ELEMENTS OF AUTOMOBILE ENGINEERING

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A GENERAL INTRODUCTION TO AUTOMOBILE ENGINEERING FOR STUDENTS

BY

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PREFACE

THE aim of this book is to bridge the gap which exists between manuals of the simple "How It Works" variety and the more advanced textbooks of automobile design. At the same time an endeavour has been made to provide the student with a fair account of the principles upon which motor-car design is actually based. These are all strictly related to the true function of the vehicle, which is to provide a rapid, safe, and comfortable means of personal transport at a minimum of expenditure on purchase and operation.

Thus the book commences with a study of popular car sizes and weights in relation to the space required by the occupants. This develops into a discussion of the lay-out and constructional features of the body and frame, the performance expected of the complete vehicle, and the size of engine needed to provide this performance. The other mechanical components—transmission, suspension, steering, axles, and brakes—are then considered in relation to their special functions.

In endeavouring to treat so extensive a subject in a single volume of modest size the problem of what to include and what to omit becomes very difficult to solve wisely. It is not likely that the selection which has been made here will fully satisfy every reader. However, the author hopes that his book will fulfil its purpose as a general introduction to automobile engineering and that it will convey some idea of the absorbing interest of that highly developed and complex machine—the modern motor-car.

MAURICE PLATT

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ELEMENTS OF AUTOMOBILE ENGINEERING

CHAPTER I

CAR TYPES—PRICES—WEIGHTS—SIZES

Two fundamentals govern the design of a motor-car: the passenger accommodation which is to be provided and the price class in which the vehicle is to be sold. In Great Britain, owing to the heavy tax imposed upon rated horse-power, it has become a general practice to associate size and cost with engine rating. Thus various leading dimensions and prices have become recognized attributes of the Eight, the Ten, the Twelve, and (to a lesser degree) the Fourteen.

It is true that in every one of these classes the size of the cars concerned has gradually increased through the years. Concurrently, the habits of the buying public have gradually altered and the total number of cars sold annually has steadily increased, with varying fortunes for the individual horse-power classes.

General Tendency. A broad picture of the position at the time of writing (1939) will help the student to form an impression of current practice and future possibilities. But the preceding paragraph shows that it is a mistake to suppose that car sizes and rated horse-powers are fixed quantities in terms of price and popularity.

Here it may be well to emphasize the fact that the modern automobile engineer is vitally concerned with many matters which were once the exclusive province of the sales manager. Only by studying the market trends, the preferences of the motorist, and the features of competitive products, can he hope to design a thoroughly acceptable and successful motor-car.

The graph in Fig. 1 shows annual new car registrations in the principal horse-power classes over a period of years. The overwhelming numbers of the low-powered cars will be noticed, due principally to heavy taxation on rated horse-power and fuel. There is also the fact that other operating costs (such as insurance, garaging, tyres, etc.) all tend to increase with horse-power. The net result is that cars rated at 8 h.p. to 14 h.p. inclusive account for about 85 per cent of the new automobiles sold annually in Great Britain.

In comparison with the four well-established classes (8, 10, 12, and 14 h.p.) the intermediate ratings are of relatively small importance. Increasing prosperity helped the sale of cars rated at 22 h.p. and upwards over the period 1931 to 1937, but a setback occurred in 1938, a year when total sales fell by 15 per cent. The intermediate class, 15 to 21 h.p. (Fig. 1), has suffered mainly from the competition of the Fourteen, which has steadily improved in size and performance.

A complementary aspect of the matter is provided by a graph (Fig. 2) which shows that the share of total sales which has been secured by each class of car has varied during the past few years. The success of the Tens and Eights is clearly marked, likewise the fairly steady

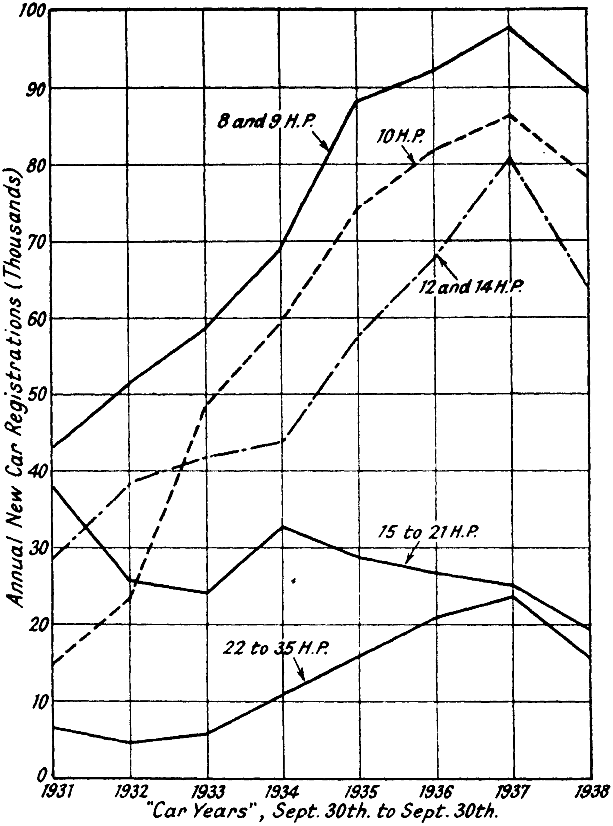

FIG. 1. CAR SALES CLASSIFIED BY RATED HORSE-POWER (GREAT BRITAIN)

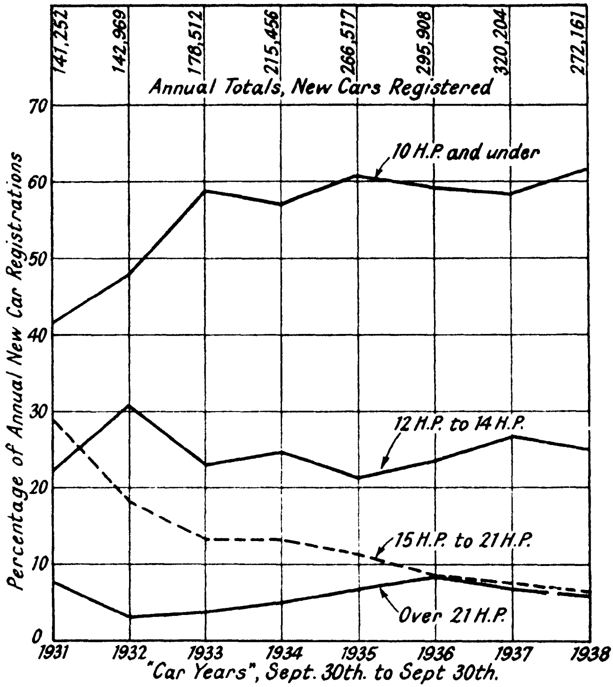

FIG. 2. DISTRIBUTION OF TOTAL ANNUAL CAR SALES (GREAT BRITAIN)

percentage enjoyed by the 12 h.p. and 14 h.p. models. In all classes an overwhelming proportion of the cars sold are saloons, for which reason no other body style is considered in this chapter.

Before concluding this brief market study it is interesting to notice that, as small cars have tended to "grow up," a gap has been left at the bottom. Thus the man who was content with the really small 8 h.p. model of, say, 1930 has now an extremely limited choice—unless he is prepared to buy a somewhat bigger vehicle. Already there are signs that the requirements of this class of motorist are being studied afresh.

Passenger Accommodation. As the function of a motor-car is to transport passengers, a logical study of the elements of automobile design must commence with the space requirements of a seated person. The space which is actually acceptable naturally varies upwards, from an irreducible minimum, according to the size and price of the car concerned.

Passenger space must be considered in three dimensions; two of these (length and height) are to some extent inter-dependent. This is because the leg-room needed by a seated passenger is reduced if the height of the seat (above the floor) is increased. Extremes are represented by a sports car, with low seats requiring an almost horizontal leg position, and the cab of a heavy lorry in which length is economized by high seating. In Fig. 3 these extremes are illustrated for an average man, 5 ft. 9 in. in height, with laden cushion heights of 6 in. and 18 in. respectively.

In recent motor-cars the trend has been towards higher seating and, at the present time, unladen cushion heights vary between 12 in. and 15 in. (above floor level) in most popular models. The laden height will be reduced by 2 in. to 2½ in. according to the compression of the cushion and the weight of the occupant.

Fig. 4 shows the approximate effect of compression on the cushion and squab (i.e. the back of the seat) by broken

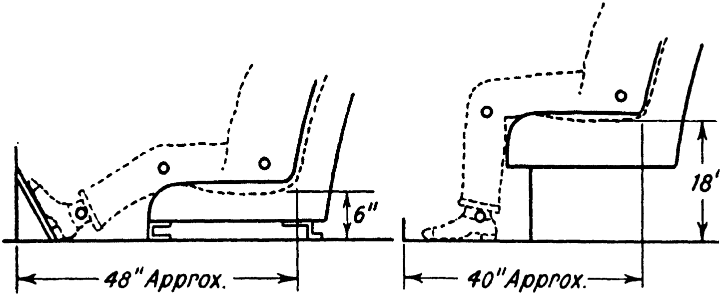

Fig. 3. Leg-room is Reduced when Seating Height is Increased, as shown by these Extremes

lines, the full lines representing the unladen contour of the seat. The shapes and angles of the cushion and squab, although very important to passenger comfort, are the business of the trim and upholstery specialist and need not concern us here, except dimensionally. Thus we should note that cushion rake is usually between 10 and 15 degrees and that the angle included between cushion and squab is often about 100 degrees. Cushion thickness (or depth) is also important, as it affects head-room; minimum figures for comfort, with sprung cushions, are about 7 in. front and

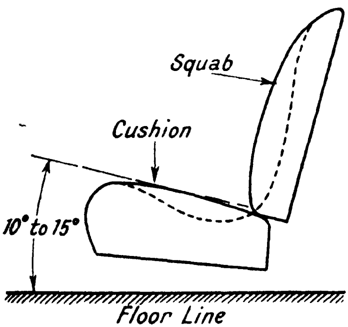

Fig. 4. Typical Seat Contours: Unladen (*full lines*) and Laden (*broken lines*)

$4\frac{1}{2}$ in. rear, measured vertically and uncompressed. Pneumatic cushions enable depth to be saved; an important point in low-built sports cars.

We can now place our average man in an average seat in order to find out how much space he requires (Fig. 5).

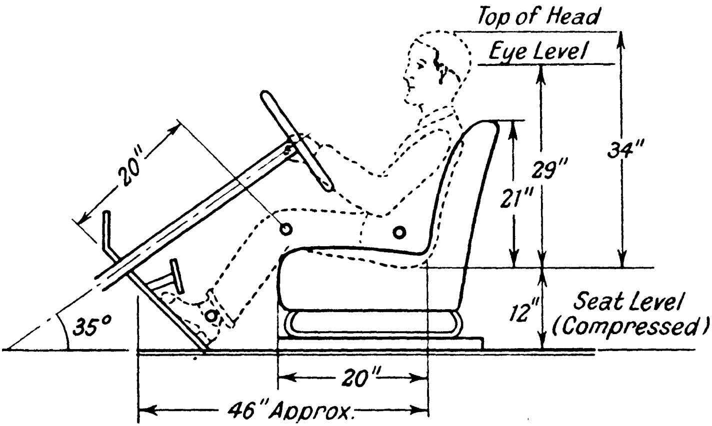

FIG. 5. AN "AVERAGE MAN," 5 FT 9 IN. IN HEIGHT, IN RELATION TO SEATING SPACE, EYE-LEVEL, HEADROOM, ETC.

The broken lines again indicate the approximate contour of the upholstery when compressed. Taking the minimum height of the loaded cushion as a datum (12 in. above the floor), the eye-level of our average individual will be 29 in. higher and the top of his head is 34 in. above the datum. If he is to wear a hat comfortably, he will need an extra 2 in., giving a total height, from floor to interior of roof, of 48 in. minimum. A common figure is actually 49 to 50 in., giving latitude for taller people.

In passing, it may be noted that the tall man often owes his height to length of leg and does not then require extra head-room when seated.

For our average man a leg-room of about 46 in. is needed, with the seat height shown in Fig. 5. It is the usual practice

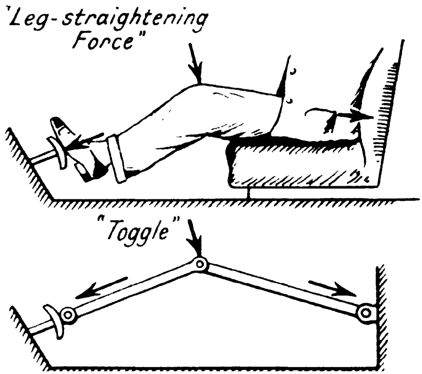

FIG. 6. THE DRIVER'S LEG OPERATES THE BRAKE PEDAL BY A TOGGLE ACTION AT THE KNEE JOINT

to mount the seat slidably, with a 5 in. range of adjustment, so as to allow for individual leg-room requirements. This is particularly important for the driver, as his comfort and the safe control of the vehicle depend upon his seating position in relation to the pedals. These he operates by a toggle action of the leg: a powerful mechanism, in the correct adjustment, but one which quickly loses effect if the distance be either too great or too small (Fig. 6).

Arm reach to the steering is capable of wider latitude, partly because the driver has a choice of hand position around a raked wheel. Nevertheless, there is a growing tendency to provide a steering-wheel adjustment (towards and away from the column), and a means for altering column rake is not uncommon in the more expensive cars. Pedals adjustable for reach have also been provided in certain cases.

Individual seats for the driver and front passenger are general in British cars, but a one-piece full-width seat is standardized in most American and continental productions. Single seat-cushion widths vary from 20 in. to 24 in.; cushion depth (front to rear) varies from 19 in. to 21 in.

The rear-seat design of the conventional 4/5-seater car is governed by much the same considerations in respect of height from floor, depth and thickness of cushion, height of squab, head-room, etc. There are some restrictions, however, which are absent from front-seat design.

First there is the fact that in a small car, after providing the driver with reasonable leg-room, the maximum body length permissible does not leave a very generous balance for rear-seat leg-room. One of the advantages of raising the front seats on tubular frames, clear of the floor, is that space is left beneath them fot the feet of the rear passengers whose leg-room is thereby increased. Another expedient, found in some continental cars, is to use a hammock type of front-seat squab so as to increase knee-room at the back (Fig. 7). Floor wells, once widely employed, have gone out of fashion owing to their awkwardness when entering or leaving the rear compartment.

Broadly, it can be said that in most 10 h.p. cars it is possible to provide approximately equal and reasonable leg-room dimensions, front and rear, with the front seats set mid-way along their sliding adjustment. This requires an overall body length, from the scuttle to the back of the

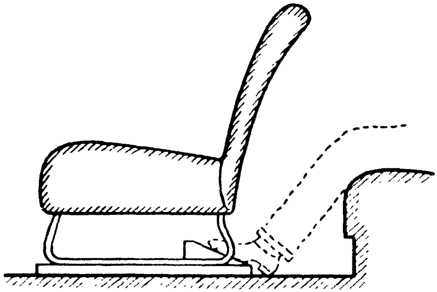

FIG. 7. EXPEDIENTS FOR INCREASING THE LEG-ROOM AVAILABLE TO REAR-SEAT PASSENGERS

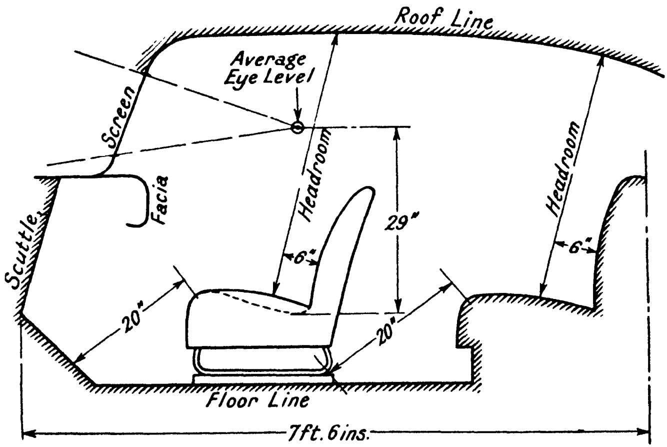

FIG. 8. LEADING INTERIOR DIMENSIONS OF A SALOON BODY; AVERAGE FIGURES FOR THE 10 H.P. CLASS ARE QUOTED

rear-seat squab, of about 7 ft. 6 in. (Fig. 8). In 8 h.p. cars the shorter body implies reduced rear-seat leg-room; in cars rated at 12 h.p. and more the longitudinal space is progressively increased.

Head-room. Next, as regards head-room, the comfort of the rear-seat passengers is often sacrificed to some extent in a desire to achieve a fashionably curved roof contour. It is true that the lounging position of a passenger does not demand quite so much head-room as does the alert, more upright posture of the driver. Nevertheless, this important dimension is too frequently cut down to an unreasonable extent.

Head-room should logically be measured along the line corresponding to the body and head of a seated person. This can be taken as approximately 10 degrees from the vertical, and 6 in. ahead of the squab (Fig. 8). In plan view, the measurement should be made about 9 in. distant from the side arm-rest; this point is important in an arched roof, where centre-line height is deceptive. Measured in this manner, rear head-room should not be less than 37 in.; front head-room is usually 39 in. to 40 in. These figures refer to the uncompressed cushion.

A third and major restriction on rear passenger accommodation is the width available between the wheel arches. Only in fairly large cars is it possible to provide a track sufficiently wide to accommodate a "three-seater" cushion. In small cars every effort is made to economize on wheel clearances and wheel arch shapes, but, even so, the narrow track is a severe limitation upon rear-seat width.

The wheel arches are usually built up and upholstered to form side arm-rests between which the cushion is fitted. A minimum cushion width to seat two people is 36 in.; above this the body of a really small car may be capable of giving an elbow and shoulder width of about 45 in. (Fig. 9). Rear cushion widths range from 38 in. to 41 in. in 10 h.p. cars, and from 40 in. to 43 in. in 12 h.p. cars. These differences may seem small but they are very important to rear-seat comfort. Cushions which reach 48 in. in width are often claimed to seat three people and will actually accommodate two adults and a child. For a real three-seater cushion a minimum width is 54 in. Some typical seating dimensions, and corresponding track widths, are listed in Table I.

