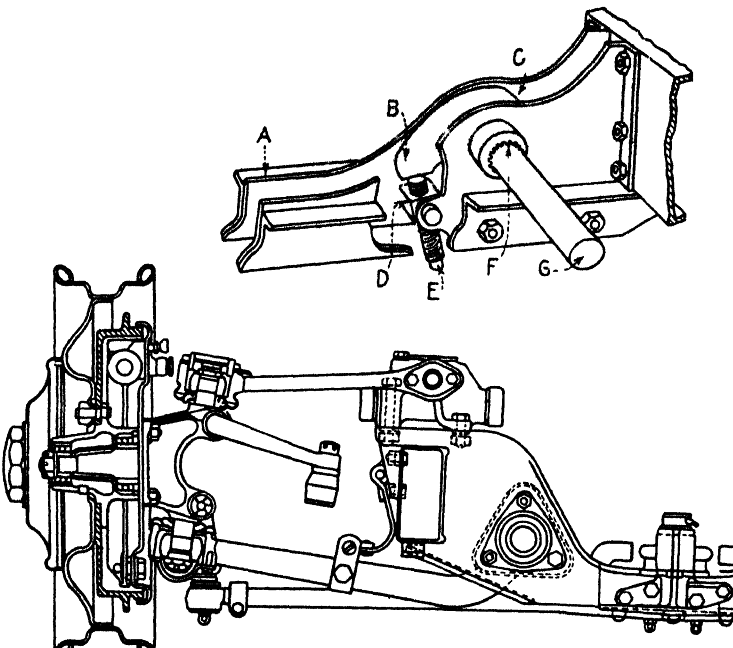


Fig. 10.—Armstrong-Siddeley Front Independent Suspension on Torsion Bars

splined to a shaft with splined sleeves on opposite sides, these sleeves being supported in rubber bushings in bearing brackets bolted to opposite sides of the box-section frame cross member. The rear splined sleeve engages the forward splined end of the torsion bar. At the rear end the torsion bar is splined to a lever located between two gusset plates *C* on a frame cross member *A*. An adjusting screw *E* passing through a trunnion *D* makes it possible to adjust the rear end of the torsion bar angularly, thereby raising or lowering the front end of the chassis.

Jowett Front Suspension—Front independent suspension on torsion bars is employed also on the Jowett, a rather small British car of 91 cu in. piston displacement, 102-in. wheelbase, and 2168 lb shipping weight. The Jowett front suspension is illustrated in Fig. 11. Shock absorbers of the direct-acting type are used and extend from a point near the outer end of the lower to a point near the inner end of the upper link. The ends of the torsion bars are upset and machined to octagonal form. All bearings in the suspension are rubber-bushed. A rubber bumper on the bearing bracket for the lower link limits the rebound.

The Jowett front torsion bar has a diameter of 0.88 in. and a length of 36.25 in., and the lower link is 15 in. long between centers. The deflection of the front springs under static load is 7 in., and the frequency of front-end vibration is 71 cycles per minute.

Volkswagen Front Suspension—The front suspension of the German Volkswagen (Fig. 12) is an improved version of the Porsche torsion-bar front suspension referred to in Chapter XVI. The Volkswagen has a frame of the backbone type, and the whole front assembly, including a pair of cross tubes, torsion bars, trailing links, steering heads, wheels, and steering gear, can be removed as a unit, being held to the backbone by means of four bolts. Each cross tube contains a torsion bar made up of four flat steel strips which together have a square cross section of 18 mm. The cross tubes are spaced by four welded-on pressed-steel brackets, of which the inner ones serve to hold them to the backbone, while the outer ones carry bumpers to limit the up-and-down motion. At the centers of the tubes the torsion bars are surrounded by collars which are held in position by indentations in the tubes and by set screws. The suspension links, which are drop-forgings, are provided with relatively long hubs, broached out square to take the ends of the torsion bars, and supported in bearings of plastic material in the tubes. Steering knuckles are of the inverse Elliott type and have the steering arms forged integral with them. Direct-acting shock absorbers are interposed between the upper suspension links and the bracket on the tubes. While a pack of four flat bars is not as efficient as a circular bar, it is cheap to manufacture, and its advantages from the production standpoint are considered to outweigh the theoretical disadvantage of less energy-storing capacity per pound of steel.

Rear Independent Suspension—In passenger cars independent suspension is not used as much at the rear as at the front. Previous to World War II rear independent suspen-

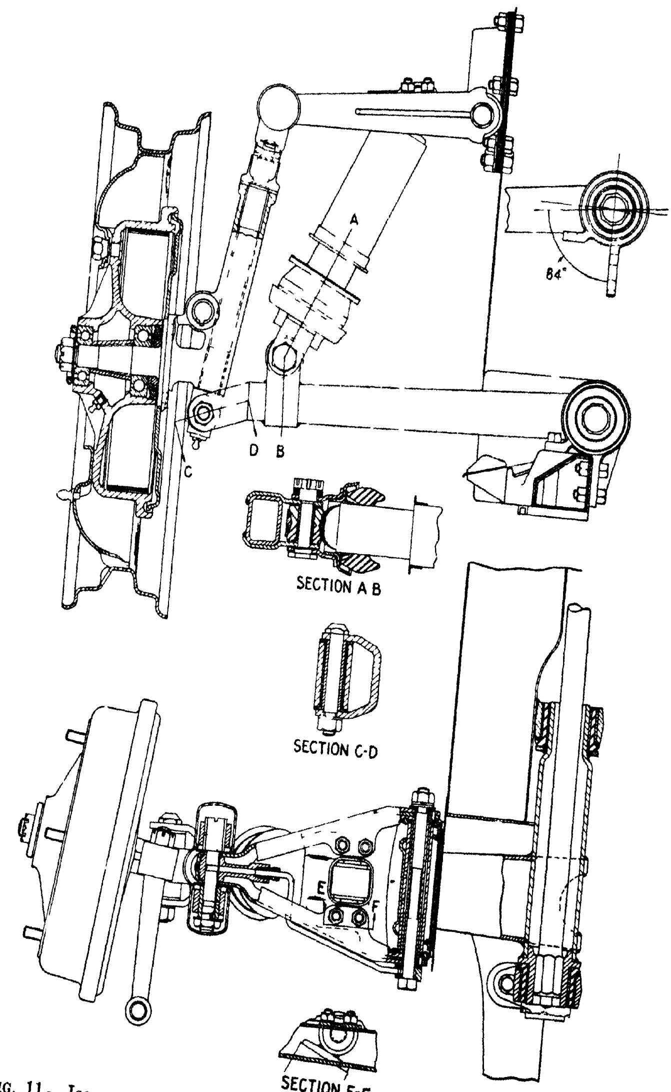


FIG. 11.—JOWETT FRONT INDEPENDENT SUSPENSION ON TORSION BARS.

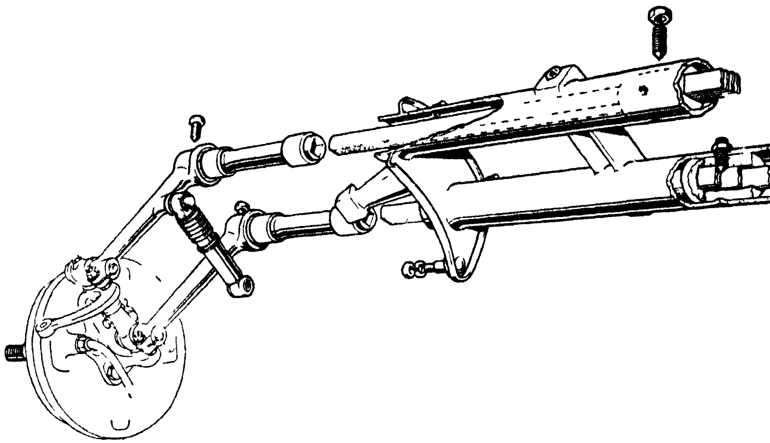


FIG 12.—VOLKSWAGEN FRONT INDEPENDENT SUSPENSION.

sion was confined to Continental makes, but in 1950 at least four British makes also had it. Among the earliest users of this suspension system were Steyr, Citroën, Austro-Daimler, Mercedes (rear-engined model), and the Volkswagen. The rear-suspension of the last-mentioned is illustrated in Fig. 13. The torsion bar is of circular section and extends transversely through a tube forming part of the understructure of the car. Laterally flexible arms have serrated joints at the outer ends

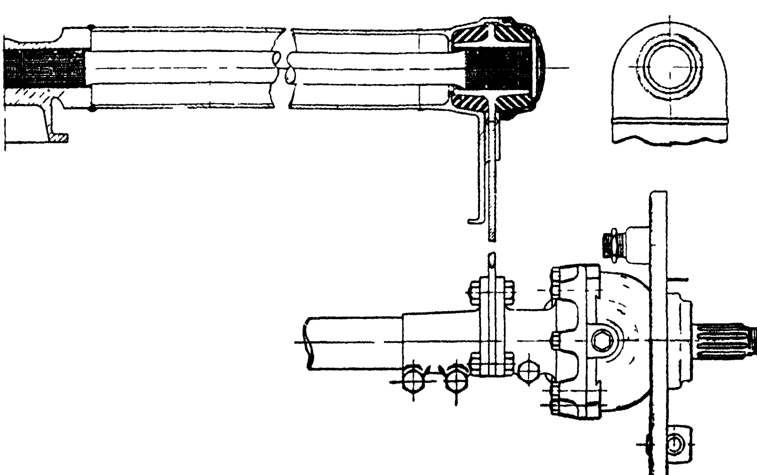


FIG 13.—VOLKSWAGEN REAR INDEPENDENT SUSPENSION.