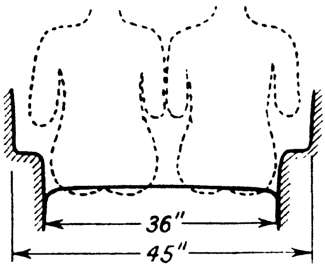

FIG. 9. MINIMUM REAR-SEAT CUSHION AND ELBOW WIDTHS

TABLE I

LEADING DIMENSIONS

(WELL-KNOWN DE-LUXE SALOONS; 1939 MODELS)

	10 hp.		12 h.p.		14 h.p.	
	Min.	Max.	Min.	Max.	Min.	Max.
Wheelbase, in.	90	94	96	106	104½	114
Rear Track, in.	45	50	49½	56	51	56
Rear cushion width, in.	37	41	40	45	42	46
Kerb weight, lb.	1850	2130	2140	2900	2500	3050

Driving Vision. The safety with which a car can be handled under modern traffic conditions is largely governed by the vision afforded to the driver, yet body styles are often elaborated with but little regard for this essential matter. The effect of eye level in relation to bonnet height is demonstrated in Fig. 10 which shows a typical 10 h.p. car in side view; from an eye level 29 in. above the cushion the nearest point at which the road can be seen (above the bonnet centre line) is 48½ ft. distant. If the eye level is dropped 3 in. this blind range is nearly doubled.

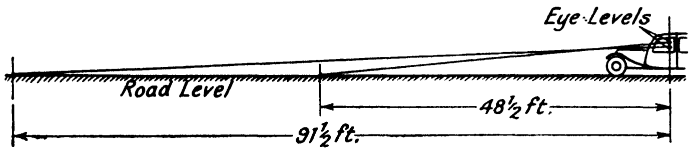

FIG. 10. EYE-LEVEL IN RELATION TO BONNET HEIGHT AND THE RANGE ON FORWARD VISION AT ROAD LEVEL

In hilly districts the combination of steep gradients and twisting roads makes a car with poor forward vision really dangerous to handle. No doubt it is for this reason that certain continental makers have developed a style of bonnet which falls away in a curve towards the front wings.

Referring back to Fig. 8, the upward angle of vision beneath the peak of the roof is almost as important because it controls the facility with which traffic signals, road signs, etc., can be seen, and also because it affects driving fatigue. In order to avoid blind areas the pillars should be as narrow as is possible, consistent with structural requirements. In planning generous window lay-outs the rear window must not be forgotten, particularly in its height relationship with a mirror mounted above the windscreen.

Typical Chassis Lay-outs. We can now see how these ideas of passenger accommodation fit in with the general planning of a motor-car. The four-seater 10 h.p. class provides a good starting-point from which smaller cars are scaled down to "least practicable" seating dimensions, while larger cars are scaled upwards to give progressively greater space and comfort.

The outline of the body can be regarded as an "envelope," shaped to give the head-room, seating width and length that we have already discussed. The next step is to arrange this envelope in relation to the wheels, axles, engine and other mechanical units in the manner that seems to offer the greatest economy in respect of weight and overall dimensions.

As there is a very large number of ways in which this arrangement can be carried out, a typical example will be of greater practical utility than a prolonged discussion. Such an example is shown in elevation and plan, with leading dimensions, in Fig. 11; the car in question is the Vauxhall Ten saloon in its 1939 form.

Characteristic of the modern small car are the relatively small space occupied by the power unit and the short distance separating the scuttle line from the front-wheel centres. The floor and roof heights shown in the elevation are those corresponding to a loaded condition. With a wheelbase of 94 in., the rear axle is located just behind the rear seat and the outswept tail of the body provides reasonable luggage space within a moderate overhang.

The leading dimensions of a range of well-known cars of the 1939-model series are shown in Table I. It will be noticed that, with few exceptions, a considerable degree of uniformity has been reached, although in outward appearance and mechanical design there are still some wide variations between one make and another.

As compared with an equivalent series of cars built, say,

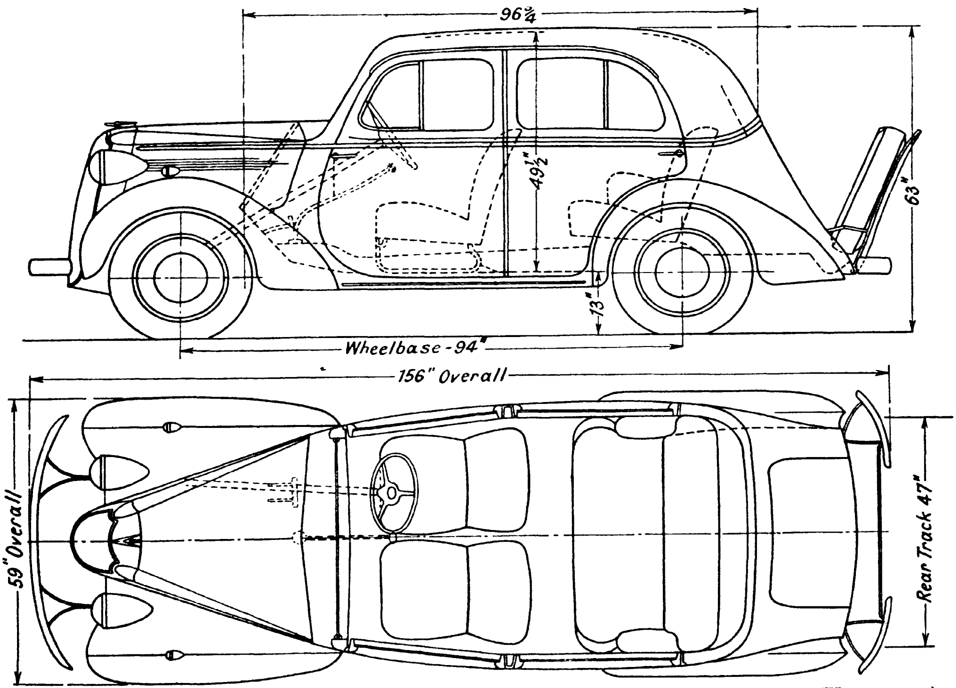

FIG. 11. ELEVATION AND PLAN VIEWS OF A 1939-MODEL 10 H.P. SALOON CAR (VAUXHALL), SHOWING LEADING DIMENSIONS

in 1932, the major differences are found in the rearrangement of the chassis in relation to the body. Thus the engine is now placed so far forward that it is often necessary to employ a stepped sump in order to clear the front axle or, in cars with independent springing, to give room for the front cross member. The whole body has been shunted forwards along the frame until the scuttle line is only about 12 in. from the centre line of the front wheels. As a result, the rear-seat passengers can now be placed just ahead of the back axle and in a lower position than was possible when axle clearance had to be left beneath the cushion. Space has thus been found for a body extension to house the petrol tank, spare wheel and luggage, without too much rear overhang.

While these developments have been taking place, body floors and frame side members have been lowered by several inches, and because of this trend (together with the widening of the body) running boards have been discarded in many cases. To permit a passenger to take a single step from the road into the car, the height of the floor above the road should not be more than 13 in. and the extent to which the body sill projects must be reduced to a minimum. Allowing for an interior height of 49 in. or so, the complete car will stand at 5 ft. 3 in. approximately, road to roof, when laden.

Wheelbase and track are dimensions which have not suffered nearly so great a change as the lay-out of the car above the wheels; it is not far wrong to say that the superstructure has been moved forwards in relation to axles which remain at much the same distance apart as formerly. In this way the forward end of the car, which is necessarily narrow because space must be allowed for steering the wheels, has ceased to take up an uneconomical proportion of the wheelbase. At the rear end the passenger space and the doors have become less restricted by the axle and wheels because they have been moved forwards.

Laterally, the rear wheels continue to encroach upon seating space except in a few large cars and in, perhaps, two or three continental makes of small car. In these exceptions an abnormally long wheelbase and/or the adoption of an unusually forward position for the body have enabled the rear seat to be placed wholly ahead of the wheel arches.

The ideal of a flat floor is very difficult to achieve in conventional cars owing to the presence of a gearbox and propeller shaft under the body. Thus the floor is commonly divided into halves by a longitudinal ridge comprising a gearbox and propeller shaft tunnel. An advantage shared equally by the front-engined front-drive car and the rear-engined rear-drive model is a space amidships clear of the transmission.

These remarks on motor-car lay-out and development have been mainly concerned with medium-sized models ranging from 10 h.p. to 14 h.p. in rating. Smaller cars are tending in the same direction but, owing to their shortness, there is still a dispute for room between the back axle and the rear seat. Although four-door bodies are now available in many 8 h.p. cars as alternatives to the two-door style, it is difficult to provide a wide enough entrance at the back, ahead of the rear wing, for easy entrance and exit.

Conversely, in the large car there is not the same need to keep floor and roof heights down to minimum dimensions (as we shall see later), while the wheelbase and track are very materially extended. Furthermore, the space required by the transmission relative to the floor has been reduced by several inches in most American cars by the use of hypoid bevels for the final drive; a practice which requires a productive technique that has not yet been widely disseminated in Europe. Consequently it is usually found possible to eliminate the central tunnel, to provide wide rear doors, and to place the rear seat well away from the axle.

The Question of Weight. Owing to its fundamental influence upon cost, performance, fuel consumption and many running expenses, the weight of a car is one of its most important attributes. It can appropriately be mentioned here because shape, size and lay-out have as big an influence upon the weight of the vehicle as the individual design of its various components.

First, we need a clear definition of what we mean by the weight of a car; this is particularly desirable when comparing one vehicle with another and yet it is often misunderstood. A car as produced by the factory carries no petrol, oil, or water and its weight in this condition can conveniently be described by the term "shipping weight." It is a useful means of measurement because it represents a net weight of material which is independent of variables such as the capacities of the petrol tank and the radiator. On this basis the cost of a car can be compared with that of a competitor, dividing the lowest selling price of the model concerned by its shipping weight in lb. At the present time the majority of low-priced British cars range between 1s. 6d. and 1s. 10d. in cost per lb.

When studying performance we require the weight of a car in the condition in which it is driven on the road. The simplest plan is to add to the shipping weight the total weight (in lb.) of the petrol, oil and water required to fill the various components to top levels. The Americans use the term "kerb weight" for the total reached in this way, meaning the weight of the car as it stands at the kerb, filled up and ready to drive.

It remains to add an allowance for the occupants which can conveniently be taken at 150 lb. per person. Individual designers differ as to the number of people for which they make allowance in performance and other calculations, but for 4/5-seater cars a fair average would seem to be 450 lb., considering that loads vary in practice between extremes of five people with luggage, and "driver only."

In studying a new or modified design the weight of individual components must be estimated and compared. In order to do this in a logical and satisfactory way each manufacturer follows his own established practice when dividing the parts into groups. There are many cases in which a definite ruling must be made to avoid confusion; thus a clutch housing might be regarded either as part of the engine or a portion of the gearbox; brake drums might either be included with the braking system or with the wheel hubs, and so on. There would be no point in describing such a system here, as the subdivision of weights is largely a matter of convenience. The important point is to adhere to a uniform classification for every car examined.

Although in many British factories complete car weights are still expressed in cwts. and quarters, it is in many ways more convenient to use only pounds and tons: a practice maintained throughout this book. American engineers make good use of the "short ton" in their calculations; a unit equal to 2000 lb.

The usefulness of weight analysis can be extended by expressing the weights of various components as percentages of the total. When comparing cars of similar type this method quickly indicates any unit which is heavier than "average." A typical analysis is given in Table II, in round figures based upon a number of medium-sized cars.

TABLE II

A SPECIMEN WEIGHT ANALYSIS

Parts concerned	Weight, lb.	Per cent of Shipping wt.
Body (complete) and frame	900	40·0
Front and rear suspension and axles	300	13·4
Braking system and drums	90	4·0
Engine and clutch	400	17·8
Gearbox and propeller shaft	75	3·3
Steering gear and linkage	30	1·3
Five wheels and tyres	180	8·0
Wings and bonnet	110	4·9
Miscellaneous parts	165	7·3
Total = Shipping weight	2250	100·0
Add Petrol, oil and water	100	—
KERB WEIGHT	2350	—

Productive Equipment. As this chapter is concerned with fundamentals it would not be complete without reference to the way in which the productive equipment available, in relation to the quantities in which a car is to be produced, influences the basic design of the vehicle concerned. It is part of the duties of the sales department to provide a forecast of sales volume which is based upon a study of market conditions, new car registrations, the price at which it is hoped to sell the new model, and other similar considerations.

So far as British manufacturers are concerned, few popular cars have ever exceeded an output of 20,000 vehicles per year of a single type. A three-year run without any major change in appearance or design may therefore mean that the original dies, jigs and tools will produce some 60,000 units before becoming obsolete. In contrast with this situation a medium-priced American car enjoys an output of over 60,000 units *per year*, and in the popular low-priced types this figure is, of course, very greatly exceeded. Consequently the American manufacturer can face a very much higher tool and equipment cost than can any British maker. This factor, together with the advantage of buying materials and components in very large numbers, accounts in part for the fact that the basic cost of the American car, in pence per pound, is roughly half that of the more popular British cars.

As will be seen in Chapter II, a considerable proportion of tooling costs are absorbed in the body structure, if this be of the all-steel type. Consequently a body of this kind—particularly one of the self-supporting variety (needing no frame)—can only be contemplated when a large output is assured. For smaller outputs the amount of tooling required must be progressively reduced, and the extent of hand-worked components proportionately increased. The increased cost of production involved and the consequent increase of list price are made commercially practicable by offering certain individualities in style, performance and equipment for which limited numbers of critical users are prepared to pay. The ultimate extension of this policy is seen in the few exclusive types of car which are built as chassis only and are sold with bodywork specially designed and constructed by a coachbuilding concern to the wishes of the individual purchaser.

It is also worth noting that makers with limited outputs are assisted in their competition with lower-priced cars of similar size and performance by the fact that their productive programme is more flexible and enables changes of design to be made more easily. Nevertheless the total number of makers tends steadily to decrease and it is probable that in Great Britain the six leading manufacturing groups, producing ten makes of car, can now claim over 85 per cent of annual new car registrations, in addition to the lion's share of export business.

In the United States this trend has gone much further, so that three leading producers claim nearly 90 per cent of annual car sales, while the number of makes on the market has been reduced to 20. Although these facts may seem to have only an indirect bearing upon the work of the automobile engineer, the influence which they exert upon the trend of design and upon methods of manufacture is very great indeed.

CHAPTER II

THE CAR AS A STRUCTURE

In the preceding chapter we saw how the leading dimensions of a car are built up around the seating space accorded to the passengers, and the way in which the engine and wheels are disposed in relation to this body space. The next step is to consider how the structure of the vehicle can best be designed in order to give the strength and rigidity necessary to carry the load and to maintain all the parts in reasonable alignment. The design of this structure also has an all-important influence upon the durability of the vehicle and the degree of quietness with which it operates.

" Frameless " Construction. Ten years ago the frame and the body were almost invariably treated as separate components, brought together on the final assembly line and secured by bolts for mutual reinforcement. Nowadays it is appreciated that, even when these parts are built independently, they function as a single structure in the vehicle. For cars built in quantities there is an increasing tendency to bring the design and construction of body and frame into the closest possible relationship; a trend which finds its logical outcome in the frameless or "integral" type of saloon car. As a result of this policy of development, together with the great reduction in the distance separating the scuttle from the front wheels, the overall stiffness of cars in torsion has increased enormously without any corresponding alteration in weight. At the time of writing, new methods of construction are being exploited to maintain this stiffness while effecting a valuable saving in weight.

One of the factors which has helped engineers to evolve a light yet rigid structure is the overwhelming popularity of the saloon style of body for low- and medium-priced cars. This provides a depth and width of cross-section such that great rigidity can be obtained from relatively thin sheet-steel pressings. The body, instead of serving mainly as an "envelope" for the passengers (and giving added rigidity to the frame as an incidental), has become an essential structural part without which the frame would be incapable of working satisfactorily. And in the integral cars it has supplanted the frame entirely. In the same period of development the frame, still used in very many cars of all sizes (and essential in some form for carrying open or convertible bodies), has also been lightened and stiffened. This has been done largely by using box-section side and cross members, which have much greater torsional rigidity than have open channels. Furthermore, methods of connecting these members have been improved, notably by the application of welding technique in ways which have also contributed to the strength and stiffness of the all-steel body.

In Fig. 12 a simple rectangular frame is shown, carried by trestles and supporting a distributed load as indicated by arrows. The structure is subjected to simple bending, and the stresses so produced can be calculated from the well-known beam formulæ. They will be proportional to the distance between the supports (L). The structure of a car is loaded in this way when the four wheels stand on a horizontal plane—a comparatively rare occurrence. Another point is that in a car structure we are very much concerned with rigidity; indeed, if the job is stiff enough it is pretty well certain to be strong enough. The importance of the distinction can be seen by studying the beam formulæ for deflections which show that the "sag" is proportional to L^3 and is inversely proportional to the modulus of elasticity of the material used. This modulus is much the same for all classes of steel, so that (as the stresses are low)

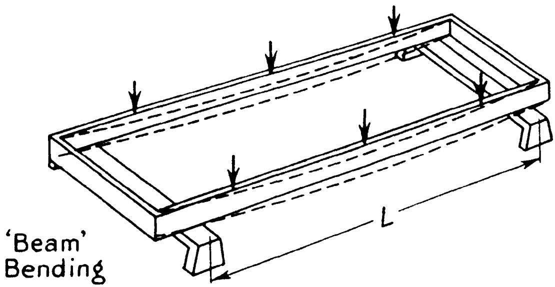

FIG. 12. SIMPLE BENDING LOADS APPLIED TO A RECTANGULAR FRAME

there is no point in using high-tensile materials in a motor-car structure.

The diagram of a simple frame is repeated in Fig. 13, but here the loading is by couples which produce torsion. A car experiences this form of twisting as the wheels roll over uneven surfaces, and the desire to provide adequate torsional stiffness is a major reason for using frame members of box section.

The end view in Fig. 13 shows clearly what is happening to the corners of the twisted frame, and it is easy to see that unless these connexions are capable of maintaining their rigidity the torsional resistance of the side members and

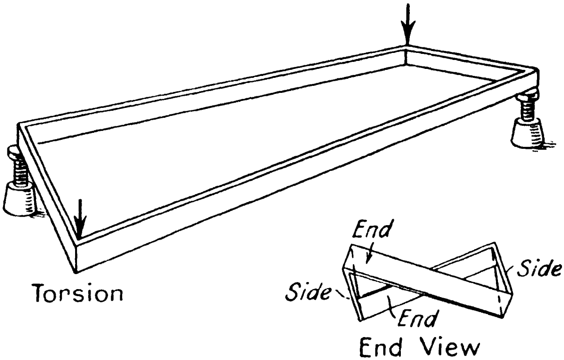

FIG. 13. TORSION OF A RECTANGULAR FRAME PRODUCED BY COUPLES

cross members cannot properly be exploited. As an example of this statement the sectional views in Fig. 14 show two ways in which a cross member and side member (each of

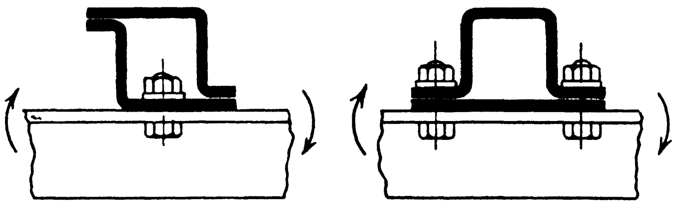

FIG. 14. ALTERNATIVE CONNEXIONS BETWEEN A SIDE MEMBER AND A CROSS MEMBER

box section) might be joined. In the first case the narrow base and central fixing will allow one part to rock on the other; in the second, a much stronger job results from extending the base and employing widely spaced bolts.

The cruciform bracing which has been so widely employed to stiffen car frames in torsion is usually made from two parts of V-shape which are bolted (or welded) to the side members and are joined at the centre to form an "X" (Fig. 15). The ability of this bracing to resist torsion depends

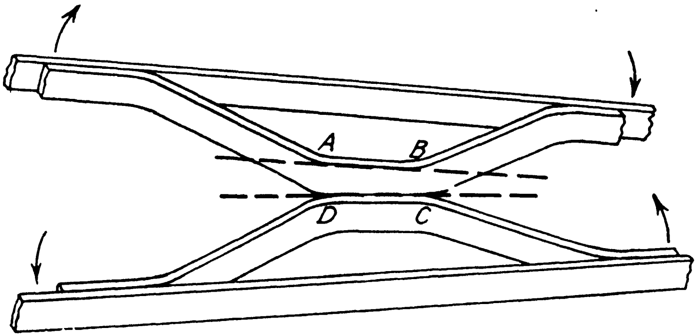

FIG. 15. STIFFENING EFFECT OF A CRUCIFORM BRACING AGAINST TORSION DEPENDS UPON A RIGID CONNEXION AT A-B-C-D

upon the rigidity of the connexion at the centre (*A-B-C-D*), and many designs have been open to criticism for weakness in this respect.

The Body as Reinforcement. The depth of a saloon body when viewed in elevation, and the size of its cross-section, show at once the value of this component as a reinforcement for the frame, in respect of bending and torsion respectively. For the majority of saloon cars it would be much nearer the mark to say that the frame merely reinforces the body, while it also provides convenient attachments for the power unit, suspension, and steering. The stiffness of a beam is proportional to the cube of its depth, so that the relatively thin material in the body (usually 20 B.G. sheet, the thickness of which is ·0392 in.) can contribute more rigidity than the much thicker, but very much shallower, side members of a frame. Similarly the stiffness of a thin-walled tube in torsion is proportional to the cube of the diameter, and a well-braced body is roughly equivalent to a tube of large size.

As in all sheet-metal structures, the rigidity of a saloon body, in relation to the efficient use of a given weight of material, depends very largely upon the success of the designer's efforts to prevent the occurrence of local buckling. For example, the door openings are an obvious source of weakness which can be met by using a rigid central pillar, reinforced roof shoulders, and sills of box section. Other necessary openings which reduce torsional stiffness are those required for the windscreen, rear window, and luggage space. In some cases a steel instrument panel provides a tie at the base of the windscreen pillars, while other makers have used lattice bracing behind the rear seat to stiffen the structure just ahead of the luggage compartment.

In comparison with the cars of ten years ago, modern examples gain very greatly in stiffness by the reduced overhang between the front wheels and the scuttle; thus the length of unsupported frame which extends beyond the body has been greatly curtailed. This is especially important in that the vogue of the rubber-mounted power unit makes it very difficult to provide any sort of cross member between the front of the frame and the rear end of the gearbox.

The way in which a steel body can be utilized to support beam loading (L, L, etc.) is shown diagrammatically in Fig. 16. Points of support are at the cross member which carries the independent front suspension (S_1) and at the front and rear ends of the rear springs (S_2 and S_3). The load is the weight of the body and its occupants together with that of the engine and other mechanical components. It will be seen that, provided there is a sufficient number of

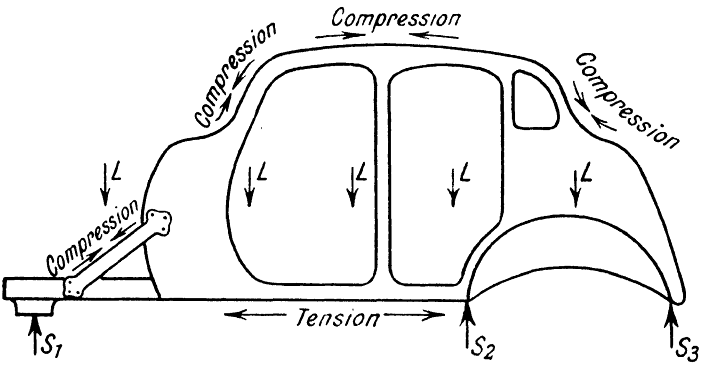

FIG. 16. BEAM LOADING OF A SELF-SUPPORTING SALOON BODY, SHOWN DIAGRAMMATICALLY

sound connexions between the body and frame, the bending effect of the load can be resisted by compressive stresses in the roof and tensile stresses in the sills and the floor. Another proviso is that the front and rear ends of the body should be strong enough to throw a useful part of the stressing into the roof.

Examples from Current Practice. In the space of a single chapter we can describe only a few of the many variations of body and frame design which are used in present-day cars. For nearly all the larger types, and for many smaller vehicles, a full-scale frame is used; this practice has an advantage in that the manufacturer is able to offer a range of body styles including the open and convertible varieties, which have little inherent rigidity as compared with a saloon.

A good example of modern frame design is provided by

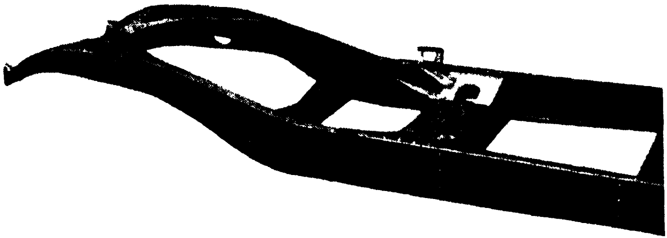

FIG. 17. EFFECTIVE BRACING OF THE FRONT END OF A MODERN FRAME (16 H.P. ROVER)

the 16 h.p. Rover, as shown in Fig. 17. The side members are of channel section, but it will be noticed that they are "boxed" by extensions of two bracing members which provide a triangulated reinforcement for the front end. This boxing extends rearwards as far as a rigid cross member which is located behind the power unit.

For the small cars in which the British designer is perforce chiefly interested the saloon is the predominant type, and there is every incentive to reduce both weight and floor height to a minimum. Consequently, increasing use is being made of the strength and stiffness of the steel body, and this trend is accelerated by the fact that the depth available for a frame is becoming "squeezed" between the descending floor and the irreducible demand for road clearance.

In many small cars the frame has become quite a light and simple component with relatively shallow side members which are reinforced by the adjacent sills of the body. In some cases the sills and the frame are connected by a

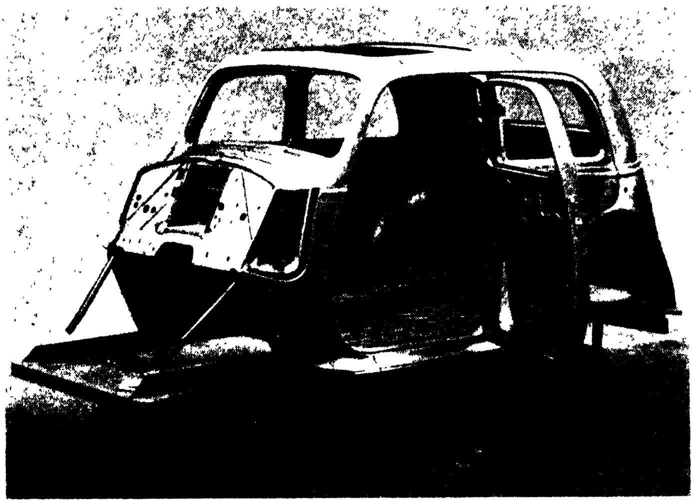

FIG. 18. SELF-SUPPORTING BODY SHELL OF THE 10 H.P. VAUXHALL SALOON

number of bolts, while in others—notably in the 8 h.p. and 10 h.p. Ford saloons—they are joined by welding.

An interesting type of integral body is that developed for the 10 h.p. and 12 h.p. Vauxhall models in which the all-steel saloon body is self-supporting but carries a short detachable frame which projects at the forward end (Figs. 18, 19, and 20). The side members of the frame are

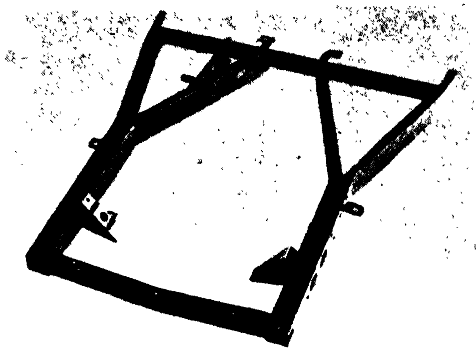

FIG. 19. THE SHORT FRAME WHICH IS BOLTED TO THE FORWARD END OF THE VAUXHALL BODY

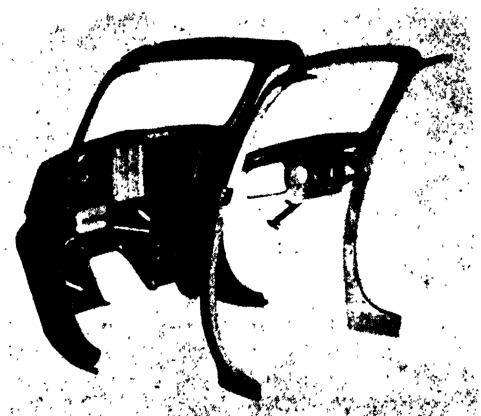

FIG. 20. INNER AND OUTER PARTS OF THE VAUXHALL SCUTTLE STRUCTURE, SHOWING THE DIAGONAL STRUTS

bolted to the sills beneath the scuttle, and diagonal struts (of steel tubing) provide the necessary triangulation. Each strut is continued upwards (within the scuttle) to the base of the corresponding windscreen pillar.

The way in which the scuttle structure is built up from an outer shell and an inner framework is shown in Fig. 20, the upper part of each strut being also visible in this view. From the scuttle rearwards, beneath the doors, body sills of box section take the place of a frame and are continued in the form of reinforcements over the wheel arches to the rear bumper supports. The stiffening effect provided by the propeller shaft tunnel and by the cross member between the rear wheel arches will be noticed in Fig. 18.

Another kind of integral construction is exemplified by the 10 h.p. Morris saloon (Figs. 21 and 22). Here the body shell and scuttle are extended forwards to the extreme front end of the car, there being no separate frame. In order to provide sufficient depth of sheet metal to resist "hinging" at the base of the scuttle, the inner portion of each front wing and wheel arch is used as an integral part of the job and diagonal bracing members are also built into the structure. The floor, which is shown separately in Fig. 22, has side channels which match up with the body sills to form box sections. Both Morris and Vauxhall bodies are, of course, built up in assembly jigs, from a large number of individual pressings, by spot welding at close intervals.

Another example of modern practice is found in the latest 8 h.p. and 10 h.p. Austin models. Here the designers use what might be called a "platform frame," which consists of a steel floor, suitably reinforced along each edge, and by cross members, all these parts being pressings (Figs. 23 and 24). Although the steel body is built as a separate

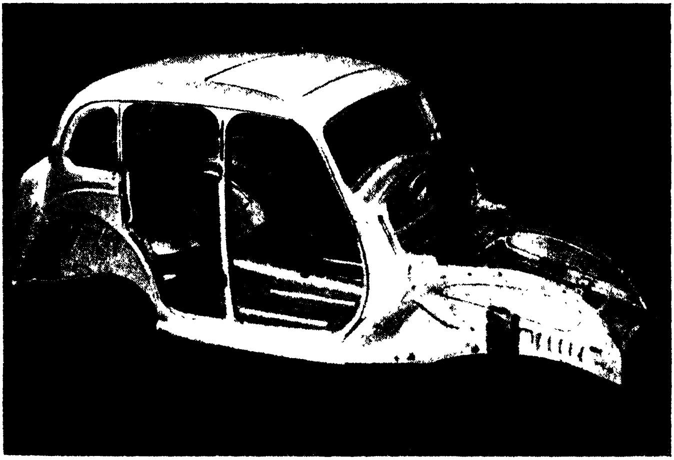

FIG. 21. BODY SHELL AND SCUTTLE STRUCTURE OF THE 10 H.P. MORRIS INTEGRAL SALOON

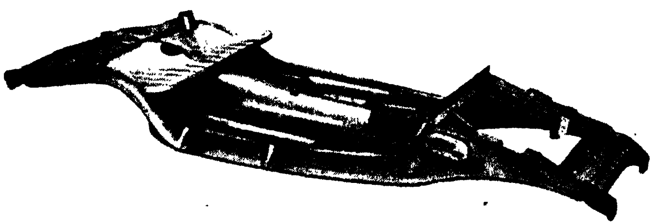

FIG. 22. THE FLOOR, OR "UNDER-BODY," OF THE MORRIS TEN INTEGRAL SALOON, SHOWING REINFORCEMENTS

component it is joined to the platform by a number of closely spaced bolts; the sills form side members of box section in conjunction with the reinforcement along the

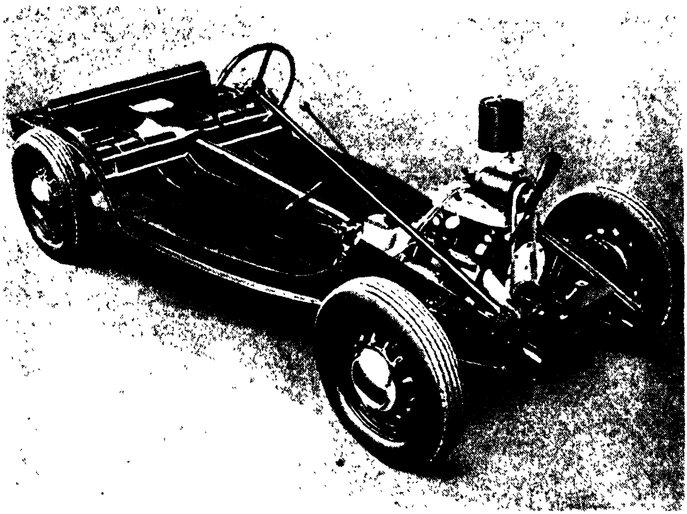

FIG. 23. "PLATFORM" FRAME OF THE AUSTIN EIGHT, TO WHICH THE BODY IS SECURED BY A NUMBER OF BOLTS, FORMING BOX-SECTION SILLS

edges of the floor. The detailed view in Fig. 24 shows how these parts meet one another; the cross member which is provided just in front of the rear wheel arches will also be noticed.

Design of Body Components. The design of a complicated structure such as a modern steel body is not a suitable subject for mathematical analysis. It has been evolved by a long process of practical study on the road, using full-scale components which are laboriously produced (by skilled labour) to emulate the parts which will eventually be formed between dies. Experimental work can also be

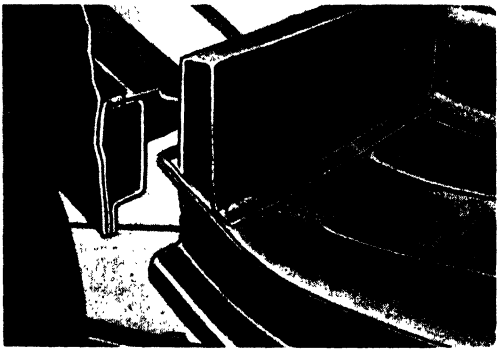

FIG. 24. REAR-END DETAIL OF THE AUSTIN ALL-STEEL STRUCTURE, SHOWING BOX-SECTION MEMBERS

carried out in physical testing-machines on parts such as cross members, door pillars, etc. Useful information is obtained by mounting the complete structure on trestles and loading it progressively in bending (by dead weights) and in torsion (by screw jacks), while recording the deflections which occur at a series of points throughout the length of the job.

Another important side of body development is the elimination of noise, vibration and low-frequency "shakes" which again is a matter of trial-and-error experimentation, aided by a few general principles and a great deal of experience.

Of the various reasons why rigidity is of great importance the following deserve special attention. First of all, the alignment of the various attachments for the suspension and steering systems must be maintained with as little deviation as possible if the car is to behave in a satisfactory manner. Secondly, there is the desire to "tune" the vibrational frequency of the structure, in torsion and bending, to the fairly high figure which is required to avoid resonance with the disturbances to which it is subjected on the road. This means stiffening the job until the frequencies are well above 800 per minute. Thirdly, the body is likely to be quieter on the road if it is rigid; this statement refers to noises excited by the engine, running gear, and road wheels. A fourth point is that a rigid structure, with components effectively tied together by welding, will be relatively free from developing creaks and rattles in service of the kind which are occasioned by "loosening up" of bolted joints and attachments.

Even when all precautions are taken, noise investigation often demands specialized research and is made more difficult by the fact that a hand-built experimental car will not necessarily behave in quite the same manner as the first production models which are constructed from die-pressed panels and components. Noise-insulating materials of various kinds are widely used, particularly for parts such as the floor where large and flat areas are liable to vibrate at audible frequencies. It is also essential that the exhaust system should be separated from the body structure by flexible hangers.

Rubber insulation is becoming more and more extensively used to separate the parts which carry the road wheels from the body on which they are mounted, so interrupting road-excited vibrations which may otherwise produce noise in the car. Thus many makers employ rubber pads in the clamps which connect the springs to the axles, and rubber bushings in the spring eyes are frequently used. Insulation of similar kinds is being fitted to independent front suspensions between the cross member (which carries the assembly) and the frame, and at the pivots of the radius arms.

CHAPTER III

ROAD PERFORMANCE FACTORS

THE term "road performance" as applied to a motor-car should properly cover all aspects of the behaviour of the vehicle. In this chapter, however, the study of performance is limited to the following: acceleration, hill climbing, maximum speed, and fuel consumption. All these are factors which can be measured on the road and can be predicted by calculation.

Other aspects of performance which are of equal importance but are less easily defined include riding comfort, handling, and the degree of noise heard by the occupants. These are considered in later chapters in connexion with the parts of the mechanism most closely concerned—suspension, steering, power unit, transmission, etc. Brake performance is separately described in Chapter VIII, in conjunction with braking systems.

Every car of any consequence has a "character" in its road performance which is not easy to measure or even to describe. It is compounded of the way in which the car responds to the controls, the kind of ride and road holding provided by the springing, the volume and quality of the sound produced by the vehicle and the general manner in which it behaves, rather than the precise results which the car may give in terms of acceleration, speed, and so on. Only long experience and a highly-developed critical sense can enable a driver to estimate character when trying a new model on the road. For this reason great importance attaches to the development work carried out by the road-test staff of an engineering department. The members of this staff should be able to provide constructive suggestions to the designers whose work they supplement.

In short, motor-car design has not reached a stage such that a vehicle can be built "off the drawing board" to give exactly the performance, in the broadest sense, that the engineers intended. Having made this point we can proceed to a closer consideration of those performance factors that can be predicted, with reasonable assurance, from the general specification and design of the car concerned.