of the torsion bars and are rigidly bolted to brackets near the ends of the swinging half-axles. In operation the axle ends and wheels swing around diagonal lines connecting the universal joints on opposite sides of the differential gear with the hubs of the torque arms, and this calls for torsional flexibility of the arms. These arms are of built-up construction, flat strips of pressed steel being butt-welded to dropforged hubs. This weld is shown in the sketch in the upper right-hand corner of Fig. 13. The arms not only take the moment due to the weight of the car supported on the axle, but also the driving thrust and the brake torque. There are serrations on the torsion bars at both ends, and to permit the replacement of the bars in case of failure without removing a wheel, the serrations at the inner end are made smaller in diameter than those at the outer end Adjustment for chassis

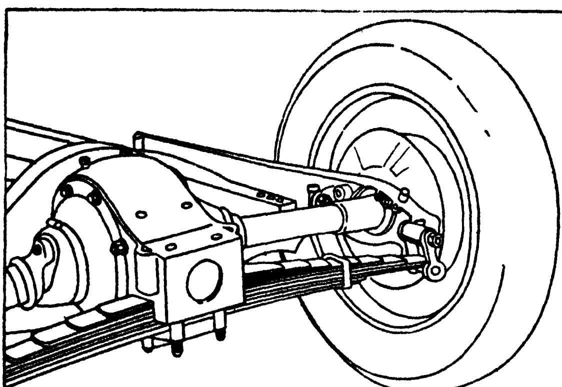


FIG. 14.—STEYR REAR INDEPENDENT SUSPENSION.

height can be made by turning the torsion bars in their serrated fittings. Double-acting hydraulic shock absorbers with detachable arms are mounted on brackets welded to the cross tube. Torsional deflections in both directions are limited by bumpers. The torsion bars are comparatively stiff, and the static deflection at the rear is considerably less than that at the front. Rear spring action evidently is limited purposely in order to minimize changes in the tread due to spring play.

Fig. 14 is a sketch of the rear suspension of the Steyr car. Radius rods connecting the ends of the axle housing to the frame side rails have universal connections to the latter. These rods extend a short distance back of the axle, and have the ends of an inverted semi-elliptic spring connected to them, the center of the spring being secured to the under side of a rearward extension of the center housing. This car has a panel-type chassis frame which, however, terminates in front of the axle, its rear cross member being bolted to the forward side of the drive-gear housing. The frame thus is supported at three points, and is said to be practically free from distortion.

In the Austro-Daimler there are three inverted semi-elliptic springs, directly below the axle. One of these has its center portion clamped to the under side of the center housing, while its ends are shackled to the centers of the two other springs, which extend between the center housing and the ends of the axle tubes, to which latter they are shackled. The Mercedes has coil springs at the rear.

It should be pointed out that all of the foreign rear independent suspensions described in the foregoing, with the exception of that of the Volkswagen, date back to the thirties, and may have been either modified or abandoned since, but few drawings of Continental cars have become available since the end of the war. No American manufacturers of passenger cars used independent suspension at the rear up to 1951.

Effect of Suspension on Swaying Motion—When a car is being driven around a curve, a centrifugal force acts at its center of gravity, in a direction radially outward from the center of the curve. Since the center of gravity usually is some distance above the springs, this force creates a moment which has the effect of increasing the load on the outside, and decreasing that on the inside springs, thereby causing the car to sway. From the standpoint of their effect on swaying, independent suspension systems may be divided into two classes, viz., those in which the wheels always remain perpendicular to the pavement, and those with which they tilt with the body. In analyzing the phenomenon of swaying, it will be well to consider first a conventional car with rigid axles (Fig. 15). Suppose a centrifugal force C to be acting at the center of gravity of the spring-supported mass and transferring a load W from the inner to the outer springs. The combined load on the two pairs of springs remains the same, and point A therefore remains the same distance above the axle and the road, from which it might be assumed that this was the center of swaying motion. Experience, however, has shown that the actual center of swaying motion is lower, usually at the approximate level of the front-axle center.

With a centrifugal force C and a moment arm D we have a moment CD, which must be balanced by the couples created by the reduction of loads on the springs at one side, and the increase in loads on those at the other side. As a rule, the distance between spring centers on opposite sides of the car will not be the same at front and rear, but we can take the mean between these two distances and refer to it as the spring base, E. Since the sprung mass of the car forms a rigid unit, the sway angle, or angle of inclination relative to the axles,

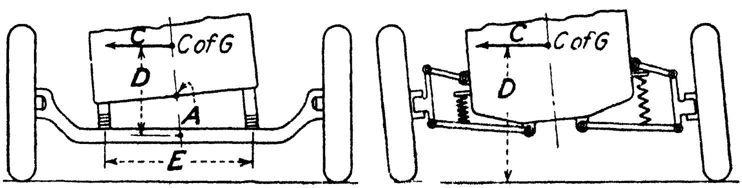


FIG. 15 (*left*).—SIDE-SWAY DIAGRAM FOR CONVENTIONAL CAR.
FIG. 16 (*right*).—SIDE-SWAY DIAGRAM FOR PARALLEL-LINK SUSPENSION.

will be the same at front and rear. If the resulting change in deflection of the springs at each side, referred to the (mean) spring base E, is denoted by f, and the combined rate of the two springs on each side by R, then the weight transfer W from one pair of springs to the other is fR, and we have

$$WE = fRE = CD,$$

from which it follows that

$$f = CD/RE.$$

The tangent of the sway angle is equal to the quotient of the deflection f by one-half of the spring base, hence

$$\tan \theta = f/0.5E = 2f/E.$$

Substituting the value of f found in the foregoing, we have

$$\tan \theta = 2CD/RE^2.$$

That is to say, the tangent of the sway angle—and also the angle itself, because tangents of small angles are practically proportional to the angles—varies directly with the centrifugal force and the height of the center of gravity above the sway axis, and inversely as the spring rate and as *the square* of the spring base.

Now let us assume a car having independent suspension of the parallel-link type at both front and rear. With such suspensions the spring rate is always taken as the load on one wheel which raises that wheel 1 in. relative to the plane of the frame. This is much smaller than the rate of the spring itself, as the latter is located nearly midway between the center plane of the wheel and the pivot axis of the lower control bar. Referring to Fig. 16, as with suspensions of this type the wheels tilt with the body, the swaying axis is at the level of the road surface, and the moment arm therefore is equal to the height of the center of gravity above the road. The spring base is equal to the tread. The same equation for the tangent of the sway angle applies in this case as in that of the conventional car, but the moment arm and the spring base both will be about 1.75 times as large. Since the sway angle varies as the first power of the moment arm, but inversely as the square of the spring base, with the same spring rate the sway angle of the independently sprung car would be less than that of the car with conventional springing in the proportion of 1:1.75.

In the foregoing the car was supposed to have parallel-link independent suspension at both front and rear because that simplifies the analysis. Most actual cars have this type of suspension only at the front, together with conventional springing on rigid axles at the rear. That case does not lend itself so readily to mathematical analysis, but undoubtedly the roll axis, or axis of swaying motion, is located between the axes in the two cases analyzed, that is, some distance above ground level. This reduces the moment arm, as compared with the car with independent suspension at both front and rear, so that the sway angle is reduced further.

While the car with front independent suspension thus compares very favorably with the conventional one as regards swaying tendency, nevertheless, on account of the soft springs now generally employed, swaying on turns would be unpleasantly pronounced if nothing were done to control it. To limit the swaying motion, passenger cars now are generally equipped with stabilizers or anti-sway bars, sometimes at one end only, but more frequently at both front and rear.

Stabilizer—A stabilizer consists of a steel bar extending across the car below the frame and supported from the latter in bearings secured to the under side of the side rails, the ends of the bar being turned toward the rear to form lever arms from which links connect to the lower control bars in the case of front suspensions of the parallel-link type, and to the axle or a bracket thereon in the case of conventional suspensions. The bearings for the bar usually are rubber-bushed, and the link connections are "rubber-insulated" to prevent rattles from developing in this part of the mechanism. As the bushings offer only little resistance to torsional movement, the bar has little effect on the combined rate of the two springs, with which it is connected in parallel as it were, but if the deflection of only one of the springs changes, a moment is imposed on the stabilizer bar, which transfers some of the additional load which caused the deflection, to the spring on the opposite side, thus tending to equalize the loads on both springs and preventing excessive sway. One end of a front stabilizer is show in Fig. 17.

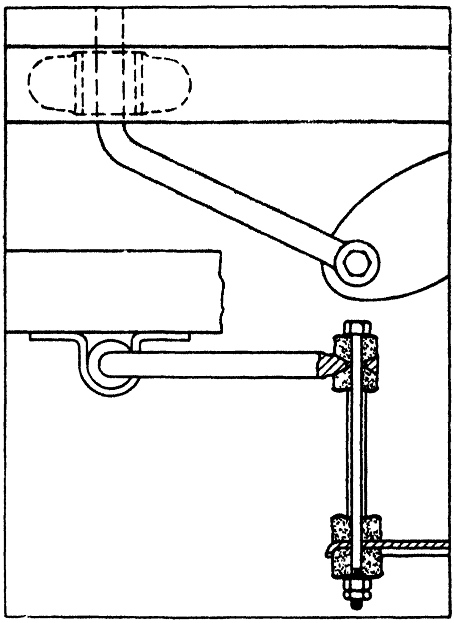


FIG. 17.—TWO VIEWS OF A STABILIZER.

Spring Load vs. Wheel Load—In a car with rigid axles the load on each front spring is equal to one-half of the weight of the spring-suspended mass carried by the front axle. In a car with independent suspension of the parallel-link type, where the spring-supported load is transferred to the knuckle spindle by means of a lever, the leverage factor must be taken into account. In Fig. 18, let W be the reaction of the wheel to the load carried by it; L, the load on the spring; R, the reaction of the pivot bearing on the lower control arm; a, the distance from the axis of the spring to the lower control-arm pivot axis, and b, the distance from the axis of the spring to the center plane of the wheel. Taking moments around the pivot axis of the control arm,

$$La = Wa + Wb$$

and

$$L = W(1 + b/a).$$

The vertical movement of the wheel bears the same proportion to the vertical movement of the spring rest as the spring load to the wheel load. Denoting the spring deflection by f_s and the wheel movement by f_w,

$$f_w = f_s(1 + b/a).$$

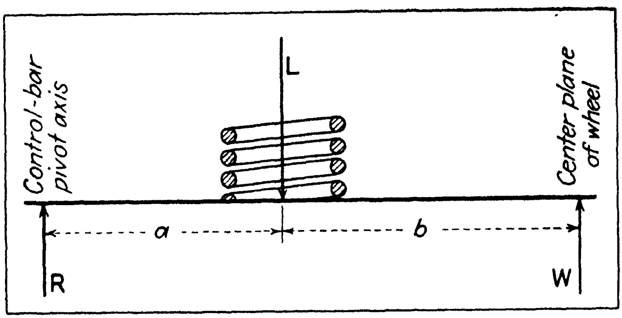


FIG. 18.—DIAGRAM SHOWING RELATION BETWEEN SPRING LOAD AND WHEEL LOAD.

In practice b usually is about 1.3 times a, hence the spring must be designed to take a load equal to about 2.3 times the spring-supported load on each front wheel, and to have a maximum deflection equal to about 44 per cent the maximum wheel movement relative to the plane of the frame. Actual deflections under static load of coil springs in front independent suspensions range between 4.5 and 5.5 in. The "metal-to-metal" clearance under static load, referred to the center plane of the wheel, should be between 4.5 and 6.5 in.

The relationship between wheel load and spring load is similar in the case of rear independent suspensions with pivoted or swinging axles.

CHAPTER XIX

Rubber Suspensions

A good deal of experimental work on rubber suspensions for road vehicles has been done both here and abroad, and in recent years there have been a number of commercial applications. This development probably received its first impetus from the use of rubber shock cords in the landing gears of early airplanes. Rubber has a much higher capacity for storing energy elastically than even the best grades of spring steel. The latter is used to best advantage in torsion bars, where it is capable of storing from 1000 to 1500 in.-lb per pound. With rubber it is possible to store from 2000 to 4000 in.-lb per pound of material, according to the composition and the uniformity of stress distribution. In a "tank" equipped with rubber springs during World War II as much as 4875 in.-lb per pound was stored in the "bump" position, but, of course, the suspensions of such vehicles need not be designed for long life.

In addition to its greater capacity for storing energy elastically, rubber offers a number of other advantages as a medium for vehicular suspensions. If properly applied it reduces the static friction in the suspension system to a minimum, thereby reducing the "harshness" of the ride; by eliminating rattles and squeaks it reduces the noise level of the vehicle; it obviates the need for lubrication of parts in metallic contact, and it makes sudden failure—such as occasioned by the breakage of spring plates—practically impossible.

Rubber in suspension systems may be worked either in compression, in tension, or in shear. Rubber blocks in compression are extensively used as auxiliary springs or spring bumpers, and these usually are so designed that their rate increases rapidly with compression. A number of attempts have been made—chiefly in England and France—to use rubber in compression for the main chassis springs, each suspension unit usually comprising a considerable number of rubber discs or biscuits. However, in compression the elasticity of rubber does not seem to be fully taken advantage of, and ever since the problem of securely bonding rubber to metal was satisfactorily solved in connection with the development of rubber engine mountings, most effort has been devoted to suspensions in which the rubber is worked in shear.

Fig. 1 shows sectional and end views of the Goodrich Torsilastic suspension unit, which has been used in Army "tanks" and is now being fitted to Twin Coach buses. The unit consists basically of two concentric metal tubes, the space between which is filled with rubber that is bonded to both. In regular production units the outer tube or shell is split, as that permits of applying a high pressure to the bond between rubber and metal during the vulcanizing or curing process, allows the rubber to shrink after vulcanization without causing internal stresses, and places the rubber and the bond under radial pressure, which has been found to increase the life of the spring. These suspension members may be used in a number of different ways. For instance, the outer tube of a unit extending lengthwise of the vehicle may be rigidly secured to the axle, and the inner tube provided with an arm which is shackled to the vehicle frame. This gives a conventional suspension system with rigid axles. If it is desired to spring the wheels independently, the outer tube of the unit may be secured to the end of a frame cross member and the ends of the inner tube provided with arms which form one of the parallel links of an independent front suspension.

Where the rubber is interposed between two concentric tubes and subjected to torsion, the greatest stress evidently occurs at the bond to the inner tube. If the bushing is considered to have plane ends, the stress in the rubber varies inversely as the square of the radius or the distance from the axis of the tube. This is easily seen, because for a given moment on the suspension member (load carried times effective moment arm), the shearing force is inversely proportional to the radius. This total shearing force is distributed over an area which increases directly with the radius. Therefore, at the inner bond not only is the total shearing force greater than at the outer bond, in the inverse proportion of their radii, but it is distributed over an area which is smaller than that at the outer bond, in direct proportion to the radii. In practice, in order to ensure a fair degree of uniformity of stress distribution and use the material to good advantage, the thickness of the rubber "bushing" must be limited.

With a spring unit of the type illustrated in Fig. 1 the torque T which produces a stress S_1 at the inner bond is

$$T = 2\pi r_1{}^2 l S_1 \text{ lb-in.},$$

and the deflection due to this torque is

$$\theta = \frac{45T}{4\pi^2 lG}\left(\frac{1}{r_1{}^2} - \frac{1}{r_2{}^2}\right) \text{deg},$$

where G is the modulus of rigidity of the rubber, in psi, and r_1, r_2 and l are the dimensions indicated in the drawing, in inches.

Quite naturally, both the modulus of rigidity and the permissible stress vary with the compound used. It appears that the rubbers best suited for use in torsion springs have a durometer hardness of from 45 to 60 and that their modulus of rigidity ranges from about 60 to about 100 psi. As to the permissible shear stresses, in a certain bus application the stress at the inner bond under static conditions with a full seated load is 111 psi; with a full complement of "standees,"

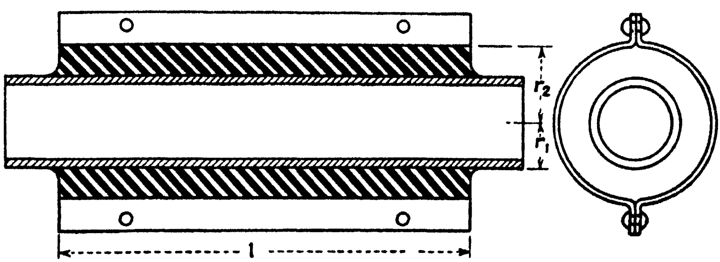


FIG. 1.—TORSILASTIC SUSPENSION UNIT IN SECTION AND END VIEW.

132 psi, and with the spring deflected to the "full-bump" position, 180 psi.

A difficult problem arising in connection with rubber suspensions is that of settling of the vehicle due to "creep" of the rubber. Vehicles equipped with steel springs also will settle in the course of service, but this settling can be reduced by presetting, as explained on p. 474. Fig. 2 is a so-called "creep record" of a pair of rubber springs in actual service, extending over a period of nearly three years. It will be seen that the car settled rather rapidly at first. Adjustments were made after two days and 13 days. Settling continued at a reduced rate for a whole year, and some engineers have expressed the opinion that the "creep" never stops completely as long as the vehicle remains in service. Creep varies with the stress to which the rubber is subjected, but not in direct proportion. It is also dependent on the temperature and on agitation or exercise of the rubber. That is to say, the springs may settle at night while the car is in the garage but will recover again, at least in part, during a run the following day.

The problem of settling may be solved by providing means for either manual or automatic adjustment. Manual adjustment is exemplified in the Torsilastic suspension system of the Twin Coach bus illustrated in Fig. 3. In this installation the outer tubes of the suspension units extend in the direction of vehicle length and are fixed to the axles, while to the ends of the inner tubes are fitted two-part moment arms or torque arms. Both parts of the arm are mounted on the end of the tube, one part being free thereon, the other fixed. The fixed part is provided with an eye at its outer end and the free part has sectors with a series of holes in them which can be made to register with the hole in the eye of the fixed part. The two parts are locked together by means of a bolt passing through both, and by moving the bolt from one hole in the

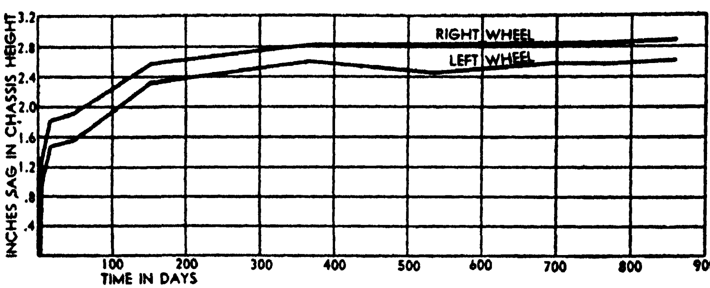


FIG. 2.—CREEP RECORDS OF RUBBER SPRINGS IN SERVICE.

sector to the next, the vehicle can be raised or lowered. The end of the moment arm connects by a rubber-bushed, link-type shackle to a bracket fastened to a body-frame member.