Power, Weight, and Gear Ratio. In making comparisons of road performance, or when forecasting the potentialities of a new model, it is usually convenient and permissible to assume that the torque developed by an engine is proportional to the "capacity," i.e. the piston-swept volume. In Chapter IV it is explained that this view implies a uniformity in mean effective pressures which is not actually attained in practice; however, provided that this variant be remembered, capacity provides a good guide to output. As British and continental makers invariably state the bore, stroke and capacity in metric units, it is convenient to deal in terms of litres rather than in cubic feet, or cubic inches. Engine capacity in litres is, then, our first fundamental factor in road performance, and this is usually settled in a new design by the taxation class, the number of cylinders and the stroke/bore ratio which are considered to be desirable.

The second fundamental is the weight of the vehicle. Current practice is not uniform as regards the weight taken for performance tests and calculations, but the writer believes that a fair figure is the kerb weight of the vehicle plus an allowance of 450 lb. to cover a driver and two passengers. As most popular saloon cars may be employed to carry anything from the driver only to a load of five people with luggage, 450 lb. represents an intermediate condition.

The rate at which the total load (car plus occupants) can be set in motion by an engine of given capacity is often

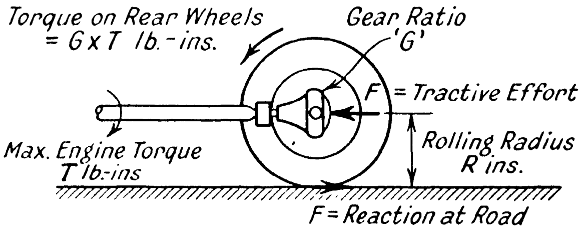

FIG. 25. CONVERSION OF ENGINE TORQUE INTO TRACTIVE EFFORT

considered in terms of power-weight ratio or, more precisely, in lb. per c.c. While this is a handy figure it does not give sufficient information for fair comparisons, because no account is taken of gear ratio or wheel size: in short, of the leverage through which the engine moves the load.

Thus, referring to Fig. 25, the torque exerted by the engine at the flywheel (T lb.-in.) is transmitted to the rear wheels through the final drive, by which it is multiplied in accordance with the top-gear ratio (G). Consequently, neglecting mechanical losses in the transmission, the rear wheels receive a turning effort equal to the product $G \times T$ lb.-in. This statement refers to top gear running, in which case there is no multiplication of torque in the gearbox.

In virtue of the turning effort ($G \times T$) the rear wheels apply a tractive effort (F lb.) which is equal to the reactive "kick" of the tyres on the road surface. These two equal and opposite forces constitute a couple with a leverage equal to the rolling radius of the tyre (R in.). The couple and the applied torque by which it is created must be equal, and therefore—

$$F \times R = G \times T.$$

This can be rewritten in the form—

$$F = GT/R.$$

This expression means that the tractive effort available to move the car is proportional to the gear ratio and is inversely proportional to the rolling radius of the road wheel, i.e. the distance from the hub centre to the road with the loaded tyre inflated to its normal working pressure. The tyre makers provide data for rolling radius; some typical figures for popular tyre sizes are given in Table III.

TABLE III

APPROXIMATE ROLLING RADIUS AND REVOLUTIONS PER MILE, FOR POPULAR TYRE SIZES

(*Note*. Precise figures depend upon the make of tyre in question, number of plies, speed, load, and inflation pressure.)

Nominal Size (inches)	Rolling Radius (inches)	Revolutions per Mile
4·50 × 17	12·7	790
5·00 × 16	12·6	795
5·25 × 16	12·8	780
5·50 × 16	13·0	770
5·75 × 16	13·2	760
6·00 × 16	13·3	750
6·50 × 16	13·6	735
7·00 × 16	14·0	715

As a means for estimating acceleration we can now combine the four factors in a single formula, using the symbols V (piston-swept volume), G (gear ratio), W (kerb weight plus 450 lb., expressed in tons) and R (rolling radius in inches). Then acceleration must depend upon the fraction F/W, which can be rewritten as TG/WR; this simply means that it is proportional to engine torque and gear ratio, but is inversely proportional to weight and wheel size. Or assuming torque (T) to be proportional to capacity (V), we find that acceleration depends upon the ratio VG/WR.

In order to give this expression a more definite meaning we can conveniently state the result in terms of the volume of mixture passing through the engine per mile at full throttle and per ton moved, assuming that the former factor depends solely upon piston-swept volume and crankshaft revolutions and that the volumetric efficiency is 100 per cent (see Chapter IV). In other words, we take as a basis the maximum volume of mixture that the car could conceivably utilize per ton of weight moved and per mile of road covered. Calling this factor L (litres per ton-mile) the resulting formula is—

$$L = 5025\ VG/WR$$

An example will show how this formula is used. Suppose that a 10 h.p. car has an engine capacity of 1140 c.c. (1·14 litres) and a top gear ratio of 5·28 to 1. The kerb weight is assumed to be 2080 lb., to which must be added 450 lb., giving 2530 lb. in all (1·13 tons). If the tyre size is 5·00 in. by 16 in., Table III gives the rolling radius as 12·6 in. Consequently the performance factor required is—

$$L = (5025 \times 1{\cdot}14 \times 5{\cdot}28) \div (1{\cdot}13 \times 12{\cdot}6)$$
$$= 2120 \text{ litres per ton-mile.}$$

This is a fairly representative figure for many 10 h.p. cars. It is increased in larger vehicles up to the maximum of about 3800 litres per ton-mile reached in some American cars with big engines, unrestricted by heavy taxation on horse-power.

The net tractive effort actually available for acceleration or for climbing hills is less than the total effort applied to the car by an amount usually called the rolling resistance. As both the acceleration and the gradient climbable depend upon the ratio between the net tractive effort and the weight of the car, it is convenient to express the effort and the rolling resistance in terms of lb. per ton moved.

The resistance depends upon various factors such as tyre size, inflation pressure, and road surface. It is often taken as 40 lb. per ton for good roads and average tyre equipment. Consequently, if the total tractive effort (in lb.) applied to the car, per ton of weight moved, is denoted by P_1 (lb. per ton), we can say that the surplus effort available for acceleration or hill-climbing is simply $(P_1 - 40)$ lb. per ton. Clearly this surplus effort per ton moved must be directly related to the factor L (litres per ton-mile), but before dealing with this relationship the manner in which both effort and resistance vary with car speed requires some attention.

Velocity and Acceleration Curves. The graph shown in Fig. 26 displays some typical time-velocity curves for the top-gear performance (from 10 m.p.h. upwards) of various car classes. They can be taken as average road-test results for representative models in each class, but in taking this average a few examples that are well below the normal modern standards were omitted.

It will be noticed that from 10 m.p.h. up to 30 m.p.h. each graph is very close to a straight line, so that the acceleration (rate of gain of speed, and therefore proportional to the slope of the time-velocity curve) is approximately constant over this range. Above 30 m.p.h. the acceleration gradually falls away until finally the velocity

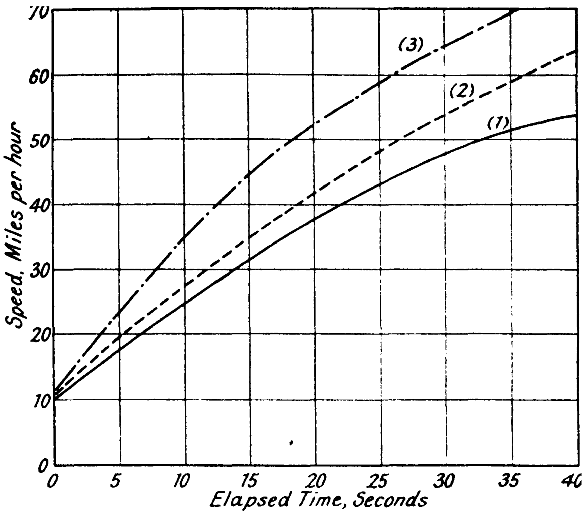

FIG. 26. TYPICAL TIME-VELOCITY CURVES FOR VARIOUS CAR CLASSES: 8–10 H.P. (1); 12–14 H.P. (2); 25–30 H.P. (3)

curve becomes horizontal at a limit representing the maximum speed of which the car is capable. As we shall see later, the falling off is partly due to a progressive reduction in engine torque at the higher speeds and is also occasioned by the rapid increase in air resistance—a factor of relatively little importance below 30 m.p.h.

Time-velocity curves for the indirect gears show similar characteristics, the rate of gain of speed (as compared with top gear) being improved by the lower gear ratios. On the indirect gears, however, the reduction in engine torque at high revs. is the predominating factor and obviously

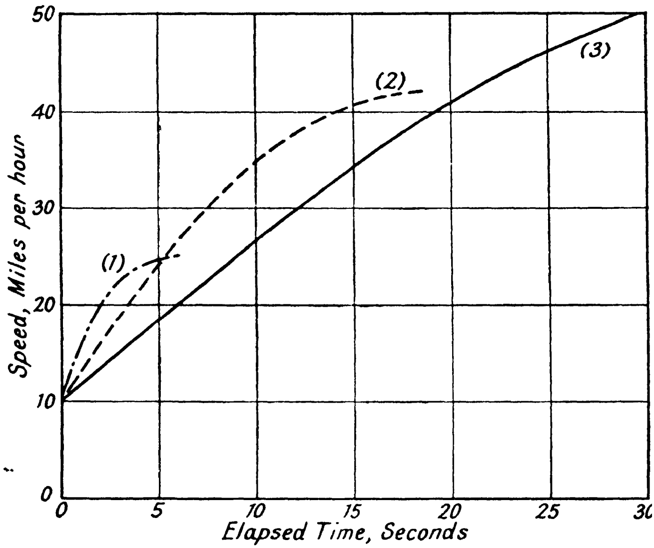

FIG. 27. TIME-VELOCITY CURVES FOR A SMALL CAR ON INDIRECT GEARS (FOUR-SPEED BOX)

sets an early limit to the speed which can be reached. This is made clear by typical time-velocity curves for a small car, on top, third and second speeds, as shown in Fig. 27.

Another way of plotting performance curves is shown in Fig. 28, these being actually based upon road tests of a typical high-powered American car. Here acceleration, in units of m.p.h. gained per second, is plotted against car speed in miles per hour. On top gear the almost horizontal trend of the curve shows how the car holds a steady acceleration from 10 m.p.h. to 30 m.p.h. At higher speeds

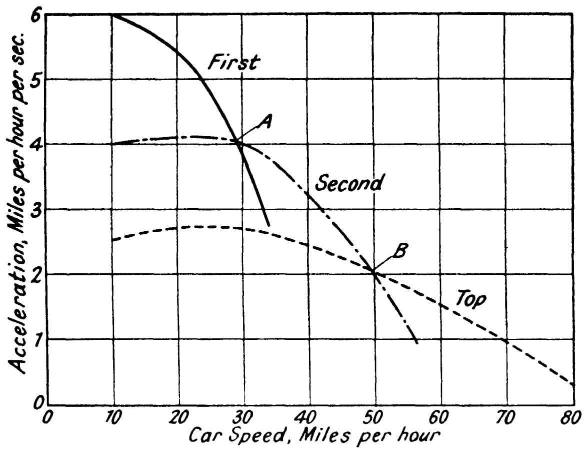

FIG. 28. ACCELERATION CURVES FOR A HIGH-POWERED CAR

the acceleration progressively falls off until it approaches zero at 80 m.p.h.; a speed close to the maximum of which the car is capable.

Similar curves are shown for the first and second gears of the three-speed box, from which it will be seen that the best figures attainable on these gears are 6 m.p.h. per sec. and 4·1 m.p.h. per sec. respectively. Another interesting point is that the intersections marked *A* and *B* show where upward changes of gear should be made to secure the best possible getaway. Thus above point *A*

(29 m.p.h.) the acceleration obtainable by holding first speed falls below that which could be maintained in second speed. Similarly top gear provides a better acceleration than second gear from 50 m.p.h. upwards (point *B*).

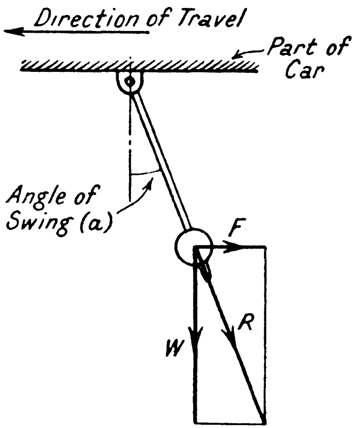

FIG. 29. PRINCIPLE OF THE PENDULUM-TYPE ACCELEROMETER

Although acceleration figures are commonly obtained from tests made with a stop watch and a calibrated speedometer, a very useful adjunct to testing equipment is an accelerometer or "performance meter," such as the well-known Tapley instrument. Such meters work on a pendulum principle, the moving member hanging vertically when the car is running at a steady speed but taking up an inclined position whenever the speed is changing.

The forces which act upon the pendulum are shown in Fig. 29 under conditions of steady acceleration. The bob weight is in equilibrium under the action of its mass (W) and a horizontal force (F) set up by the inertia of the mass and the acceleration of the car conveyed to the pendulum as a whole. The resultant of these forces (R), represented by the diagonal of a rectangle drawn to depict them, will fall in line with the pendulum itself. The ratio F/W will be equal to the ratio between the surplus tractive force (accelerating the car) and the mass of the car—

in other words, it will represent surplus traction in lb. per ton.

It follows that a scale can be placed behind the pendulum on which this figure of lb. per ton could be read directly. Alternatively, the scale could be calibrated to show the

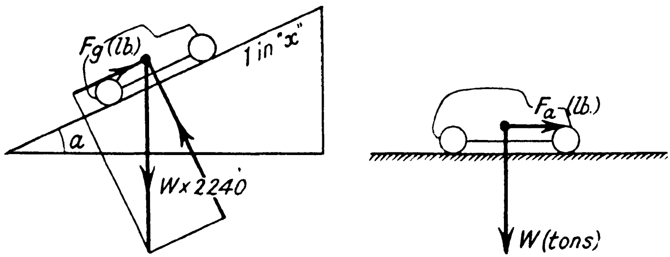

FIG. 30. CONDITIONS FOR A STEADY-SPEED CLIMB (*left*) AND FOR ACCELERATION ON THE LEVEL (*right*)

corresponding acceleration of the car in terms of m.p.h. per sec., or other convenient units.

From the classic formula connecting force, mass and acceleration it is easy to show that if the reading is $P_a$lb. per ton the equivalent acceleration on the level, in m.p.h. per sec., is equal to $P_a/102$. Thus a car which gives a maximum top gear figure of 204 lb. per ton will be capable of gaining speed at a rate of 2 m.p.h. per sec. for as long as this maximum effort can be held; usually, as we have seen, over a speed range of 10–30 m.p.h. Under such conditions the car will obviously require 10 sec. in which to reach 30 m.p.h. on top gear when accelerating from 10 m.p.h. on a level road.

Hill-climbing. The two diagrams in Fig. 30 show a car accelerating on the level and a car climbing a gradient of

1 in x. It is assumed that the engine is exerting maximum torque at full throttle (and on top gear) in each case, and that the slope of the hill is such that it can just be climbed at a steady speed. The gross tractive effort exerted through the rear wheels is not quite the same in each case because there is a slight difference between the torque which an engine will give under steady pulling conditions and the torque which it will provide during a period of acceleration. The latter is always a little smaller than the former, but the percentage difference is not uniform for all types of engine.

This introduces a factor which prevents hill-climbing figures from being absolutely interchangeable with acceleration figures on the level. A second factor is that under the latter condition a certain amount of the available traction is expended upon accelerating the rotating masses—road wheels, transmission, etc.—whereas when climbing a hill at a steady speed the rotational speeds are also uniform.

Referring again to Fig. 30, the surplus tractive force available to accelerate the mass of the car on the level (F_a) and the weight of the car in tons (W) give a ratio F_a/W, which determines the acceleration obtained. This we will denote by the symbol P_a (lb. per ton). Similarly on the gradient the surplus tractive force (F_g) enabling the car to climb at a steady speed can be expressed in lb. per ton (P_g), namely as a ratio F_g/W, which is proportional to the sine of the angle of slope (a). When we say that the gradient is 1 in x we simply mean that $\sin a = 1/x$. Consequently, when the ratio F_g/W is known, the corresponding gradient is easily determined; thus—

$$1/x = F_g/2240W, \text{ and therefore } x = 2240/P_g$$

As an example, suppose that a car will give 200 lb. per ton when climbing steadily in the maximum torque range on top gear. This means that $\sin a = 200/2240$, this being the ratio $F_g/2240W$. It follows that x is the reciprocal of this ratio, namely 11·2, so that the gradient climbable is 1 in 11·2.

In order to establish a practical relationship between level-road acceleration and maximum gradient climbable we now want to know the discrepancy between forces F_a and F_g. Although this varies to some extent between one car and another it can usually be taken that, on top gear, $F_a = 0{\cdot}9\ F_g$, i.e. that there is a discrepancy of 10 per cent. The same relationship will hold between the ratios F_a/W and F_g/W; that is to say, between tractive effort available in lb. per ton, first on the level (P_a) and secondly on the hill (P_g).

A very convenient comparison can be derived from this relationship. We have already seen that maximum acceleration in m.p.h. per sec. is given by $P_a/102$, so that the time (t) in seconds required to reach 30 m.p.h. from 10 m.p.h. will be approximately given by

$$t = 20/(P_a/102) = 2040/P_a.$$

Now we have also seen that the maximum gradient climbable by this car is 1 in x, where $x = 2240/P_g$, and that the lb.-per-ton figure (P_a) for acceleration on the level is about 10 per cent less than the similar figure (P_g) obtained on a hill. Consequently, as an approximation, it is true to say that a car which will accelerate from 10 m.p.h. to 30 m.p.h. on the level, in top gear, in t sec., should be able to make a steady climb up a maximum gradient of 1 in t.

Predicting Performance. Referring back to page 43, we can now consider how to establish the relationship between these ideas of tractive effort, acceleration, etc., and the basic performance factor (L) which can be calculated from engine capacity, gear ratio, car weight and the rolling radius of the tyres.

For any established range of cars of similar design this relationship can easily be determined by plotting maximum surplus tractive effort, in lb. per ton (calculated from road-test results of acceleration or hill-climbing), against the performance factors (L) for the cars in question. Experience shows, however, that so long as we are considering the ordinary run of popular types of car it is possible to establish a wider relationship of the kind shown by a graph in Fig. 31.

The plotted points indicated by crosses were obtained by analysing road-test results for a number of popular cars as published in the motoring journals for the 1939 season; they range from 8 h.p. to 35 h.p. in rating but do not include special high-performance models. The mean line drawn through these points happens to be satisfied by the following simple and convenient formula in which P_g is the surplus (maximum) tractive effort (per ton) on hills and L is the calculated performance factor in litres per ton-mile—

$$P_g + 50 = L/10.$$

To give an example, the factor L for a 10 h.p. car (as calculated on page 43) was 2120 litres per ton-mile, so that, by the above formula, the surplus tractive effort (P_g) should be $212 - 50 = 162$ lb. per ton. From this it follows that the maximum gradient climbable at a steady speed on top gear is 2240/162, or 1 in 13·8. Hence the car would accelerate from 10 m.p.h. to 30 m.p.h. on the level in approximately 13·8 sec. These figures are extremely close to the known road performance of the car in question.