With rubber suspensions of the concentric-tube type, in addition to angular movement of one tube relative to the other, there are also slight axial and tilting motions. These motions, however, are quite small and are said to be beneficial in that they reduce the harshness of the ride. Rubber springs of the type described have a substantially straight load-deflection line; that is to say, a given increase in load produces the same increase in angular deflection throughout the range. Of course, the vertical motion of the wheel relative to the chassis depends not only on the rubber member but also on the linkage, and may, therefore, be variable.

For passenger-car suspensions an automatic constant-level mechanism has been suggested by the B. F. Goodrich Com-

FIG. 3.—TORSILASTIC FRONT SUSPENSION OF TWIN COACH.

pany, and is illustrated in Fig. 4. In this case the moment arm which transfers the chassis weight to the wheel is secured to the outer tube of the suspension unit, and the inner tube is carried in trunnions on the frame and provided with an arm that presses against a piston in a hydraulic cylinder. Oil can be pumped into and out of the cylinder by means of a pump driven by an electric motor. At each of the four

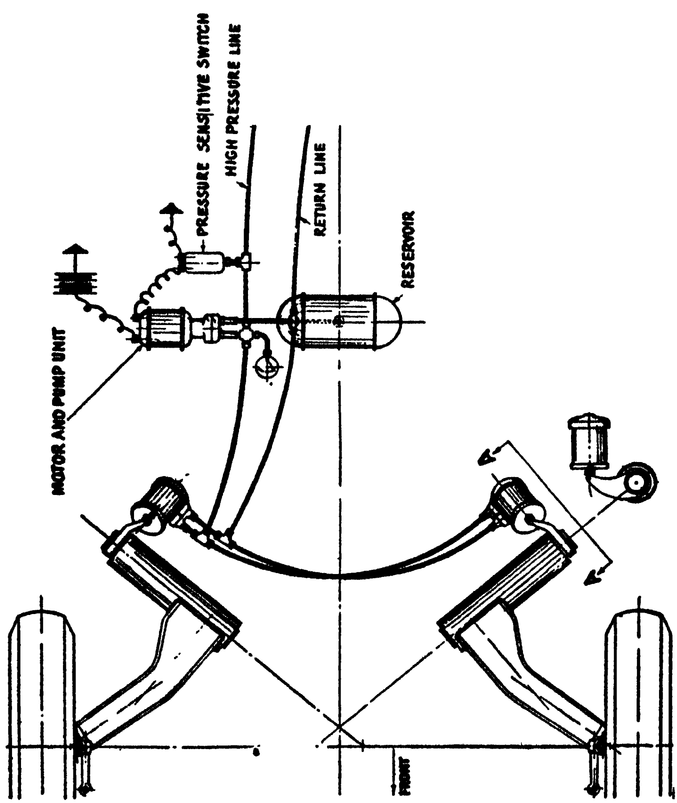


FIG. 4.—AUTOMATIC CONSTANT-LEVEL MECHANISM, FRONT SECTION.

suspension units there is a lever which moves up and down with the wheel-support arm. This lever controls a slide valve which connects the hydraulic cylinder to either the pressure or the return line of the hydraulic system. Small coil springs are interposed between the lever and the slide valve, and when there is the normal wheel movement due to road irregularities the dashpot on the slide valve prevents the valve from moving, the coil spring changing its length with the wheel movement. When the level of the car changes as a result of added passenger load, the wheel-control arm moves from its normal position long enough to move the slide valve by withdrawing fluid from the dashpot. Thus the vehicle frame or body always remains at a definite height above the pavement. The predetermined level is restored not only when there is a change in the passenger load, but also when the body settles as a result of rubber creep.

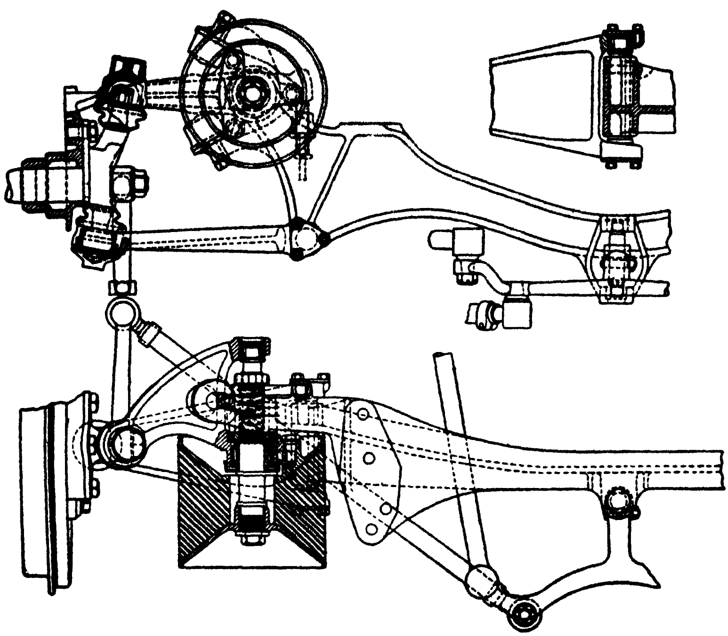


FIG. 5.—COLIN CAMPBELL RUBBER SUSPENSION.

Campbell Rubber Suspension—Another rubber suspension unit was developed by Colin Campbell in England, and one application of it is illustrated in Fig. 5. In this design the rubber is bonded to two opposed dished discs or truncated cones, and motion takes place around the common axis of the cones. The two cones are held in alignment by a shaft secured into one of them and carried in a bearing within the other. The unit is so designed that the axial width of the rubber increases in direct proportion to the distance from the axis, hence when the unit is subjected to torsion all of the rubber is stressed substantially uniformly. The angular deflection of a Macbeth spring is given by the equation

$$\theta = \frac{114.6Tt}{G(r_2^4 - r_1^4)} \text{ deg},$$

where T is the torque in lb-in.; t, the mean axial thickness of the rubber; r_1, the inside and r_2, the outside radius of the rubber, in inches. It will be seen that the outer edges of the cones are flared, the object being to prevent failure of the bond at the edges. Fig. 5 shows an application of the Campbell unit to an existing car. One of the cones was firmly secured to the end of a cast-steel frame cross member which in turn was bolted to the under side of the frame side members. A moment arm secured to the shaft of the suspension unit formed the short link of a parallel-link independent front suspension. In descriptions of the installation which appeared in the British technical press it was stated that provision was made in the design for shock absorbers to act around the axis of the lower link or wishbone, but that none were actually fitted.

Damping—Rubber springs always produce a certain hysteresis or damping effect. If a spring in the free state is subjected to a given load it will show a definite deflection, but if the load is then increased and thereafter reduced to the original value, the deflection will not be the same, but somewhat greater. Therefore, two different load-deflection curves will be obtained if the data for the curves are secured from tests in which the load is gradually increased and gradually decreased, respectively. The area under the curve obtained with increasing load is a measure of the energy which is being stored in the spring, and that under the curve obtained with decreasing loads is a measure of the energy restored by the spring during the unloading process. The difference, which is measured by the area of the so-called hysteresis loop, is a measure of the energy loss due to hysteresis, or of the damping effect. Rubber chemists know how to produce rubber compounds having relatively large damping effects, but such compounds always contain large proportions of mineral fillers and therefore have poor elastic properties, which makes them unsuitable for suspension units. Therefore, although rubber has inherent damping qualities, shock absorbers probably will be called for in most applications, especially on passenger cars.

CHAPTER XX

Wheels and Tires

Few parts of the motor vehicle have undergone such extensive changes since its inception as the road wheels. Most of the earliest cars had wood wheels, a type then in common use on horse vehicles, while some were equipped with wire wheels, which had been developed for bicycles a short time previously. Neither type was entirely satisfactory. Wood wheels were rather heavy, and during extended dry spells their spokes would dry out, shrink and come loose, while wire wheels in many cases were too light for the service and gave trouble from breaking of spokes or stripping of threads. An improved and more substantial type of wire wheel was introduced here from England during the twenties, and for a time was regarded as the finest type of passenger-car wheel; but it was rather expensive, and in service it was difficult to clean. Ford Motor Company for a number of years used a wire wheel of special design, in which the spokes, instead of being assembled to the hub and rim by means of threaded nipples, were welded thereto. Wire wheels, however, have become impractical for passenger cars, because with the small wheels now used the distance between hub and rim is so short that any saving in weight due to the use of wire spokes is insignificant, and does not warrant the higher production cost and the more troublesome cleaning operation.

Following horse-vehicle practice, wheels of comparatively large diameter were used. One argument in favor of large wheels was that they would not drop so deeply into pot holes. For a long time passenger cars were equipped with wheels of from 32 to 36 in. diameter, and a few even had 40-in. wheels. This applies to what might be called conventional cars. A type of "motor buggy" with wheels of as much as 48 in. diameter, shod with solid rubber tires, enjoyed a certain vogue for a number of years, being specially designed for use on unimproved roads. However, its disadvantages proved to outweigh its advantages, and it was soon dropped.

Wheel Diameters—American full-size passenger cars in 1951 were built with wheels of effective rolling diameters ranging from about 26 in. to about 30 in., 28-in. wheels predominating. However, the diameter is now rarely used as a measure of wheel size. Instead the tire and rim sizes are given. For instance, an 8.00/15 in. tire is a tire of 8.00 nominal width designed to fit a rim of 15 in. base diameter. The effective rolling radius of the tire or wheel varies with the degree of inflation and also with the state of wear of the tire. Rolling radii of truck tires at standard inflation pressures were given in Chapter II.

There were a number of reasons for the reduction of wheel diameters from an average of 34 in. during the high-pressure-tire era to an average of 28 in. For one thing, it led to a lowering of the center of gravity of the car, which is conducive to greater safety and improved appearance. Lateral thrusts on the wheels, due either to side skids or to hitting the curb while maneuvering, impose less strain on the axle and axle shafts, because the moment arm is shorter. With smaller wheels less gear reduction is required, and with the same engine torque the rear-axle torque is lower, which permits of the use of a lighter differential and lighter axle shafts. Finally, the smaller tires cost less.

Tires for highway trucks and buses are being manufactured in sizes ranging from 6.50/17 in., having an effective rolling diameter of about 28 in., to 14.00/24 in., with an effective rolling diameter of about 48 in. The number of plies in the tires and the recommended inflation pressure vary with the size, and load ratings of the tires range from 1300 lb to 9150 lb. For very large, so-called "earth-moving" vehicles tires are made in as large sizes as 30.00/40 in., with load ratings of up to 45,000 lb.

Metal wheels now are being used exclusively by the motor-vehicle industry. Most of them are made of steel by either a cold-pressing or a hot-working process, but cast wheels also are used to a considerable extent for heavy commercial vehicles. Pressed-steel wheels are fabricated from steel of relatively low carbon content—the only kind well adapted to cold-working. Wheels that are forged or otherwise fabricated while hot are made from medium-carbon steel, which has superior mechanical properties, and therefore permits of the use of lighter sections, and of a consequent saving in weight. A further reduction in weight is made possible by the fact that when wheels are forged, the sections need not be uniform, and the material therefore can be distributed to better advantage.

Tires—All road vehicles now are equipped with pneumatic tires. Such tires are being manufactured in a number of different types, for different classes of service. The pneumatic tire comprises a shoe or cover, an inner tube, and sometimes also a flap which prevents the tube—when deflated—from being caught between the tire beads and the rim. The tube forms an air-tight chamber and is provided with a valve through which it can be inflated, while the cover supports the tube against the air pressure within it. The cover is mounted on a steel rim.

Three basic types of rim have been developed and widely used. First came the clincher rim, which is used together with a clincher cover having extensible beads. A sectional

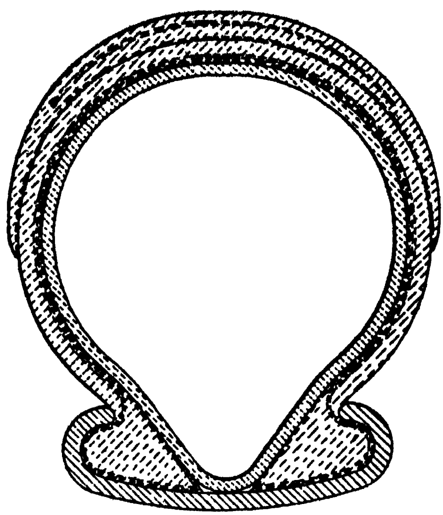


Fig. 1.—Cross Section of Clincher Tire and Rim.

view of a clincher tire and rim is shown in Fig. 1. When the tire is inflated, the air pressure forces the beads firmly against the clincher, whereas when it is deflated, one of the beads can be pushed out of the clincher, and then stretched and pried over the rim, so that the cover can be removed from the wheel. Clincher tires and rims are still being used for bicycles and motorcycles, but their use on motor vehicles was discontinued years ago. One reason for this was that in large sizes it was difficult to remove the tire from the rim, especially if it had become attached thereto by rust. This type gave some trouble also from pinching of the tube between the tire bead and rim, from injury to the tube by tire tools, and from failure of the side walls of the cover if the vehicle was driven with the tires insufficiently inflated.

Next in order—as far as automotive practice is concerned—came the straight-side tire and rim. As shown in Fig. 2, the straight-side tire is made with inextensible beads having wires embedded therein, and in order to get it onto the rim, the latter must be split either circumferentially or transversely. Circumferentially split rims, or rims with a detachable side ring, are now in most extensive use. All passenger cars, as well as many commercial vehicles, are equipped with

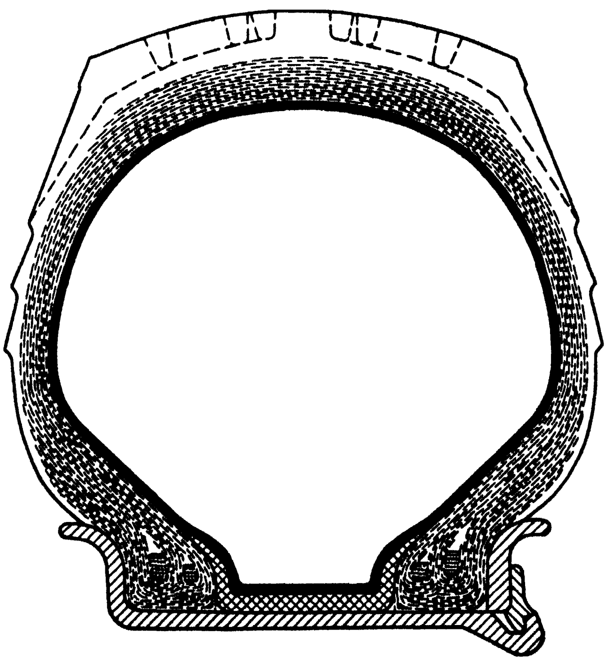


FIG. 2.—CROSS SECTION OF STRAIGHT-SIDE TIRE AND RIM.

demountable wheels of the disc type, and with such wheels the rim is riveted to a flange on the circumference of the disc, hence splitting the rim transversely would be of no help in mounting the tire. Circumferentially-split rims are provided either with a split, detachable side ring fitting into a groove formed at the side of the base, which in service is held in position by the lateral pressure of the tire against it; or with a continuous side ring which slips over the base of the

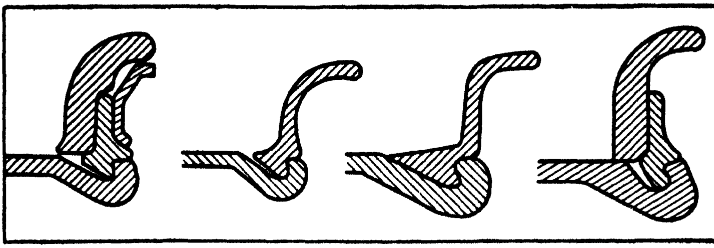
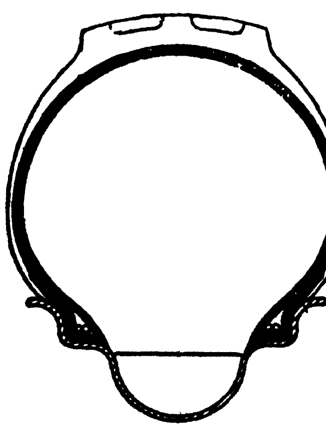


FIG. 3.—SHOWING VARIOUS METHODS OF SECURING REMOVABLE FLANGE.

rim and in service is held in place by a split locking ring which enters a groove at the side of the base. Several designs of detachable side rings, with and without locking rings, are shown in Fig. 3.

The third type of tire, like the clincher, is mounted on a one-piece rim, but unlike the clincher tire, it has non-extensible beads, and is mounted on what is known as a drop-center rim. This tire and rim combination was originated by the Dunlop Rubber Company in England for use on bicycles. It came into use on motor vehicles only when the low-pressure or balloon type of tire was introduced, during the middle twenties. A sectional view of the original Dunlop "well-base" rim with its tire is shown in Fig. 4, and Fig. 5 illustrates the method of placing the tire on the rim. When the tire is deflated, the beads at one end of a diameter are



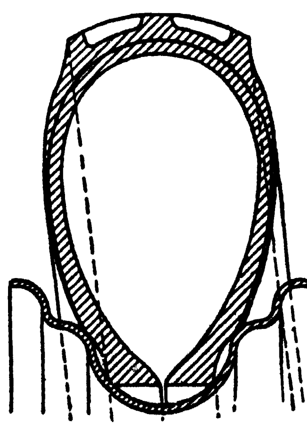


FIG. 4 (*left*).—THE ORIGINAL DUNLOP "WELL-BASE" RIM WITH TIRE.
FIG. 5 (*right*).—TIRE BEING PLACED ON "WELL-BASE" RIM.

pushed into the well at the center of the rim, and at the opposite end of the diameter they can then be slipped over the sides of the rim. This type of tire has come into practically universal use in the passenger-car and light commercial-vehicle fields. Fig. 6 is a section of a modern American drop-center rim with tire. The form of the "well" differs somewhat with different manufacturers. Where the rim is riveted to a disc wheel the bottom of the well naturally is made flat, as that facilitates the riveting operation.