Over a period of years, cars have tended to develop in such a way that the performance factor (L) remains static or, at best, improves very slowly. This is largely because

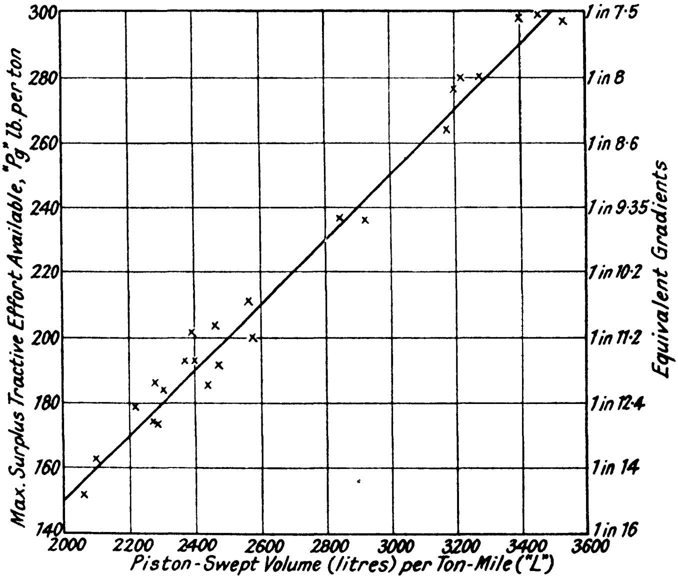

FIG. 31. RELATIONSHIP BETWEEN CALCULATED AND "TESTED" VALUES OF TOP-GEAR PERFORMANCE

of the steady increase in the size and weight of the cars in each horse-power category, which has offset the favourable trends. At the present time, representative values for the factor L in cars of various types are as shown in Table IV.

So far we have only considered top-gear performance, but

TABLE IV

"LITRES PER TON-MILE" (L) REPRESENTATIVE FIGURES

Rated horse-power	8	10	12	14	16 to 24	25 and over
Factor L Min. . .	1800	2000	2200	2300	2300	3000
Factor L Max. . .	2250	2500	2500	2500	3000	3800

the principles can easily be extended to show the surplus tractive effort available on the lower gears. Probably the easiest method is first to make a calculation for top gear in order to obtain the gross tractive effort and the surplus tractive effort for this ratio, each expressed in lb. per ton. Thus in the car already taken as an example these figures were 212 and 162 respectively. Now suppose that this car is operating on a bottom-gear ratio of 20 to 1, instead of the top-gear of 5·28 to 1 on which our calculations were based, the road speed being within the maximum torque range of the engine, as before.

Neglecting additional transmission losses on the indirect gear, the gross tractive effort will increase in inverse proportion to the change of ratio and will therefore be given by—

$$212 \times 20 \div 5{\cdot}28 = 803 \text{ lb. per ton.}$$

From this we have to subtract an allowance for the extra transmission loss which may be as much as 4 per cent, giving a figure of 770 lb. per ton approx. For the resistance we shall be justified in taking an allowance of 50 lb. per ton because, although air resistance will be negligible at low speeds on bottom gear, rough and severe gradients will usually provide a rolling resistance considerably greater than that of a hard main road. Hence the surplus tractive effort available to climb the hill will be 720 lb. per ton, from which the gradient is calculated as 1 in 3·1.

An advantage of this method of calculation is that it lends itself to easy adjustment in respect of any of the factors involved. Thus, if engine capacity, gear ratio, car weight, or tyre size are altered, the effect upon the factor *L* (and, with it, on gross tractive effort in lb. per ton) can at once be determined by simple proportion.

For example, our calculations for a specimen car were based upon a total weight of 2530 lb., which included an allowance of 450 lb. for three passengers. In considering the steepest gradient climbable on bottom gear, however, we might wish to take a more severe condition, such as a load of four passengers plus luggage—say 700 lb.—giving a total weight of 2780 lb. Previously, the gross tractive effort for bottom gear was calculated to be 770 lb. per ton, but it will now be reduced in inverse proportion to the increase of weight and will become $770 \times 2530 \div 2780 = 700$ lb. per ton. After deducting 50 lb. per ton, as before, we find that the maximum gradient climbable with the heavier load is 1 in 3·45.

Performance at Higher Speeds. So far, we have been considering performance over a relatively low speed range such that engine torque increases slowly and maintains an approximately steady "surplus" over resistances to motion. At higher speeds these conditions alter rapidly; engine torque decreases for various reasons and air resistance builds up in proportion to the square of the speed. Consequently it is no longer possible to estimate performance by simple methods, and recourse must be had to the accumulated data on previous models which every manufacturer possesses. For the purpose of this book it will suffice to indicate the general lines upon which these investigations are made.

In passing, it should be noted that although rolling resistance is usually taken to be constant, irrespective of speed, this convenient assumption breaks down at really high rates of travel. Thus normal tyres which display a uniform rolling resistance up to (say) 80 m.p.h., will often produce a sharp increase in drag at higher velocities. This factor becomes important when considering the probable maximum speed of a fast car.

The general formula for air resistance is as follows, F_A being the total resistance from this cause (in lb.), A the frontal area of the vehicle (in sq. ft.), and S the speed (in m.p.h.)—

$$F_A = K A S^2$$

The aerodynamic coefficient K has a value which depends upon the shape of the vehicle but is generally taken as 0·002 for saloon bodies of average modern design. It has been considerably reduced in various special streamlined models. There are many practical difficulties in attempting to establish values of K by full-scale road tests, which is one good reason for basing estimates of the speed of a new car upon the known engine output, shape and speed of its immediate predecessors.

The frontal area is usually taken as the area of the biggest cross-section of the car or might, perhaps, be better defined as the shape which the car would cut out of a taut sheet of paper if driven through it.

Although it is easy enough to evolve a formula for horse-power requirements which includes all the variables, a more simple and direct process is to calculate the tractive effort required at a series of road speeds; from these figures the equivalent torque at each speed can readily be derived.

Probably the clearest way of showing the procedure will be to take an example, working out air resistance and adding rolling resistance through the speed range at 10 m.p.h. intervals, the torque required at the rear wheels being calculated for each speed. The data assumed will be: weight of car with 3-passenger load, 1·25 tons; frontal area, 22 sq. ft.; coefficient K, 0·002; rolling resistance (assumed constant), 40 lb. per ton (and therefore 50 lb. in all); rolling radius of tyres 13 in.; top gear ratio, 4·9 to 1. These figures are not derived from an actual car but are representative of the 12 h.p. class.

Air resistance at 10 m.p.h. is found to be—

$$\text{Resistance} = 0{\cdot}002 \times 22 \times 10^2 = 4{\cdot}4 \text{ lb.}$$

As this factor is proportional to the square of the speed, it is easily found to be 17·6 lb. at 20 m.p.h., 39·6 lb. at 30 m.p.h., and so on. To each of these results is added the constant rolling resistance (50 lb.), and the total is multiplied by the rolling radius of the tyre (13 in.) to obtain rear-wheel torque requirements.

From these figures the final stage is to derive engine (flywheel) torque requirements, knowing the top-gear ratio (4·9 to 1) and allowing 5 per cent for transmission losses. Thus at 30 m.p.h., for example, the rear-wheel torque needed is—

$$(39{\cdot}6 + 50) \times 13 = 1163 \text{ lb.-in.}$$

From this we find—

Engine torque $= (1163 \div 4{\cdot}9) \times (105 \div 100) = 250$ lb. in. approx.

This method of calculation yields the figures shown in Table V and those in the final column have been plotted against car speed to give the "torque required" curve

TABLE V

TORQUE REQUIREMENTS—TYPICAL 12 H.P. CAR

Speed m.p.h.	Tractive Resistance, lb.			Torque, lb.-in.	
	Air	Rolling	Total	Rear-wheel	At Flywheel
10	4·4	50	54·4	707	151
20	17·6	50	67·6	880	189
30	39·6	50	89·6	1163	250
40	70·4	50	120·4	1565	336
50	110	50	160	2080	447
60	158·4	50	208·4	2710	582
70	215·6	50	265·6	3450	740

depicted in Fig. 32. On the same graph the full throttle torque available from a typical engine (of the 12 h.p. class) is also shown plotted against car speed; the derivation of this curve is given in Chapter IV.

It will readily be understood that, at any given speed, the surplus full-throttle torque (available for acceleration or hill-climbing) is represented by the vertical distance between two curves. Over the speed range from 10 m.p.h. to 30 m.p.h. the surplus is fairly uniform, accounting for the steady acceleration available between these speeds. Above 30 m.p.h. it gradually decreases, and at 65 m.p.h. it vanishes as the curves intersect. This is therefore the maximum speed of which the car is capable, if unassisted by wind or gradient.

It is interesting to see how we can work back from these curves to the tractive effort in lb.-per-ton. Thus, over the 10–30 m.p.h. range, a mean figure for the surplus torque is 560 lb.-in. approximately. Allowing for a 5 per cent loss

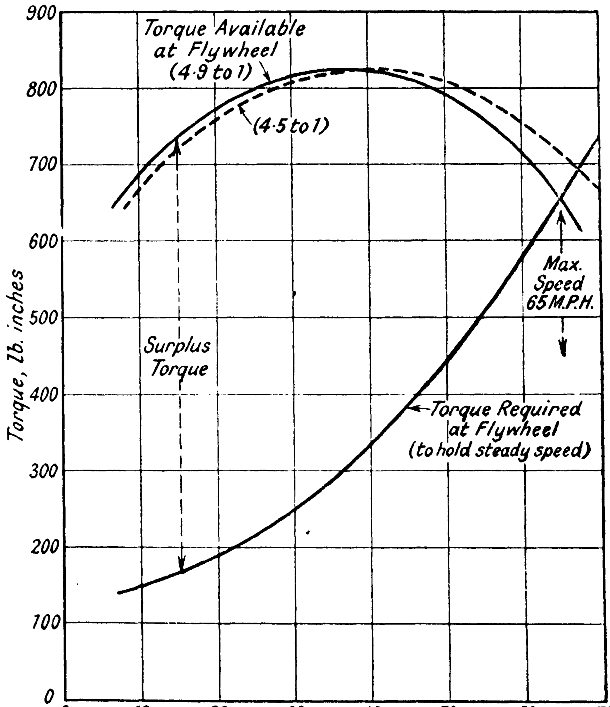

FIG. 32. TORQUE AVAILABLE AND TORQUE REQUIRED, PLOTTED OVER A RANGE OF ROAD SPEEDS

and a 4·9-to-1 gear ratio, this gives a rear-wheel torque of 2600 lb.-in. Dividing this result by the rolling radius (13 in.), we get a total surplus tractive effort of 200 lb., or 160 lb. per ton (for a car weight of 1·25 tons). From this result the gradient climbable and acceleration available can readily be deduced, as already described. At all other speeds the surplus tractive effort will vary in proportion to the surplus torque, as measured between the two curves.

Choice of Gear Ratios. The broken line in Fig. 32 shows how the "torque available" curve will be affected if the top-gear ratio is altered from 4·9-to-1 to 4·5-to-1. Torque in relation to engine speed remains unchanged, but engine speed has altered relationship with car speed. Hence a torque curve plotted on a car speed base is shifted laterally by a change of gear ratio. Thus, in our example, a torque of 790 lb.-in. is available at a speed of 50 m.p.h. on the 4·9 to 1 ratio; on the altered ratio the same point in the torque curve is reached at 54·5 m.p.h. ($50 \times 4{\cdot}9 \div 4{\cdot}5 = 54{\cdot}5$).

In respect of performance the result is clearly shown by the curves: namely, reduced surplus for acceleration at all speeds below 40 m.p.h.; increased surplus in the range above 40 m.p.h.; and maximum speed raised to 67·5 m.p.h.

Conversely, had the ratio been lowered (for example, altered to 5·5 to 1), then maximum speed and "upper end" acceleration would have been sacrificed, while "low end" acceleration would have been improved. The gains and losses involved in varying the top-gear ratio depend to some extent upon the shape of the torque curve but are always broadly of the character described above.

The choice of ratio must also be considered in relation to engine speed and its effect upon durability and fuel consumption; in fact these matters often take precedence over questions of acceleration and speed.

Using the previous nomenclature, the car speed in m.p.h. (S) corresponding to an engine speed (N) in r.p.m., a gear ratio (G) and a rolling radius (R in.), can be found by the formula

$$S = NR/168G.$$

This can be rewritten in the form

$$N = 168\ SG/R.$$

Now at a speed of 60 m.p.h. the car covers one mile per minute, and consequently the engine speed (in r.p.m.) is then equal to the number of crankshaft revolutions per mile. Hence to find "revs. per mile" we have only to substitute 60 for S in the above formula; thus

$$\text{Crankshaft revs. per mile} = 10{,}080\ G/R.$$

This figure is, of course, the same at all car speeds, depending solely upon gear ratio and wheel radius.

In the car previously taken as an example, $G = 4{\cdot}9$ and $R = 13$ in. Hence—

$$\text{Crankshaft revs. per mile} = 10,080 \times 4{\cdot}9 \div 13 = 4070$$

As this figure also represents engine speed, in r.p.m., at 60 m.p.h., it is easy to deduce the engine speed at any other car speed by simple proportion. At 30 m.p.h. it will be 2035 r.p.m.; at 20 m.p.h., 1356 r.p.m., and so on.

Crankshaft revolutions per mile have an obvious effect upon the durability of parts such as bearings, timing gear, etc. It is also true that "engine sensation," as felt within the car, becomes more prominent if the revolutions per mile are raised; for example, by lowering the gear ratio. As explained in Chapter IV, however, piston travel must also be considered, and this depends upon stroke in addition to "revs. per mile." Thus in a long-stroke engine the designer will have to treat crankshaft revs. more conservatively than in a short-stroke engine.

When a suitable top-gear ratio has been determined the indirect ratios have to be chosen. Bottom gear should be low enough to provide for a certain climb on a gradient of 1 in 4, with a full load, preferably with a margin to allow for the fact that few engines are maintained in first-class condition. Current practice suggests that most bottom-gear ratios provide for a theoretical steady-speed climb on a slope of between 1 in 3·2 and 1 in 3·5.

Indirect gears are frequently arranged in a sequence approximating to a geometrical progression. Thus for a small four-speed car they might be 5·3, 8·0, 12·1, and 18·3 to 1, or any ratios close to these figures which could be obtained from convenient gear-teeth combinations. In a three-speed car a series of ratios could be 5·3, 9·0, and 15·3 to 1, but this geometric series would not give a sufficiently low bottom gear and would have to be modified to meet this requirement.

Fuel Consumption. Apart from what might be termed the inherent fuel-burning qualities of the engine, mileage per gallon is affected by a wide range of factors such as car weight and frontal area, speed, rolling resistances and, in addition, by extraneous matters of which the weather, driving methods, traffic conditions and the type of road covered (hilly or flat, winding or straight, hard or soft-surfaced, etc.) are the more important. There are, indeed, so many variables as to make it difficult to measure fuel consumption fairly under operating conditions.

First as regards car speed, the curve of torque requirements shown in Fig. 32 will have given a fair idea of how resistance to motion increases with "m.p.h." and, correspondingly, the fuel expended in propelling the car. Or, viewing it in another way, crankshaft revolutions per mile are always the same on a given gear, but the throttle opening (and, with it, the volume of mixture pumped into the engine and burnt) must increase at the higher speeds. The only compensating factor is the improvement in engine efficiency at the wider throttle openings.

By driving a car over a set distance at a series of steady speeds, and taking average readings of fuel consumed, on runs in opposite directions, a series of specific fuel consumption figures is obtained of the kind shown in Table VI. It is frequently found that the average fuel consumption achieved by owners of small and medium-sized cars, under ordinary running conditions, approximates to the "specific" figure for a speed somewhere between 40 and 45 m.p.h.

TABLE VI

SPECIFIC FUEL CONSUMPTION FIGURES

Steady speed, m.p.h.	Specific Fuel Consumption, miles per gallon				
	Car A	Car B	Car C	Car D	Car E
10	58·0	43·0	—	—	—
20	54·6	39·6	38·5	32·8	25·2
30	50·8	38·0	34·7	30·6	24·9
40	44·1	36·8	30·1	26·9	23·3
50	36·4	31·9	25·7	23·0	19·6
60	27·0	26·8	—	—	—

Note. Under normal driving conditions the cars listed above gave the following average fuel consumption figures, measured over considerable distances: A, 41 m.p.g.; B, 35·8 m.p.g.; C, 28·9 m.p.g.; D, 24·8 m.p.g.; E, 21·8 m.p.g.

Most manufacturers also employ several test routes, covering a variety of road conditions, over which their cars can be driven at two or more selected average speeds. In conjunction with specific figures, such road circuits enable a fair idea to be obtained of the mileage per gallon which the cars will give in ordinary service. At the present time such mileages are, approximately: 40–50 m.p.g. for the 8 h.p. class, 32–42 m.p.g. for the 10 h.p. class, 25–35 m.p.g. for the 12 h.p. class, and 22–30 m.p.g. for the Fourteens.

Car weight affects fuel consumption by determining the energy required for acceleration and climbing hills; it also exerts a lesser influence through rolling resistance. Frontal area and aerodynamic shape largely control air resistance, and therefore fuel consumption, at high cruising speeds.

It is usually considered that a high top-gear ratio is conducive to low fuel consumption, but this is so only within reasonable limits. Thus, if the ratio is so high as seriously to restrict performance on top gear, the driver will make additional use of the lower gears and so will defeat the object of the car designer. An outstanding advantage of low weight in a car is that it permits the use of a conservative top-gear ratio without departing from normal standards of performance. Low weight is therefore both indirectly and directly conducive to low fuel consumption.

CHAPTER IV

THE POWER UNIT

ENGINE design has become such a complex and specialized study that only a review of the bare essentials is possible within the compass of a single chapter. Some of the fundamental relationships between the size of the engine, the weight, gear ratio, etc., of the car which it propels, and the performance which results, were discussed in the preceding chapter. We can now proceed to explain how "size," by which we mean capacity (piston-swept volume), is related to the torque and power output actually obtainable at the flywheel.

Before doing so, however, a brief review of the current trends in power-unit design will be appropriate. Two of these are imposed upon engine designers by the way in which the car as a whole is developing: namely, increased torque per litre of capacity (so reducing the weight and bulk of engine required for a given output), and the ability to operate for long periods without adjustment or other attention.