Demountable Rims and Demountable Wheels—In the course of its evolution the pneumatic-tired vehicle wheel has passed through four stages. In the original wheel all parts

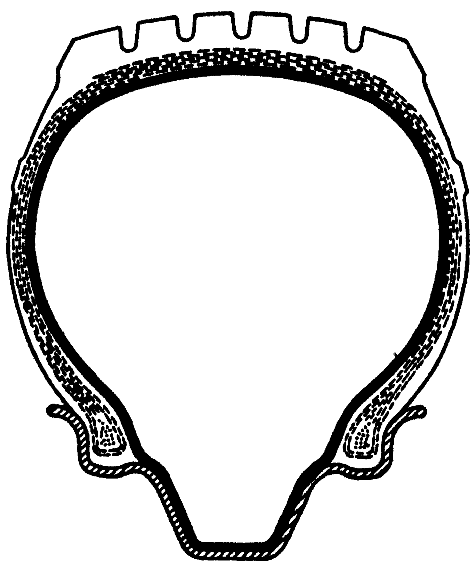


FIG. 6.—MODERN AMERICAN DROP-CENTER RIM WITH TIRE.

except the tire formed a rigid unit, and in the event of tire trouble, the tire had to be removed from the wheel, a repair made, and the tire replaced before the journey could be resumed. This was a rather irksome operation as long as clincher tires were being used, but it was eased somewhat by the introduction of straight-side tires on separable rims. Next came the demountable rim, which permitted of carrying a spare rim with inflated tire on the car, so that in case of trouble the damaged tire with its rim could be removed from the wheel and the spare substituted. The final stage came with the demountable wheel, which permits of carrying a spare wheel with inflated tire. At the time of this writing, all disc wheels and all spoked wheels made by pressing or forging processes are of the demountable type, while cast wheels are non-demountable and are provided with demountable rims.

Inflation Pressures—Early pneumatic tires were of what is now known as the high-pressure type. Recommended inflation pressures averaged 20 psi per in. of width, a 3-in. tire being inflated to 60 psi, and a 5-in. (which was the largest then made), to 100 psi. It was later found that the ride could be improved, the risk of punctures and blowouts reduced, and tire-life increased by using lower pressures in tires of larger section. This led to the development of the balloon tire, now in common use, and even a step further, to the so-called superballoon, in some examples of which inflation pressures as low as 10 psi were carried. These very low pressures proved impractical, however, for one reason because they impaired the lateral stability of the vehicle. At present passenger-car tires are made in two types, four-ply and six-ply. For six-ply tires the maximum recommended inflation pressure is 28 psi; for four-ply, 25 psi.

Load Ratings—Tire and rim sizes, inflation pressures, load ratings, etc., have been standardized by the Tire and Rim Association of Akron, Ohio, with the result that tires of all makes are interchangeable on rims of corresponding size and type. The standard maximum load ratings of passenger-car tires are given in graphical form in Fig. 7. Load ratings increase with the tire width and the outside diameter, but somewhat more rapidly than the product of these two dimensions. For instance, ratings at maximum recommended inflation pressures of six-ply passenger-car tires correspond closely to values of the expression $3.26\,(wd)^{1.18}$, where w is the nominal width and d the nominal outside diameter of the tire.

Truck and Bus Tires—The heavier trucks and buses are equipped with straight-side tires on flat-base rims, as illustrated in Fig. 2. With increase in the section of the tire, the number of plies is increased, and the larger number of plies permits of higher inflation pressures. Maximum load ratings of balloon-type bus and truck tires, as standardized by the Tire and Rim Association, are shown in graphical form in Fig. 8, where the numbers of plies of the different-sized tires and the recommended inflation pressures also are given. Load ratings at inflation pressures below the maxima given in the chart are approximately proportional to $p^{0.585}$, where p is the inflation pressure in psi. To permit of determining these load ratings without recourse to a table of logarithms, values of $p^{0.585}$ for various values of p are given in the following table:

p	24	30	36	50	60	70	80	90	100
$p^{0.585}$	6.42	7.31	8.14	9.86	10.97	12.00	12.98	13.90	14.80

Some of the tire sections represented in Fig. 8 are made also with a larger number of plies, and are then able to withstand higher inflation pressures. Thus, 6.50 tires are made

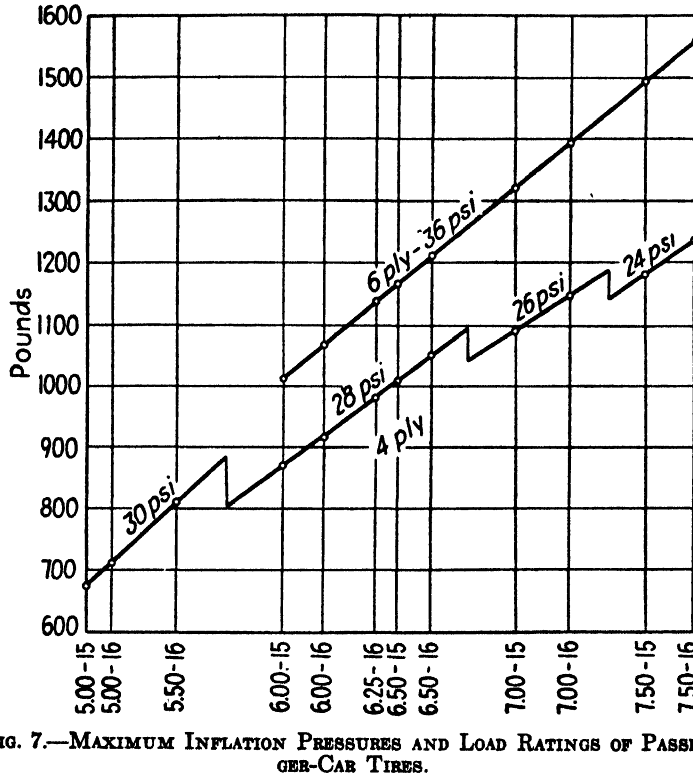


FIG. 7.—MAXIMUM INFLATION PRESSURES AND LOAD RATINGS OF PASSENGER-CAR TIRES.

also with eight plies, for an inflation pressure of 65 psi; 7 in. with ten plies, for 70 psi; 7.5 in. with ten plies for 75 psi; 8.25 in. with twelve plies for 75 psi, and 9 in. with 12 plies for 80 psi. When the number of plies thus differs from the standard, and a higher inflation pressure is used, the load capacity varies substantially as $p^{0.585}$. For instance, a 7.50/20 eight-ply tire at its maximum recommended inflation pressure of 55 psi has a maximum load rating of 2250 lb. Using values obtained from the above table by interpolation, the load rating of a 7.50/20 ten-ply tire, inflated to 75 psi, would be

$$2250(12.49/10.42) = 2700 \text{ lb.}$$

For each standard tire size there is a standard rim size. The rim is designated by the diameter of its base and the width between flanges. The proportion between the width of the standard rim and the nominal tire width varies slightly from size to size.

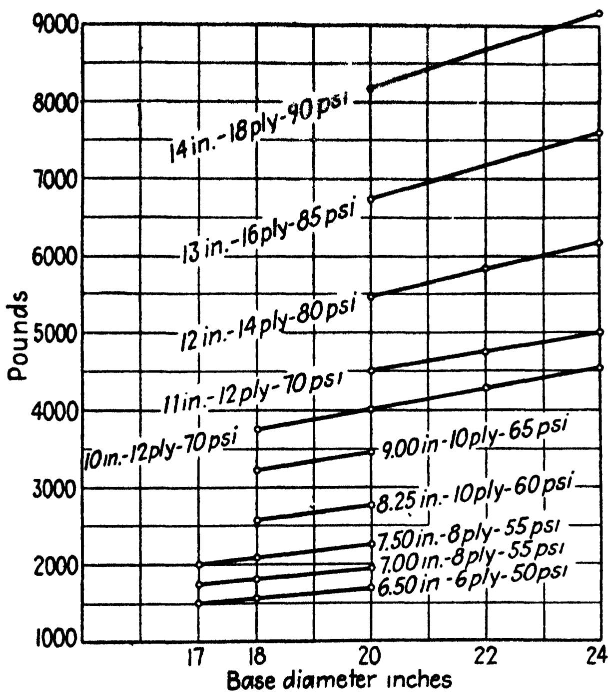


FIG. 8.—MAXIMUM INFLATION PRESSURES AND LOAD RATINGS OF BUS AND TRUCK BALLOON TIRES.

Wide Rims—As inflation pressures were decreased and tires enlarged, it was found that with rims of the original proportions as regards width (width between rim vertical flanges 50 per cent of the tire width) cars were rather unstable and had a tendency to "wander," especially when negotiating turns. This led to the use of wider rims, in a number of steps. In 1951, wide-base rims, in which the width between flanges ranges between 75 and 85 per cent of the nominal tire width, were in wide use on passenger cars. In Fig. 9 tires on standard and wide-base rims are shown side by side, the proportional widths of rims being 69 per cent and 81 per cent, respectively. It has been found that wide rims reduce tread wear and increase the stability of the car. The life of the tire fabric does not seem to be affected by rim width to any degree, but the resilience of the tire is decreased somewhat by an increase in rim width, and the flanges of the rims, being less protected by the overhanging part of the tire, are more subject to injury. Altogether, the advantages of the wide-base rim outweigh its disadvantages.

Disc Wheels—Fig. 10 shows a typical passenger-car wheel in section. It is of the disc type, the steel disc being formed with a lateral flange to which the drop-center rim is riveted. In early disc wheels—of relatively large diameter—the discs

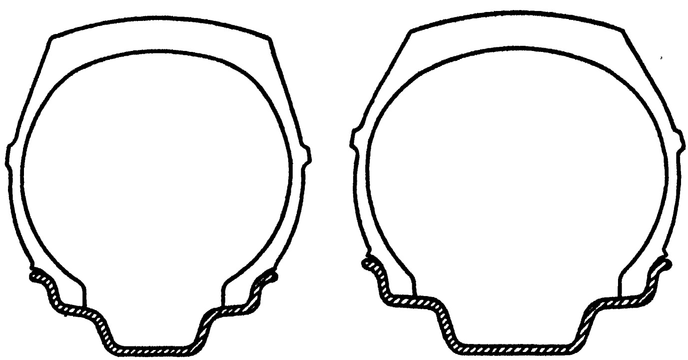


Fig. 9.—Tires on Narrow- and Wide-Base Rims.

usually were substantially flat, while at present they are nearly always curved. The reasons for curving (or cupping) the discs probably are that it facilitates some of the manufacturing operations, improves the appearance of the wheel, and adds slightly to its flexibility. The outer portion of the disc is made convex toward the outside, and wheels are provided with wheel caps having the form of spherical segments, which complete the convex surface of the disc. Sometimes the entire wheel, up to the rim, is covered by a wheel cap. The wheel cap is held in place by spring-steel clips riveted to the disc and engaging an edge either on the cap itself, which in that case is spun over inward, or on a mounting ring, which is held in position on the wheel cap by the edge of the cap being crimped over it. The discs often are stiffened by having ribs pressed in them, ribs at the outer flange increasing the radial, ribs near the inner or mounting flange the lateral stiffness. Wheels for heavy commercial vehicles always have the discs formed with louvers, to facilitate air circulation between the brake drum and the rim or rims.

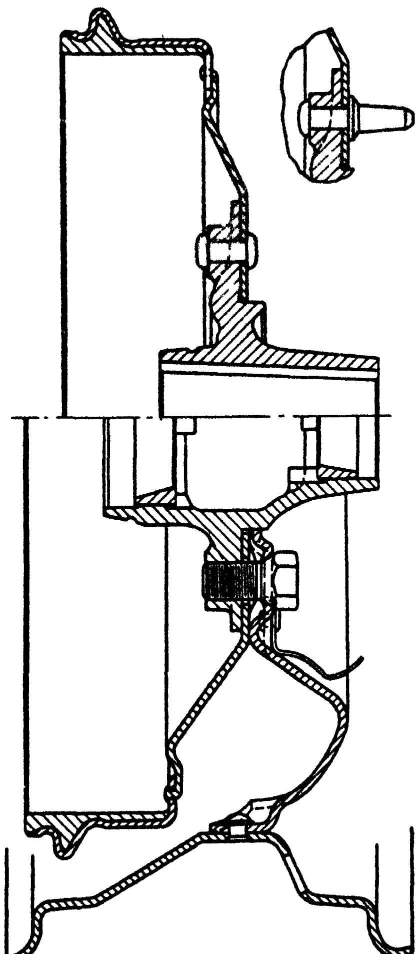


FIG. 10.—DISC-TYPE PASSENGER-CAR WHEEL (MOTOR WHEEL).

Disc wheels are secured to a flange of the hub by means of five or six attaching bolts or studs. In the case of single wheels, where only a single disc must be secured to the hub flange, cup-shaped depressions are formed in the disc at the mount the outer half of the dual wheel without disturbing the inner half. Dual wheels with rims having detachable flanges are mounted with these flanges on the inner side, that is, between tires.

Ventilation of Dual Wheels—One difficulty encountered with bus and truck wheels is that the brake drums come very close to the rims, and in services where the brakes have to be used a great deal, much heat passes from the brake drums to the rims, and from the latter to the tires, which are quite sensitive to high temperatures. With rims of 20-in. base diameter, for instance, it is quite common to use brake drums of 17 in. inside diameter, which leave a clearance of only about ¾ in. This may be further cut down by a central circumferential stiffening rib on the drum. There is very little opportunity for air circulation through such a narrow space. The drum sometimes is made with cooling ribs over its entire exterior surface, and although this will serve to keep down brake temperatures, it is doubtful whether it is of much help in keeping down tire temperatures. To protect the tires against injury by excessive temperatures, it is necessary to promote circulation through the clearance space by ventilating both wheel discs and providing helical ribs or fins on the brake drum, or radial vanes between the rims and the lower parts of the tires.

Spacing of Dual Tires—When dual tires are used they must be correctly spaced. Owing to the fact that on curves the two wheels, although rotating at the same speed, must travel different distances, there is of necessity some slip between tires and pavement, and this slip will be less the more closely the wheels are spaced. On the other hand, there must be sufficient space between the tires so that anti-skid chains can be applied, and in laying out the design it must be borne in mind also that the owner later on may want to fit wide-base tires. The Tire and Rim Association has standardized the following spacings for dual wheels with balloon tires:

Tire width	5.50	6.00	7.00	7.50	8.25	9.00
Spacing	7.25	7.75	9.00	10.00	10.50	11.50
Tire width	9.75	10.50	11.25	12.00	12.75	13.50
Spacing	12.00	13.25	14.00	15.25	16.00	16.00

Cast Wheels—Wheels are cast of either malleable iron or steel. Malleable iron is cast iron which has been partly decarburized by prolonged exposure to high temperatures, and consequently is tougher or less brittle than cast iron. An average composition of malleable iron is as follows: C, 2.45 per cent; Mn, 0.22; Si, 0.80; P, 0.20; S, 0.065.

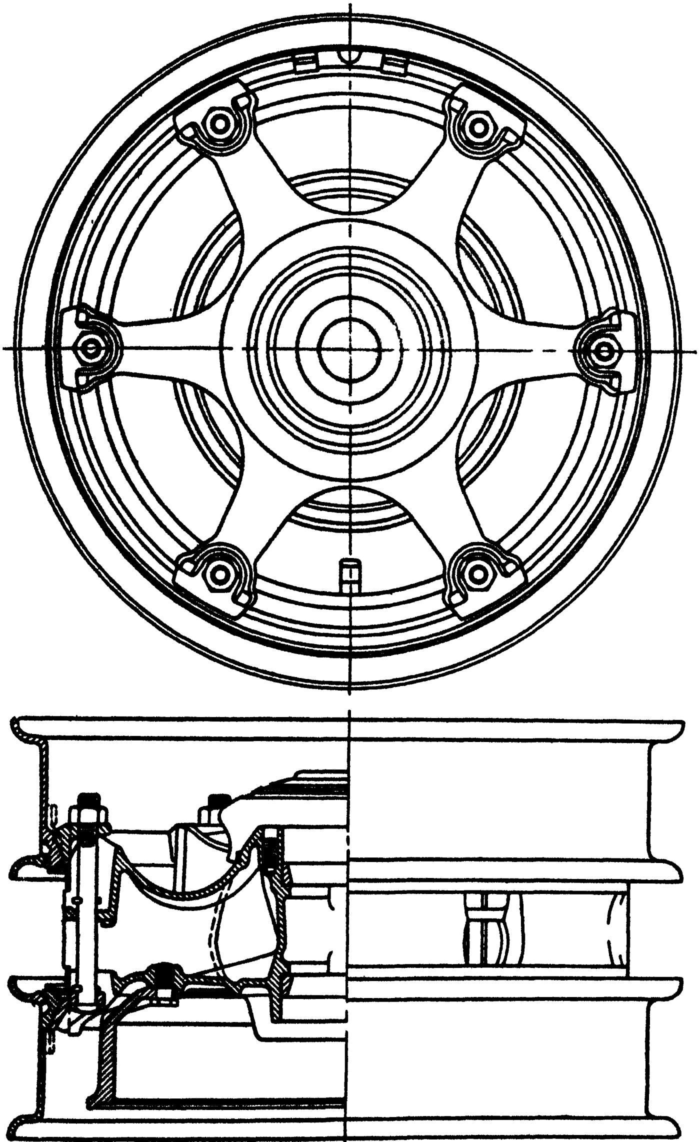


FIG. 13.—CLARK MALLEABLE-IRON DUAL WHEEL.

When cast wheels were first introduced, they were equipped with solid rubber tires, and it was then necessary to cast them with continuous rims, which made them rather heavy. At present, with pneumatic tires, the integral rim is omitted, the tire rims being mounted directly on the ends of the spokes.

Fig. 13 shows two views of the Clark malleable iron wheel. These wheels have six hollow spokes cast integral with the hub. The hub is of box section and reinforced by internal ribs. An advantage of malleable iron is that it can be cast in very thin sections, which makes for lightness. To further reduce the weight of the complete wheel, the cast-iron brake drum is made with an attaching flange of small depth, by means of which it is secured to the spokes, the back of the drum being closed by a thin sheet-metal disc clamped between it and the spokes, and extending inward to the hub.

Cast wheels always have demountable rims, which latter have an inclined mounting surface on the inside. In single wheels this mounting surface contacts a corresponding surface on the end of the spokes, against which the rim is drawn by attaching bolts passing through holes in the spokes and in rim clamps contacting the rim on the outer side. In the case of dual wheels, as shown in Fig. 13, the ends of the spokes are turned down at the sides to form shoulders, and steel rings approximating to angle irons in section are pressed over the reduced portions of the spokes against the shoulders thereon. The two rims are held between these spacer rings and clamps which are drawn together and into contact with the tapered mounting surfaces on the inside of the rims by bolts. Any standard rim can be used with what is known as the wheel spider; rims of different widths can be accommodated, and the spacings can be varied by changing the spacer rings.