A study of Chapters I and II will have shown how little space the engine is expected to occupy in a compact modern car; also, the power unit is now "buried" to such an extent (by the front wings and car structure) that it can be reached only from above. Even provision for removing the sump, to reach the big ends and enable the pistons to be withdrawn, has become increasingly difficult as the location of the engine has steadily been moved forwards over the front-wheel centre line. Amongst the expedients developed to meet inaccessible engine conditions we may mention self-adjusting tappets, spring-loaded water pump seals, and components (such as the downdraught carburettor) which have been re-arranged for easy access from above.

Changes made with the aim of increasing output, such as raising the compression pressure and increasing the rotational speed, tend to prejudice durability unless other safeguards are introduced. Hence designers have developed directed flow for water cooling in the head, combustion chambers designed to limit the rate of pressure rise after ignition, better bearing materials, stiffer crankshafts (with balance weights), and so forth.

Even in modern cars the engine is by far the heaviest of any single component except the body. It accounts for about 15 per cent of the kerb weight of the complete vehicle in the majority of cases, ranging up to 20 per cent in automobiles of the American type in which the power unit is of large size and the car is of light build. (See Table VII.)

TABLE VII

REPRESENTATIVE ENGINE WEIGHTS

(In each case the clutch is included with the power unit.)

Rated h.p.	No. of Cylinders	Engine Weight		Kerb wt. of car, lb.	Engine wt. % of kerb wt.
		lb.	lb. per litre		
8	4	216	232	1700	12·7
10	4	330	290	2070	16·0
12	4	320	222	2130	15·0
14	6	450	252	2460	18·3
24	6	495	200	2670	18·5
30	6	590	167	3100	19·0

Engine weights are often analysed in terms of lb. per brake horse-power, but for motor-car work a fairer basis is the ratio between weight and capacity. This varies from 150 lb. per litre in big units to 290 lb. per litre in some diminutive types. Certain factors handicap the small engine, such as the fact that the wall thickness of iron castings cannot be scaled down below a certain minimum, together with the influence of parts such as the dynamo, starter motor, ignition equipment, etc., which cannot be reduced in weight in the same proportion as the reduction in engine capacity.

The Engine Structure. In the majority of power units the cylinder block and crankcase are machined from a single iron casting. By using a deep, ribbed case and full-length water jackets this structure can be made rigid enough to ensure smooth operation, even in lengthy engines such as the straight eight. A one-piece iron casting is also used for V-type engines (including both banks of cylinders and the crankcase) such as the Ford V-8, Lincoln Zephyr V-12, and Cadillac V-16.

It will be appreciated that no matter what degree of balance is obtained by cylinder arrangements, the forces set up by each working unit (piston, connecting-rod, etc.), can only offset one another by virtue of the mechanical interconnexion which the crankshaft and crankcase provide. If these parts are insufficiently rigid, their deflection under load will defeat the inherent balance attempted by the designer and will lead to rough running and accelerated wear.

Present practice in respect of lay-out and proportions is illustrated by Figs. 33 to 36, showing the engines of the 8 h.p. Austin, 8 h.p. Ford, 12 h.p. Vauxhall and 16 h.p. Rover. Of these the Austin and Ford are of the side-valve

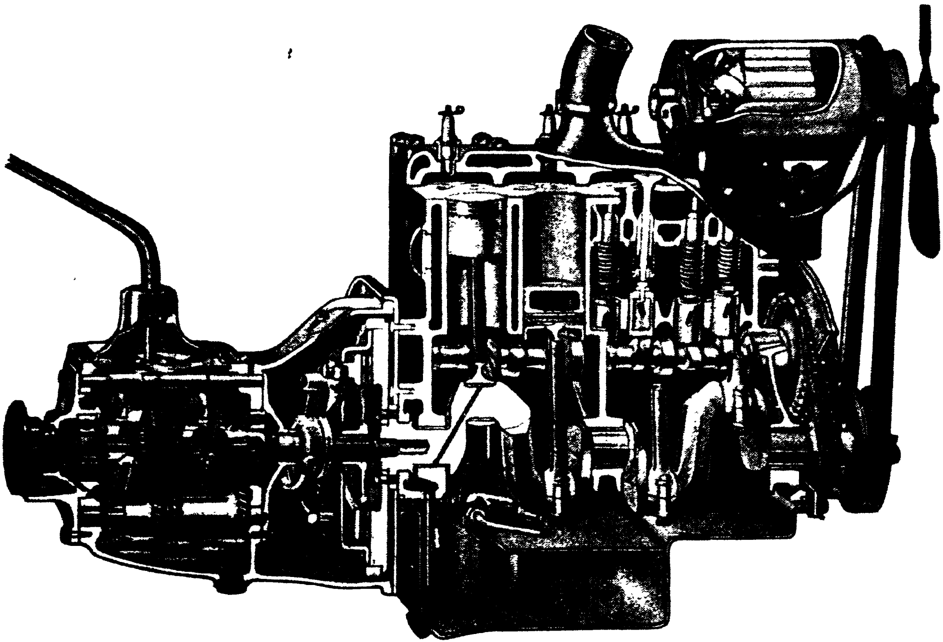

Fig. 33. Sectional Elevation of the Austin Eight Side-valve Engine, showing the Main Constructional Features; Three-bearing Crankshaft, Timing Gear, etc., together with Gearbox and Clutch

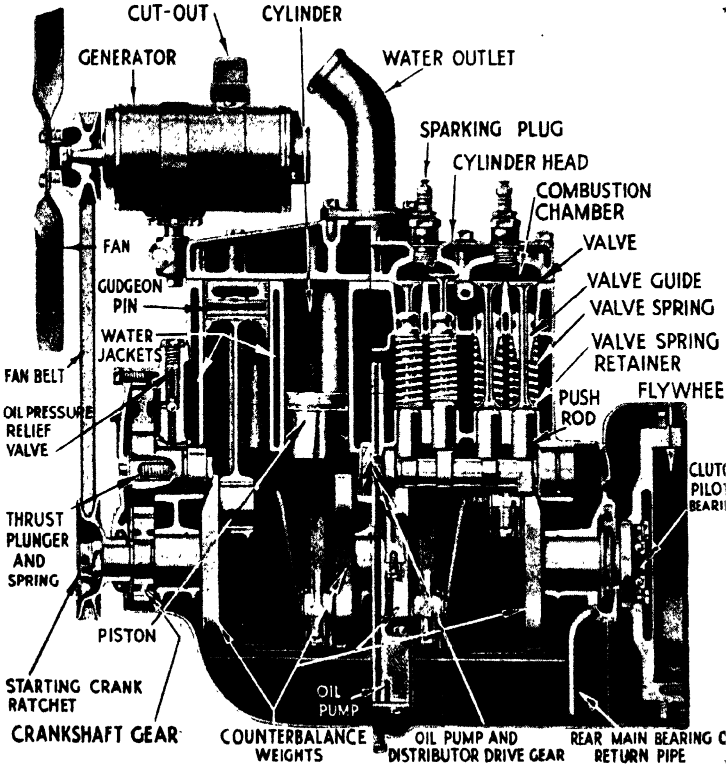

FIG. 34. THE POWER UNIT OF THE FORD EIGHT IS OF THE SIDE-VALVE TYPE AND IS NOTABLE FOR ITS LOW WEIGHT

The non-adjustable valve tappets, designed for long life, are typical of Ford practice. Note the counter-balanced crankshaft

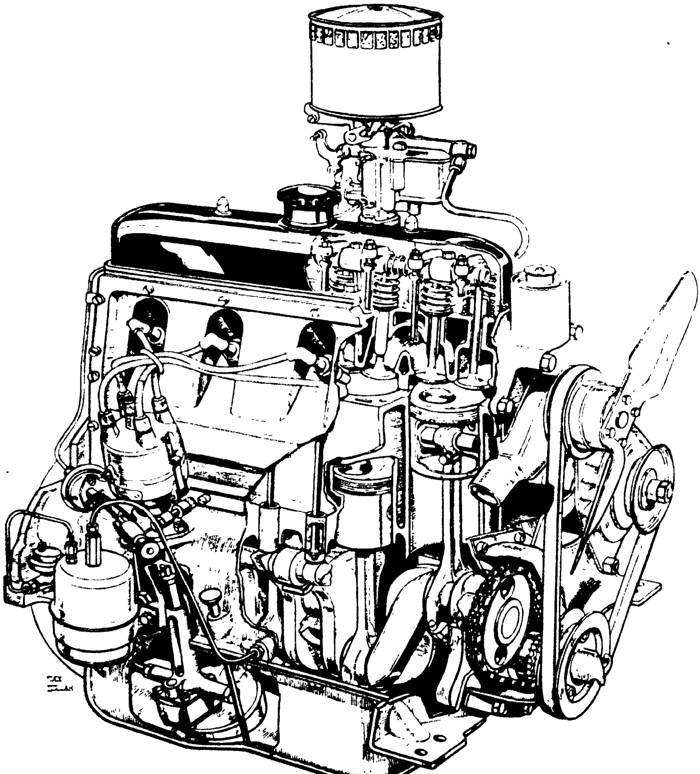

FIG. 35. PERSPECTIVE VIEW SHOWING THE LEADING FEATURES OF THE VAUXHALL TWELVE OVERHEAD-VALVE ENGINE

Specially shaped combustion chambers are obtained by "dimples" in the piston crowns. The crankshaft is counter-balanced and the webs are "finned" for rigidity

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type while the Vauxhall and Rover illustrate overhead-valve practice. These two valve locations continue an active competition which has endured for many years.

In general, the working surfaces of the cylinders are bored and finished in the main block casting. The main

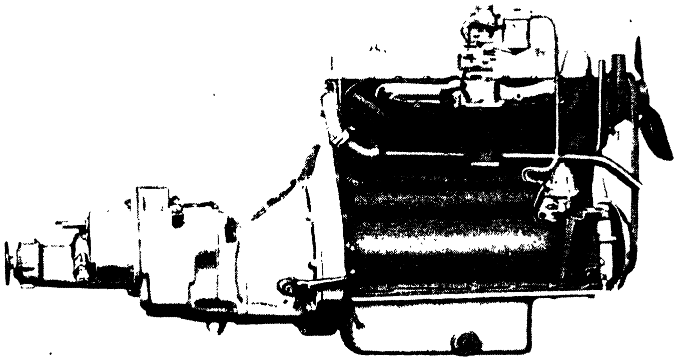

FIG. 36. AN EXAMPLE OF MODERN SIX-CYLINDER DESIGN IN THE MEDIUM-PRICED CAR CLASS: THE POWER UNIT OF THE ROVER SIXTEEN

Note the stepped sump and the use of light alloys for the rocker cover, clutch housing, gearbox and gearbox extension

exceptions are two popular French makes (Citroen and Peugeot) in which wet iron liners are employed. Intermediate practice (Fig. 37) is represented by the Standard Twelve, with a short dry liner in the upper end of each bore.

The cylinder head is also made of cast iron in many units although aluminium alloys are used by a number of important makers of side-valve engines and, in a few cases, are employed for overhead-valve units with inserted valve

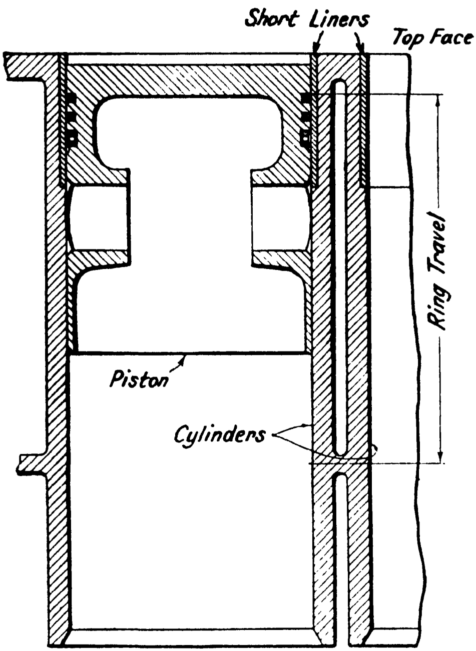

FIG. 37. THE UPPER PART OF EACH CYLINDER OF THE STANDARD TWELVE ENGINE IS COUNTER-BORED TO RECEIVE A SHORT "BRIVADIUM" LINER

This provides a specially durable surface in the location where cylinder wear is at a maximum seatings. Generally speaking, the overhead-valve design has a certain advantage in respect of output per litre, and this margin is reduced by using an aluminium alloy head (permitting a higher compression) in a side-valve job.

At the time of writing, four-cylinder units are almost exclusively used up to a 12 h.p. rating (and up to a capacity of 1·6 litres) in Great Britain, and this class covers about 75 per cent of annual car sales. On the Continent the "four" is in an even stronger position, the dividing line between the four-cylinder and six-cylinder types being in the neighbourhood of 2½ litres capacity. Apart from the effect of taxation, the "four" presents advantages in low fuel consumption and maintenance costs which are important to users of small low-priced cars. On the other hand this type of engine is used only to a small extent in America, where six-cylinder and eight-cylinder engines compete throughout the range of price classes.

Engine mountings have been developed rapidly during the past five years to give the power unit a considerable range of movement vertically, rotationally (about an inclined, longitudinal axis), and in a horizontal plane. In this manner the inertia of the engine mass operates against unbalanced forces and couples which would otherwise disturb the car; the rubber insulation also serves to intercept vibrations of higher frequencies.

The main problems introduced by flexible engine mountings are: (1) longitudinal movement of the power unit during clutch operation, which must be restrained if the clutch is to pick up the torque smoothly; (2) mass movements of the engine on uneven roads which may "tune in" to chassis "shakes" and accentuate their amplitude; (3) loss of the crankcase as a frame cross member, necessitating other means for stiffening the chassis ahead of the dash; (4) the need for designing connexions and controls, between engine and car, in such a way as to be unaffected by movements of the power unit on its supports.

Mean Effective Pressure; Indicator Diagrams. The maximum full-throttle torque which an engine is capable of giving is proportional to the total cylinder capacity and also to the mean effective pressure (m.e.p.) which is exerted on the pistons by the expanding gases. The torque curve previously used as an example (see page 59) is reproduced in Fig. 38, re-plotted against engine r.p.m., and the fact that it is not constant in value is simply due to the way in which the m.e.p. varies over the speed range.

Engine output is also measured in terms of horse-power (h.p.) and as this is a *rate* of doing work it depends upon crankshaft speed as well as torque. Thus, at any given speed (N r.p.m.) knowledge of the torque (T lb.-in.) enables the h.p. to be calculated from the usual formula—

$$\text{h.p.} = 2\pi TN/(33{,}000 \times 12).$$

The power curve shown in Fig. 38 was derived from the torque curve in this way and is typical for a small engine.

These output terms are usually quoted as "brake" torque and "brake" horse-power to show that they refer to useful output, measured at the flywheel, as distinct from the potential "input" to the pistons; the difference is represented by the torque and power used up in driving the engine itself. Similarly, there are two kinds of m.e.p.; one of these, called "indicated," represents the average useful pressure on a piston per working stroke and at full throttle, after deducting the back pressure on the compression stroke and the pumping losses on the exhaust and induction strokes. The second is known as brake m.e.p. and means the "usable" average pressure which remains after a further deduction representing that which is expended upon overcoming friction in the mechanical parts of the engine.

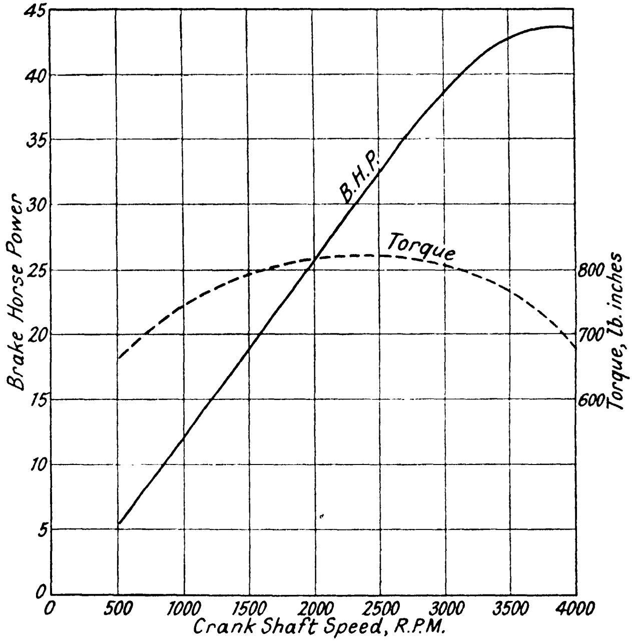

FIG. 38. POWER AND TORQUE CURVES, PLOTTED AGAINST CRANKSHAFT SPEED (IN R.P.M.)

As will be evident a little later, the ratio between b.m.e.p. and i.m.e.p. is the same as between brake torque and indicated torque, or b.h.p. and i.h.p., at any given speed. This ratio is called the mechanical efficiency of the engine, and it varies through the range of speed and throttle openings.

A "classical" style of indicator diagram for full throttle

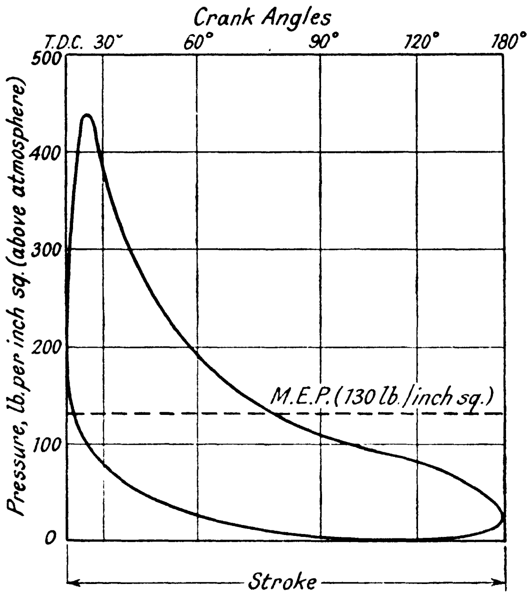

FIG. 39. THE "CLASSICAL" TYPE OF INDICATOR DIAGRAM, DRAWN FOR FULL-THROTTLE CONDITIONS

and a speed within the maximum torque range is shown in Fig. 39. In this diagram the pressure above the piston is plotted against piston travel, for compression and firing strokes. The i.m.e.p. can be calculated from the area of the diagram (representing work done per sq. in. of piston area) and is shown by a broken line at 130 lb. per sq. in. This simply means that the piston would receive the same amount of energy, per working cycle, if it carried a constant pressure of 130 lb. per sq. in. through the firing stroke and idled on the three remaining strokes.

In modern engine testing, indicated m.e.p. is seldom calculated from diagrams of this type. The general practice is to ascertain engine output at the flywheel in terms of the torque curve recorded by an electrical or hydraulic dynamometer. From this the b.m.e.p. can be calculated over a range of speeds. The mechanical efficiency is then ascertained indirectly by measuring the torque required to drive the engine under various conditions. Precautions are taken to ensure that in respect of temperatures, oil viscosity, etc., the engine friction torque "as driven" is closely representative of the friction torque which obtains when it is running under its own power. The "friction m.e.p." derived from these results can then be added to the b.m.e.p. in order to get the i.m.e.p. Thus, for the engine just described the b.m.e.p. might be 110 lb. per sq. in. and the friction m.e.p. 20 lb. per sq. in. This would give an i.m.e.p. of 130 lb. per sq. in. and a mechanical efficiency of 110/130, or 84·6 per cent.