Cast-steel wheels are being manufactured by the Dayton Steel Foundry Company, and a half-section of a Dayton dual wheel is shown in Fig. 14. Here the outer ends of the hollow spokes are closed, and are turned down to form a cylindrical surface, with a tapered portion on the inner side against which the mounting surface of the inner rim abuts. After the inner rim with its tire has been put in place, a spacer ring is passed over the wheel spider, and this is followed up by the outer rim, the whole assembly being held together by clamps and clamp bolts, as shown. An advantage claimed for these spoked cast wheels is that the spokes induce air circulation between the brake drum and rim, and protect the tires from excessive temperatures.

Farm-Tractor Wheels and Tires—Farm tractors not equipped with pneumatic tires usually carry the so-called agricultural wheels. These consist of a cast-iron or malleable-iron hub, a rim rolled up of boiler plate, sometimes with inturned edges, and steel spokes of either round or rectangular section which are secured to hub and rim by riveting or otherwise. Front wheels are provided with a skid ring, which usually is turned up of angle iron and bolted to the rim. The radial section of the ring digs into the soil and steadies the tractor, particularly at turns. Rear wheels are provided with driving lugs for use in field operations. There are a number of different types, the most familiar being angle-iron lugs,

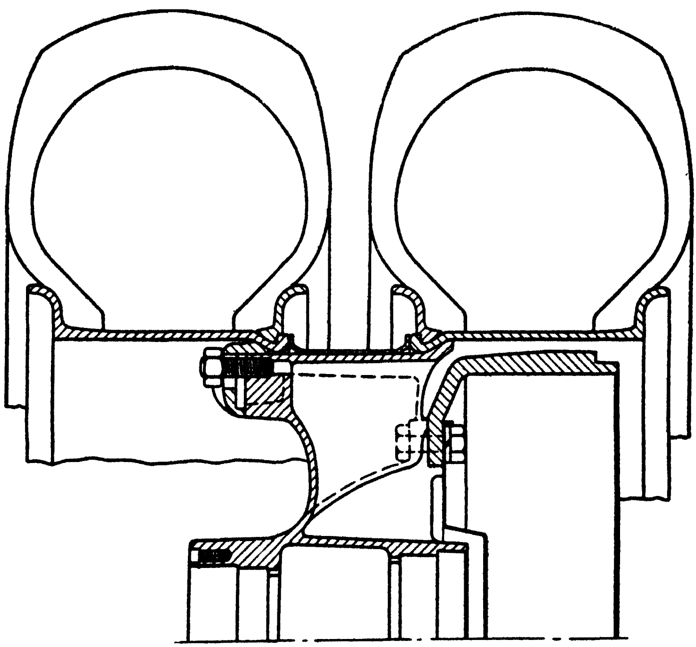


FIG. 14.—DAYTON CAST-STEEL TRUCK WHEEL.

bolted to the rim at an angle to the axis, and spade lugs, which are wedge-shaped and can be either cast or made in the press.

With rubber tires, disc wheels are used predominantly. The discs, which are conical in shape, are secured to the hub flange in much the same manner as in passenger-car and truck wheels. Shallow-well, drop-center rims are much used on farm tractors, and are provided with rim lugs by means of which the rim is secured to the wheel disc. There are square holes in the wheel discs for the attachment of cast-iron wheel weights.

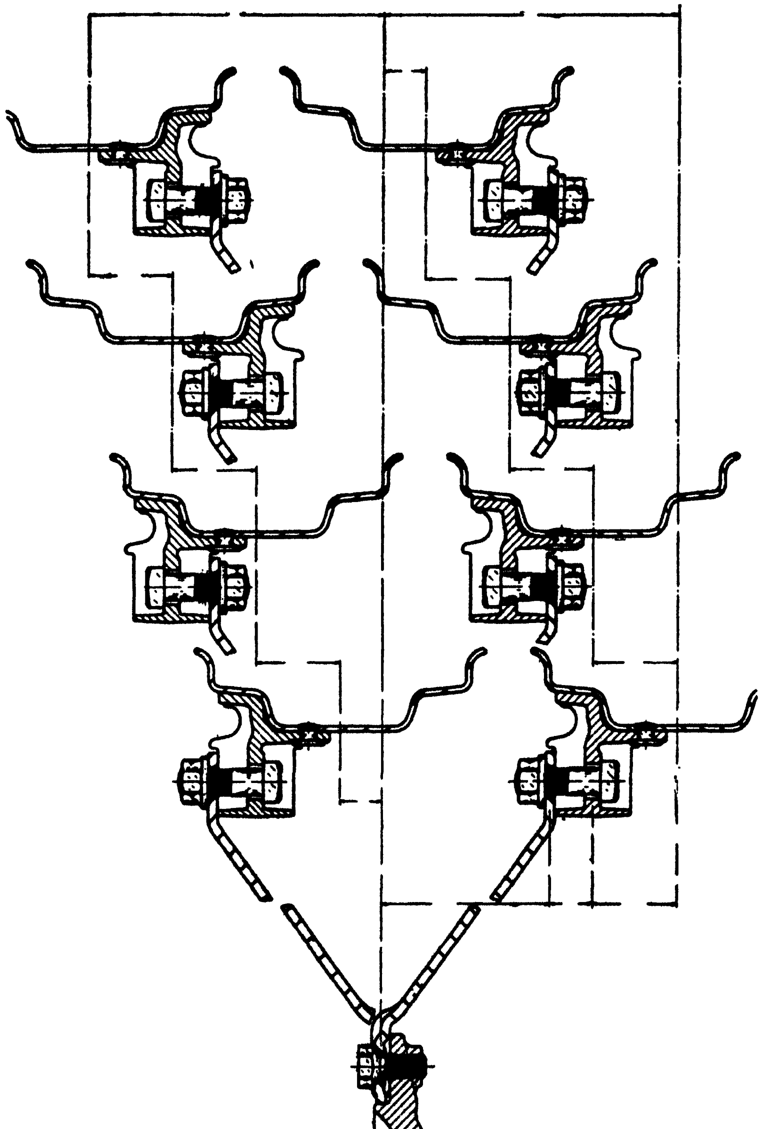


FIG. 15.—DIAGRAM OF TRACTOR TREAD ADJUSTMENT BY MEANS OF WHEEL DISCS AND RIMS (MOTOR WHEEL).

Fig. 15 shows a method of changing the tread of a tractor by a total of 28 in., in seven steps of 4 in. each. The various tread changes are obtained by reversing the conical discs on the wheel hub, moving the wheel lugs from one side of the wheel disc to the other, and by reversing the position of the wheel rim. To change the tread, it is not necessary to remove the disc from the hub, as the slots in the edge of the disc are wide enough to allow the rim lugs to pass through.

Tractor tires always carry relatively low inflation pressures, because tires of large section are needed to prevent the tractor from unduly packing the soil and from getting mired in soggy soil. This applies particularly to the tires on the driving wheels, the loads on which are greatly increased by the torque reaction when the tractor is pulling a heavy load. Inflation pressures for rear tires therefore are limited to maxima of from 12 to 20 psi, according to width and number of plies. In the case of front tires there is less need for large sections, because the driving torque—which with the tractor operating in low gear may become very large—transfers a large part of the wheel load from front to rear, and inflation pressures for front tires are limited to maxima of 28 to 44 psi. Load capacities of tractor front and rear tires are given in Tables I and II respectively. At inflation pressures less than the maxima recommended, the load capacity of front tires is reduced in proportion to the value of $p^{0.585}$, where p is the inflation pressure in psi, and that of rear tires in proportion to the value of $p^{0.78}$. The Tire and Rim Association states that when implements are mounted on the tractor, the front-tire load for any given inflation pressure may be increased up to 35 per cent, and the rear-tire load, up to 20 per cent.

Table I—Farm Tractor Front Tires

Tire Size	*No. of Plies*	*Max. Infl. Pressure*	*Load Capacity*
4.00/15	4	28	475
4.00/19	4	28	580
5.00/15	4	28	645
5.50/16	4	28	780
6.00/16	4	28	915
7.50/18	4	28	1465
5.25/21	6	36	950
6.00/12	6	36	870
6.00/16	6	36	1065
9.00/10	8	44	1650

Table II—Farm Tractor Rear Tires

Tire Size	*No. of Plies*	*Max. Infl. Pressure*	*Load Capacity*
9/24	4	16	1475
9/32	4	16	1675
9/36	4	16	1770
10/24	4	14	1600
10/28	4	14	1700
10/38	4	14	1970
11/24	4	12	1670
11/26	4	12	1725
11/28	4	12	1780
11/38	4	12	2070
11/38	6	20	2990
12/38	6	20	3500
13/26	6	20	3390
14/30	6	20	4110
14/34	6	20	4375
15/30	8	20	5080

Narrow- vs Wide-Base Tractor Rims—After wide-base rims had proved of definite advantage on motor vehicles, the question of their use on farm tractors arose, and tests on the subject were carried out jointly by the U. S. Rubber Company and the U. S. Department of Agriculture. Four-ply tractor tires of 11-in. nominal width, for use on 38-in. rims, were tested on rims of 9-, 10-, 11-, and 12-in. base width on different soils. In tractor work the factors that would count in favor of one base width over another are minimum rolling resistance and minimum slip under a given tractive force. It was found that on sand the narrow rims perform more satisfactorily than the wide ones; that on loam the width of rim has no appreciable effect on the performance, while on clay the wide rim is definitely superior, materially reducing both the rolling resistance and the slip or power loss. It was also found that on loose sand tires with low (½-in.) lugs outperform those with high (1¾-in.) lugs; that in loam the low lugs have a slight advantage, and that in clay there is no material difference in performance.

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