The "cathode ray" type of indicator has largely displaced the mechanical instruments previously used because it is so well adapted to operating at high speeds. A stream of electrons creates a spot of light on a fluorescent screen and can be deflected in either of two planes, set at right angles, by positive charges applied to paired plates. Thus one pair may be supplied with a current controlled in accordance with crankshaft revolutions, while the other is connected electrically to a "pressure element" screwed into the cylinder head. A diaphragm in this element varies the capacity of the circuit.

When the engine is running, the spot of light flashes horizontally across the screen once per crankshaft revolution, at the same time moving vertically in proportion to pressure changes in the cylinder. Hence it can be made to trace out a complete indicator diagram, or any portion of it which requires special study. The rapidity is such that the picture on the screen appears continuous to the eye and can be recorded on tracing paper or, in the case of transitory changes, by a special cinematograph type of camera. A typical record of compression and expansion curves at full throttle, obtained from a Vauxhall Ten engine at 2000 r.p.m., is shown in Fig. 40 and should be compared with the classical diagram of Fig. 39. The instrument will also give a "rate of change" curve, useful when studying combustion in relation to shock and engine roughness.

Cylinder Capacity and Output. In any range of engines of similar characteristics the maximum b.m.e.p. will tend to be uniform, and consequently the probable output of a new model of comparable design can be calculated according to the capacity which it is proposed to use. In the following mathematical treatment the formulæ have not been simplified to the utmost extent because it is more important to illustrate the principles on which they are based than to provide "ready reckoners" of the kind which so easily lead to mistakes.

Throughout we shall be dealing in terms of brake torque, brake horse-power and b.m.e.p., denoted by T lb.-in., "b.h.p." and p lb. per sq. in. respectively. In addition we need the following terms: bore, b in.; stroke, s in.; total capacity of engine, C cu. in.; crankshaft speed, N r.p.m.; number of cylinders, n.

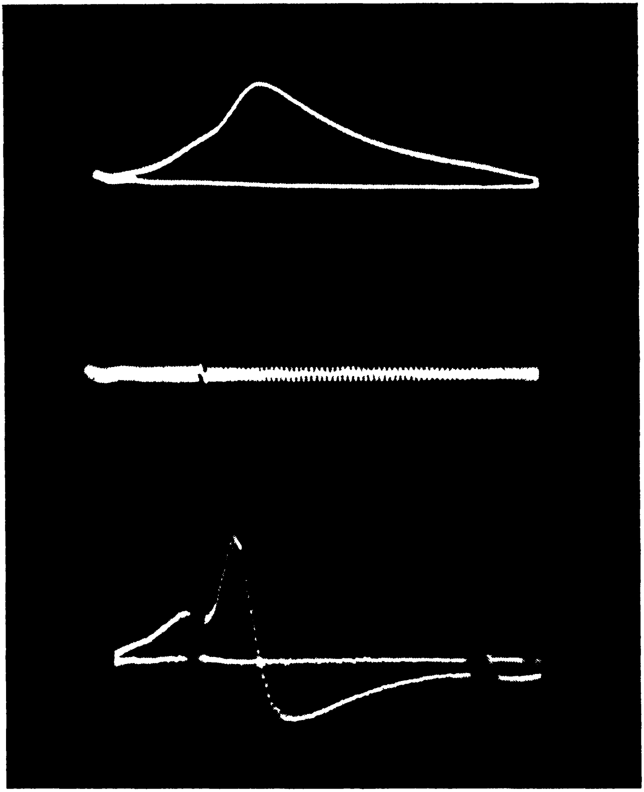

FIG. 40. RECORD OF COMPRESSION AND EXPANSION CURVES (*top*), RATE OF PRESSURE CHANGE (*bottom*), AND TIME BASE (*centre*), OBTAINED BY A CATHODE-RAY ENGINE INDICATOR

The effective load on the piston, in terms of b.m.e.p., multiplied by the length of stroke in feet, will give the work done per working stroke in each cylinder; thus—

$$\text{Work done (ft.-lb.)} = (\pi b^2 p/4) \times (s/12).$$

The number of working strokes per cylinder per minute is simply $N/2$; for the whole engine the number of working strokes per minute will therefore be $Nn/2$. Hence we can readily obtain the work done, in ft.-lb. per minute, and divide this result by 33,000 to derive the b.h.p. of the engine. Thus—

$$\text{b.h.p.} = (\pi b^2 p/4) \times (s/12) \times (Nn/2) \div 33{,}000.$$

This can be rewritten as—

$$\text{b.h.p.} = (n\pi b^2 s/4) \times (Np/24) \div 33{,}000,$$

of which the terms in the first bracket represent the capacity (C) in cubic inches. Hence we get finally the convenient result—

$$\text{b.h.p.} = CNp/(24 \times 33{,}000).$$

As capacity is so frequently expressed in litres it will be useful to set down a similar formula in which V represents piston-swept volume in this metric unit but pressure is still expressed in English units (lb. per sq. in.). Substituting for C the equivalent value ($V \times 1000/16{\cdot}39$), we get—

$$\text{b.h.p.} = VNp/(24 \times 16{\cdot}39 \times 33).$$

Now on page 74 we stated the relationship between power, torque, and speed, so that we can rewrite these equations as follows—

$$2\pi TN/(33{,}000 \times 12) = CNp/(24 \times 33{,}000)$$

and

$$2\pi TN/(33{,}000 \times 12) = VNp/(24 \times 16{\cdot}39 \times 33).$$

Then by cancelling common terms and simplifying in each case, the following results are obtained—

$$T = Cp/4\pi \qquad \text{(capacity, } C \text{ cu. in.)}$$

and

$$T = 1000Vp/(4\pi \times 16{\cdot}39) \qquad \text{(capacity, } V \text{ litres).}$$

Summarized, these formulae show that engine output at the flywheel is subject to the following rules—

(1) Torque is directly proportional to capacity and to brake mean effective pressure.

(2) Power is directly proportional to torque and to revolution speed.

(3) Power is directly proportional to capacity, brake mean effective pressure and revolution speed.

Example. Calculate the brake torque and b.h.p. of a four-cylinder engine at 2000 r.p.m., given that the bore and stroke are 63·5 mm. and 95 mm. respectively, and that the b.m.e.p. is 110 lb. per sq. in. at the speed specified.

First we find the capacity in litres (V), remembering that 10 mm. = 1 cm. and that 1000 c.c. = 1 litre; thus, for four cylinders—

$$V = 4 \times (\pi \times 6{\cdot}35^2/4) \times 9{\cdot}5 \div 1000 = 1{\cdot}203 \text{ litres.}$$

Then from the torque formula we have—

$$T = (1203 \times 110) \div (4\pi \times 16{\cdot}39) = 644{\cdot}4 \text{ lb.-in.,}$$

and from the power formula—

$$\text{b.h.p.} = (1{\cdot}203 \times 2000 \times 110) \div (24 \times 16{\cdot}39 \times 33) = 20{\cdot}4.$$

The second result can be checked from the direct relationship between power, torque, and speed; thus—

$$\text{b.h.p.} = (2\pi \times 644{\cdot}4 \times 2000) \div (33{,}000 \times 12) = 20{\cdot}4.$$

Mean Effective Pressure, and Road Performance. In the previous chapter we established the relationship between engine size, gear ratio, car weight, and the rolling radius of the tyres in a performance factor, L (litres per ton-mile), and showed that this factor gives a fair idea of the acceleration obtainable from cars of all sizes over the medium speed range where torque and b.m.e.p. are at a maximum. It is now possible to correlate this factor with the actual output of the engine.

Using the previous symbols (see pages 41 and 43) the graph established from road-test results showed that the attractive effort at the rear wheels, in lb. per ton (F/W) is approximately given by the fraction $L/10$. We also found that for an engine developing a torque of T lb.-in. this tractive effort could also be expressed as follows—

$$F/W = GT/WR.$$

The next step is to equate these two ways of arriving at tractive effort (in lb. per ton) as follows—

$$L/10 = GT/WR.$$

Substituting for L the value previously ascribed to it, we get—

$$502{\cdot}5\ VG/WR = GT/WR;$$

and therefore—

$$T = 502{\cdot}5\ V.$$

Now in terms of engine capacity and b.m.e.p. we have already found the value of the torque (T) and can use this in the above formula. As a result we get—

$$1000\ Vp/(4\pi \times 16{\cdot}39) = 502{\cdot}5V,$$

and therefore—

$$p = 104{\cdot}5 \text{ lb. per sq. in.}$$

In this calculation we have disregarded the transmission losses, which may amount to 5 per cent (at most) on top gear. Hence we have to adjust the value obtained (for p) by multiplying by 100/95; this gives 110 lb. per sq. in. as the maximum b.m.e.p. of the average engine, as deduced from road test results. Engine test figures suggest that the b.m.e.p. actually varies between 105 and 115 lb. per sq. in. in most modern power units, so that the graph given in Fig. 31 can be taken as truly representative.

Factors Controlling Mean Effective Pressure. Two of the fundamentals which control the mean effective pressure which can be developed in the cylinder are the volumetric efficiency and the compression ratio. The ratio alone is meaningless unless the extent to which the cylinder is being filled is known. This "filling" depends upon the efficiency of the engine as a pump and is affected by restrictions in the path of the mixture on its way from the intake of the carburettor to the cylinder, by valve timing, and by temperature effects.

When the charge is in the cylinder at the end of the intake period it is compressed by the rising piston to an extent expressed as the ratio between the volumes above the piston at bottom dead centre and at top dead centre. Thus if the piston-swept volume is denoted by V and the volume of the combustion space by v, the compression ratio is $(V + v)/v$.

Assuming a uniform volumetric efficiency for engines of a similar type, the compression ratio gives some indication of the anti-detonating qualities of the combustion chamber and the anti-knock characteristics of the fuel. Both these factors have greatly improved during the past ten years, so that the compression ratios employed for popular types of engine have increased from an average figure of 5 to 1 and now approximate to 6·2 to 1, ranging up to 7 to 1 in special cases. It must be remembered that the temperature reached at the end of the compression stroke exerts a big influence upon the tendency to detonation and therefore affects the maximum compression ratio that can be used. Improved cooling, particularly in the neighbourhood of sparking-plug bosses and exhaust-valve seatings, has been used to good effect. Furthermore, cathode-ray indicators and other ingenious equipment have enabled designers to study the effect of combustion-chamber shape and sparking-plug position upon the way in which the mixture burns and, in particular, their effect upon the rate of pressure rise following ignition.

The advantage obtained from successive increases in ratio gradually diminishes while the maximum pressure reached during combustion goes on rising almost in proportion to increases of ratio. In other words, we seem to be already nearing the point where the return (in the form of useful m.e.p.) from further increases of ratio will scarcely be justified by the extra chances of rough running, heavier bearing loads, etc., which the rising maximum pressure will entail.

In multi-cylinder engines it is often difficult to obtain the ultimate b.m.e.p. from every cylinder owing to the way in which the distribution system is apt to starve some of them. Furthermore, there are variations in valve timing and ignition timing between one cylinder and another which can only be avoided by almost uncommercial accuracy in manufacture. Timing is affected by valve clearances, and one of the advantages of automatic tappet adjusters is the uniformity in valve-gear operation which they make possible. Finally, the m.e.p. is dependent upon hitting the optimum mixture ratio in each cylinder, and this presents another difficult problem in carburation and distribution.

As regards brake m.e.p., it is clear that good results depend upon reducing losses to a minimum. Thus a free flow for the incoming mixture and outgoing exhaust gases is needed for minimum pumping losses, while the friction in the engine itself can be reduced by careful design, accurate manufacture and thorough lubrication. The pistons contribute the major part of these friction losses.

Fuel Consumption. Although the engine is not solely responsible for the mileage per gallon which a car will provide, it is nevertheless a major factor in the result. So far as the power unit is concerned, consumption is usually expressed in terms of pints per b.h.p.-hour. This term therefore includes the thermal efficiency with which the fuel is burnt in the cylinders and the mechanical efficiency with which "indicated power" is converted into b.h.p.

Although the internal combustion engine is an efficient type of prime mover there are, nevertheless, formidable heat losses to the water cooling system and to the exhaust. Thermal efficiency is fundamentally dependent upon compression pressure, and this fact provides another incentive for increasing the compression ratio. It also provides a partial explanation for the relatively low efficiency of an engine working under light loads as compared with full-throttle conditions.

It will be obvious that many of the factors already discussed in connexion with b.m.e.p. have an equal bearing upon m.p.g.—such as uniformity of distribution, etc., between a number of cylinders, and the reduction of mechanical losses.

Choice of Revolution Speeds. When it is a matter of securing the utmost power from an engine the crankshaft speed is pushed up to the highest possible figure; thus in racing units it may range up to 8000 r.p.m. In order to get volumetric efficiency at such high speeds the valve timing, inlet passages, etc., must be arranged in a way inimical to satisfactory operation at low speeds such as the ordinary car-driver demands. There is also the fact that problems of balance, and bearing loads due to inertia forces, increase in proportion to the square of the speed, and consequently the racing engine is expensive to construct and to maintain.

The range of speeds covered by the engines of ordinary production cars therefore represent a compromise between the conflicting requirements of power output and durability, the latter being also affected by piston speed. Thus in small engines with relatively short strokes the crankshaft speed can run up to 4500 r.p.m. without exceeding a piston speed of 2500 ft. per min. In large engines 3500 r.p.m. will often represent a reasonable limit.

It must be explained that as the piston accelerates from a standstill to a maximum speed, and then slows to a stop, in the course of every stroke, the term "piston speed" means an average velocity calculated from r.p.m. and stroke length. Thus a 4-in. stroke results in a piston travel of 8 in. per revolution and will give a piston speed of 2400 ft. per min. at a crankshaft speed of 3600 r.p.m.

This factor naturally has an important bearing upon the choice of top-gear ratio in relation to tyre size and engine stroke. The usual plan is to express the relationship in terms of piston travel per mile covered on the road. If G is the top-gear ratio, s the stroke in inches, and R the rolling radius of the tyre (also in inches) then—

$$\text{Piston travel, ft. per mile} = 1680{\cdot}7\ Gs/R.$$

An analysis of the more popular types of car shows that this figure for piston travel varies between 2500 and 2800 ft. per mile and that high and low examples can be found in almost every horse-power class. The figure selected therefore reflects the views and experience of the designer, particularly as regards the maximum travel that can be allowed without risk of undue wear in the cylinders and piston rings. On the other hand, parts such as the bearings, valve gear, etc., will be mainly influenced by engine revolutions per mile, which must obviously be higher in small cars than in larger vehicles. Current practice shows a range of 2800 to 4200 revolutions per mile, approximately, from large cars down to the smallest types.

R.A.C. Rating. The rated horse-power of a motor-car engine is determined by an R.A.C. formula according to the square of the bore (b) and the number of cylinders (n); the same formula is used for the S.A.E. rating in America, although in that country taxation is mainly based on weight. The formula is—

$$\text{R.A.C. rated h.p.} = b^2n/2{\cdot}5.$$

If the bore is expressed in mm. (instead of inches) the divisor becomes 1613 instead of 2·5.

The formula has often been criticized for the fact that it is based upon piston area and number of cylinders instead of the total capacity of the engine. However, the basis is logical in that it assumes a limiting piston speed, irrespective of stroke, and the mean effective pressure which was reached by the engines of thirty years ago. Both these quantities have since increased and to such purpose that it is now quite common for engines of the ordinary production type to develop three times their rated power. It is interesting to notice that in many cases the rated h.p. corresponds closely to the actual b.h.p. developed at 1000 r.p.m.

Apart from having greatly encouraged the production of small cars and engines in England, the formula has also restricted designers in the choice of stroke/bore ratios. Up to a few years ago these ratios varied between 1·35 and 1·5 in most car-producing countries, but latterly, both in America and on the Continent, there has been a strong tendency to equalize stroke and bore; a tendency which British designers cannot follow because of the handicap of taxation. This is unfortunate in that a number of practical advantages can be claimed for what is known as the "square" type of engine, especially for small units.

By avoiding a diminutive bore, more efficient piston rings can be used and the piston obtains a larger area of contact. The short stroke reduces piston travel per car mile and aids the design of a light yet rigid crankshaft. Furthermore, the "square" engine provides more space for bearings and valves and allows the designer to use a more favourable length of connecting-rod. The engine as a whole is slightly longer and wider than one with a long stroke but the same capacity; its height and weight both show considerable savings.

Comparisons can easily be made on the following basis. Let k represent the stroke/bore ratio (s/b); then substitute for stroke the product (bk) in the formula previously given for capacity (C cu. in.). Thus—

$$C = n\pi b^2 s/4 = n\pi b^3 k/4.$$

Hence, having selected a value for k and for the capacity, the bore and stroke can be worked out quite simply. A similar formula can be used for metric units (see page 80).

The following figures illustrate the process and can be re-calculated by the student as an exercise. A typical 10 h.p. engine will have a bore and stroke of 63·5 mm. and

95 mm. respectively, giving a ratio (k) of 1·5 approx. and a capacity of 1·2 litres. If the ratio were altered to unity,

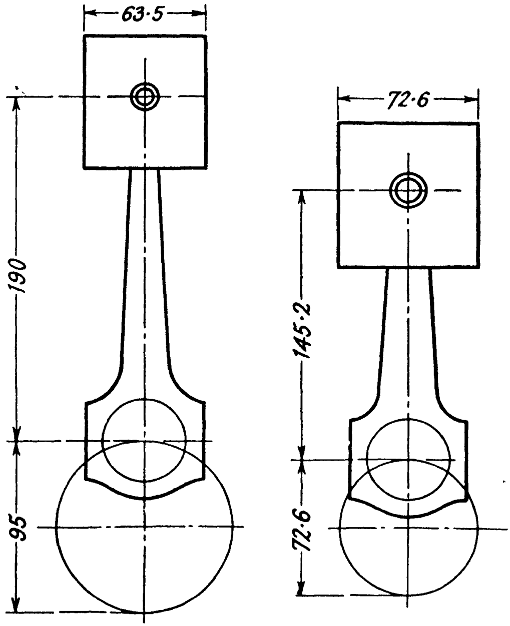

Fig. 41. Comparative Dimensions of Two 1·2-litre 4-cylinder Engines, with Stroke/Bore Ratios of 1·5 (*left*) and 1·0 (*right*)

then for the same capacity the bore and stroke would each become equal to 72·6 mm. This would increase the rated h.p. from 10 h.p. to 13 h.p. and would lengthen the engine by four times the increase in each bore; 36·4 mm., or 1·43 in. approx. Taking the length of the connecting-rod as being equal to twice the stroke in each case (this ratio determines the angular movement of the rod), we should find that the square engine shows a saving of 56 mm. (2·2 in.) in the height above the crankshaft centre line; the depth of the sump would be reduced by 11·2 mm. (0·44 in.). These comparative figures are shown in the diagrams in Fig. 41.

Engine Balance. The rotating masses in an engine can usually be arranged to balance one another "on paper" and will do so in actual fact if the engine is accurately made and if the crankshaft and crankcase are sufficiently rigid to carry the centrifugal forces without serious deformation. All manufacturers balance crankshaft and flywheel assemblies (after machining) by running them at high speed on equipment designed to show where small amounts of metal should be removed to obtain balance.

Disturbing forces of large magnitude are set up by the reciprocating masses, and the crank arrangements used in engines with four, six and eight cylinders in line are devised to secure the best possible balance between the various sets of reciprocating parts. The motion of these parts is best studied by commencing with a slotted link mechanism which gives the effect of a connecting-rod of infinite length. This is shown in Fig. 42.

If the reciprocating mass be denoted by M (lb.) and the crank angle (measured from top dead centre) by θ degrees, then for a crank of radius r in., rotating at an angular velocity of ω radians per second, the vertical inertia force in the line of stroke (f lb.) is given by

$$f = M\omega^2 r \cos\theta/12g.$$

This simply means that at top and bottom dead centres the force is at a maximum and at half-stroke positions it is reduced to zero. At the top and bottom the acceleration

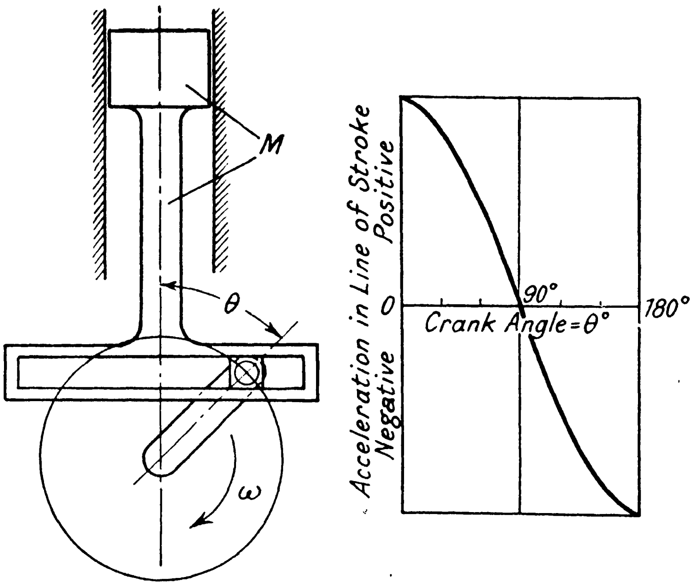

FIG. 42. SLOTTED LINK MECHANISM, EQUIVALENT TO A CONNECTING-ROD OF INFINITE LENGTH, WITH ACCELERATION CURVE

of the mass is at a maximum, where it pulls upwards, and pushes downwards, respectively. At mid-stroke the mass reaches maximum velocity and zero acceleration; the acceleration curve is also given (in Fig. 42) for this simple harmonic motion.

The effect of driving the crank by a connecting-rod, as shown in Fig. 43, is to modify the acceleration curve of the

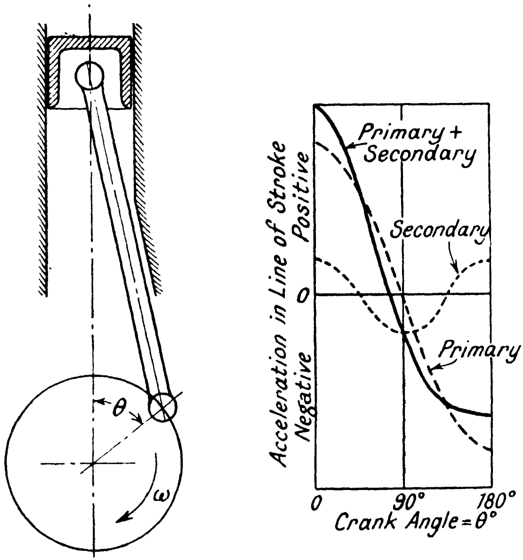

FIG. 43. CRANK AND CONNECTING-ROD MECHANISM, WITH PRIMARY AND SECONDARY CURVES ADDED TO SHOW ACTUAL ACCELERATION CURVE OF PISTON (l/r RATIO OF 4)

piston as indicated by the full line; thus at top dead centre the acceleration (and consequent inertia force) is greater than before, while at bottom dead centre it is smaller than before. Furthermore, the point at which acceleration and force become zero, and velocity reaches a maximum, is now a little above the mid-stroke position. The extent to which conditions are distorted as compared with simple harmonic motion (Fig. 42) depends upon the ratio between the length of the connecting-rod (l) and that of the crank (r), i.e. upon the ratio (l/r), which we can denote by K.

It is convenient to regard the inertia forces in this crank and connecting-rod system as being made up of a number of components of a Fourier series of which the first is simply

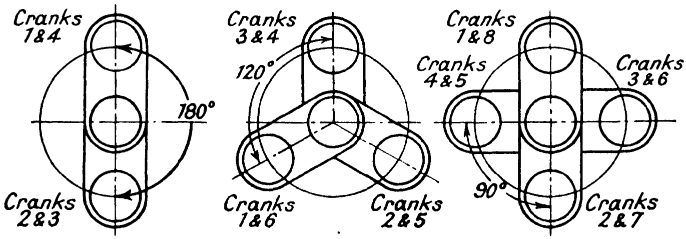

FIG. 44. HOW THE CRANKS ARE ARRANGED IN PAIRS IN ENGINES WITH FOUR, SIX AND EIGHT CYLINDERS IN LINE

that given by the above formula for simple harmonic motion. The second is equivalent to that produced by a mass of $M/4\,K$, again moving with simple harmonic motion but at twice the speed. The other components of the series become progressively smaller and can usually be disregarded. As the magnitude of the secondary force depends upon the equivalent mass and the square of the speed, it will be smaller than that of the primary force in the ratio $1/K$.

This conception is best shown in graphical form, as in Fig. 43, in which primary and secondary forces are indicated by cosine curves for a value of $K = 4$. The inertia force actually exerted in the engine is found by adding the ordinates of these curves, as shown by the full line.

In a four-cylinder engine the primary forces are in balance because the pistons move in opposed pairs (Fig. 44). The secondary forces, because of their double frequency, act always in unison and therefore give rise to an out-of-balance effect, causing the engine to vibrate in a vertical plane with a frequency of twice the speed of the crankshaft. Without analysing the matter it may be pointed out that in six- and eight-cylinder engines with cranks disposed in the normal manner (Fig. 44) both primary and secondary forces are in balance, which accounts for the superior smoothness of such units.

Another source of vibration lies in the connecting rod, one end of which moves in a straight line while the other moves on a circle. The analysis is too complex to give here, but it will be clear that the mass of the rod is subject to accelerations in a transverse plane resulting in inertia forces and in a tendency to bend the rod itself.

The fluctuating torque exerted on the crankshaft results in a precisely similar reactive torque on the complete engine which is produced by the side thrusts of the pistons on the cylinder walls. The variable crankshaft torque tends to set up vibration in the crankshaft and the variable reactive torque rocks the engine on its mountings at low speeds. The ratio between maximum and mean torque values is reduced as the number of cylinders is increased.

Inertia forces produced by the reciprocating parts become great enough to create very heavy bearing loads at high speeds. They are shown graphically in Fig. 45 (curves marked "*I*"), as calculated for a four-cylinder 10 h.p. engine running at 4000 r.p.m., and other curves (*G*) have been added to show the gas loads on the piston during the firing and compression strokes. On the exhaust and inlet strokes gas pressures are too low to require consideration in this connexion.

As all these curves are based upon connecting-rod loading in the line of stroke, through two revolutions of the crankshaft, the ordinates can be added to find the net load; this is shown by the broken line. It will be noticed that a great part of the explosive effort, after firing, is utilized to

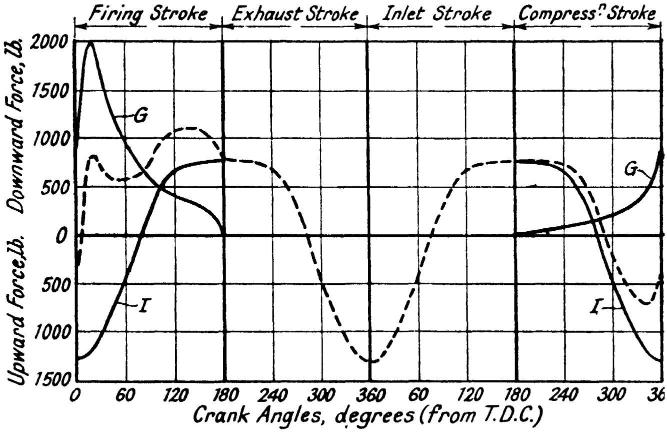

FIG. 45. GAS LOADS (G) AND INERTIA LOADING (I), BOTH IN THE LINE OF STROKE, AS CALCULATED FOR AN ENGINE RUNNING AT 4000 R.P.M.

accelerate the piston, the energy being returned to the crank later in the stroke as the piston is brought to a standstill. The greatest load on the connecting-rod is a tensile one and occurs at top dead centre between the exhaust and inlet strokes. The gas pressures curves are drawn to represent full-throttle conditions.

With forces of this magnitude, increasing in proportion to the square of the speed, it is not surprising that the weight of the reciprocating parts should be reduced to a minimum in modern fast-running units. These parts comprise the piston, rings, gudgeon pin and the upper part of the connecting-rod. The lower part of the rod rotates with the crank and these parts exert a constant radial centrifugal force on the main bearings. Consequently, although the centrifugal forces on the various cranks of orthodox engines are in balance, it is nevertheless desirable to use counter-balance weights in order to reduce bearing loads and the forces which tend to distort the crankshaft. These weights are usually formed integrally in the crankshaft forging.

CHAPTER V

THE TRANSMISSION SYSTEM

As generally employed, the term "transmission system" covers the whole of the mechanism used to convey torque from the engine flywheel to the rear road-wheels: namely the clutch, gearbox, propeller shaft, final drive, differential and axle shafts. It is a broad subject and has become subdivided amongst specialists. In modern practice the clutch and the propeller shaft are almost always made, as components, by concerns independent of the car manufacturer. The gears in the gearbox and back axle are either "bought out" or are made in the car factory on machines designed and set up by specialists in gear-cutting. Consequently we deal, in this chapter, only with the broad principles which are the concern of the chassis engineer, who maintains a watching brief over all these components and sees to their general arrangement and mounting in the car.

First there is the "drive line" to consider, best laid out on a full-scale elevation of the vehicle. The general trend is to lower the line as far as possible, so as to avoid employing too deep a tunnel in the floor of the car. In this part of the work allowance must, of course, be made for the maximum upward movement of the rear axle which is permitted by the suspension system.

General practice is to endeavour to obtain a straight drive line when the car is carrying a normal load (Fig. 46), the universal joints then operating with minimum wear and noise. As the back axle rises and falls above and below this location, the joint angles vary; unless they can be maintained substantially equal (the one cancelling the velocity fluctuations produced by the other), the rear-axle pinion speed will not be constant. In actual fact some variations in pinion speed are usually present and are absorbed by the elasticity of the tyres without over-stressing the drive. However, the smaller the variations the better chance there is of avoiding transmission rumble.

The manner in which the rear axle moves relative to the car is determined by the lay-out of the springs (in the usual

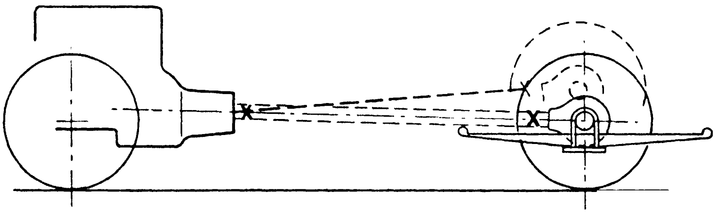

FIG. 46. LAY-OUT FOR A STRAIGHT TRANSMISSION LINE, ALSO SHOWING THE EFFECT OF FULL SPRING DEFLECTION

Hotchkiss drive), or may be more positively controlled by a torque tube or by radius arms. Each half of a semi-elliptic spring of good design will bend in a circular arc. Such springs are commonly flat (zero camber), or nearly so, under normal load. As the spring operates, the axle will trace an arc with a radius which is approximately three-quarters of the free "half-length" (see Fig. 47), a useful approximation when laying-out spring movements. Some designers make a small allowance for rear-axle torque reaction (in Hotchkiss drive), when setting out the transmission line, so as to get minimum joint angles when maximum top-gear torque is being transmitted.

The forward location of the engine which is now the fashion makes it difficult to drop the crankshaft centre-line far enough to suit as low a transmission as is made possible by a hypoid drive. This is because there is so little space available between the crankcase and the front cross member or axle, as the case may be, even when the sump is stepped. The engine and transmission line can be tilted to ease the situation, but then there is increased interference between

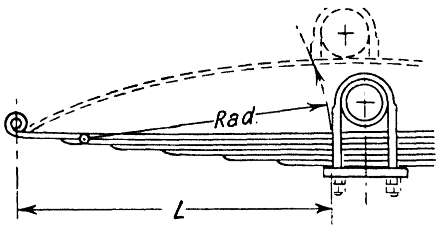

FIG. 47. AXLE MOVEMENT WITH SPRING DEFLECTION CAN BE SET OUT APPROXIMATELY WITH A RADIUS OF $3L/4$

the clutch housing and the floor. Accordingly, most lay-outs show signs of compromise in one respect or another.

Another point to decide in this lay-out stage is the length of the propeller shaft between the universal joint centres. If the shaft is to run quietly it should fall short of its whirling speed, by a comfortable margin, at the highest velocity of which the car is capable. This usually limits the length to about 50 in., although longer shafts are found in certain models. Even in small cars it has become difficult to provide for a shaft of reasonable length owing to the compactness and far-forward location of the engine. In long-wheelbase chassis the "line" may have to be divided by using an intermediate bearing, or an extension may be built on to the gearbox to carry a lengthened mainshaft.

The "safe speed" formula suggested by Messrs. Hardy Spicer is as follows (D being the external diameter of the propeller-shaft tube and d the internal diameter; the length is denoted by L—all measured in inches)—

$$\text{Safe speed (r.p.m.)} = (3{,}750{,}000\sqrt{D^2 + d^2})/L^2.$$

This shows that the safe speed is inversely proportional to the square of the length and (for the usual thin tube) directly proportional to the mean diameter of the cross-section. Denoting this diameter by D_m, we get, by simplification—

$$\text{Safe speed (r.p.m.)} = (5{,}300{,}000\ D_m)/L^2.$$

Example: for a 2½ in. shaft of normal thickness, and a length of 50 in., these formulæ give the safe speed as 5160 r.p.m. The corresponding car speed will, of course, depend upon the final drive ratio and the tyre size.

Propeller-shaft universals are usually fitted with needle-roller bearings, packed with lubricant and sealed for the life of the car. A typical design is shown in Fig. 48. At the forward end (not shown) there is a splined sliding joint to take care of variations in the distance between gearbox and rear axle which occur as the car rides on its springs. These splines must be close-fitting and accurate in order to avoid risk of binding on the one hand or undue play (allowing wobble in the shaft) on the other. There has latterly been a tendency to remove the splined connexion to the mainshaft (within the gearbox), where it is better protected and lubricated. This practice also assists in getting a well-balanced shaft.

The Clutch. In comparison with early designs the modern proprietary clutch is simple, efficient, inexpensive and low in weight. Invariably mounted within a pressed-steel cover, bolted to the flywheel, it comprises the familiar floating disc (faced with rings of friction fabric), presser plate, springs, and withdrawal levers. The splined shaft which carries the

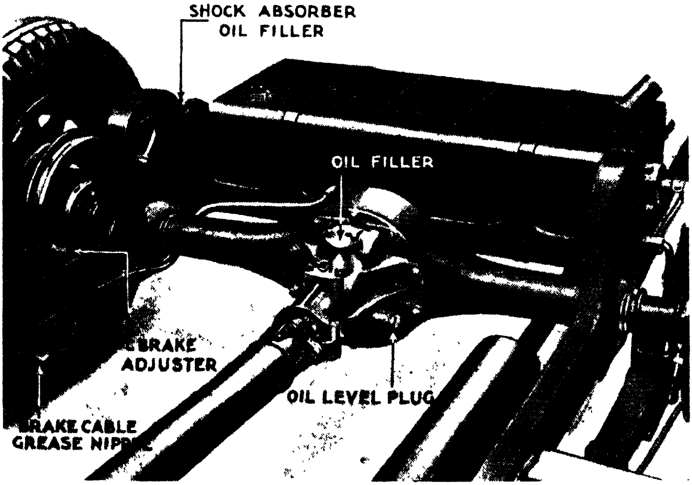

FIG. 48. TYPICAL PROPELLER SHAFT AND FINAL DRIVE LAY-OUT, EXEMPLIFIED BY THE STANDARD FOURTEEN

disc is spigot-mounted in the crankshaft and extends rearwards into the gearbox; it is usually made integral with the primary constant-mesh gear and is recessed to provide a spigot bearing for the mainshaft. Needle rollers are often used for this bearing. A good example of modern practice was illustrated in the preceding chapter (Fig. 33).

The general proportions of the clutch are controlled by the size of the friction surfaces needed to convey maximum engine torque, with a suitable margin to ensure absence of slip after prolonged use. Mr. W. H. Saunders gives the following formula in which T is maximum torque (lb.-in.), P is the total clamping load (lb.), n the number of engaging surfaces (two, in a conventional single-plate clutch), and μ the coefficient of friction (often 0·35) (the radius of gyration of the facings (r in.) can be taken as being the mean of the inside and outside radii of these "rings" with sufficient accuracy in most cases)—

$$T = (Prn\mu)/1{\cdot}4.$$

The constant 1·4 implies a 40 per cent margin of safety above the theoretical "slip" torque. The area of the facings should be such as to result in a pressure of 20 to 30 lb. per sq. in. under the clamping load required.

A high-carbon steel (approximating to a good "saw" steel) is used for the disc, to withstand heat without distortion; under severe traffic use the rise of temperature is very considerable, despite the insulating effect of the facings riveted to the disc. Various expedients are adopted to give a progressive engagement, such as by dividing the disc into crimped sectors or by slotting it to form springy tongues. A good example is the special Borg and Beck disc shown in Fig. 49.

This illustration also shows the damper springs and friction clamp which are located between the disc and hub. Their purpose is to damp out torsional resonances in the transmission line which otherwise produce rattles between the gears or in splined connexions.

Pedal operation is arranged in various ways to avoid interference from engine movement (on flexible mountings), and takes effect either through a special form of ball-bearing or through a thrust pad of carbon and graphite

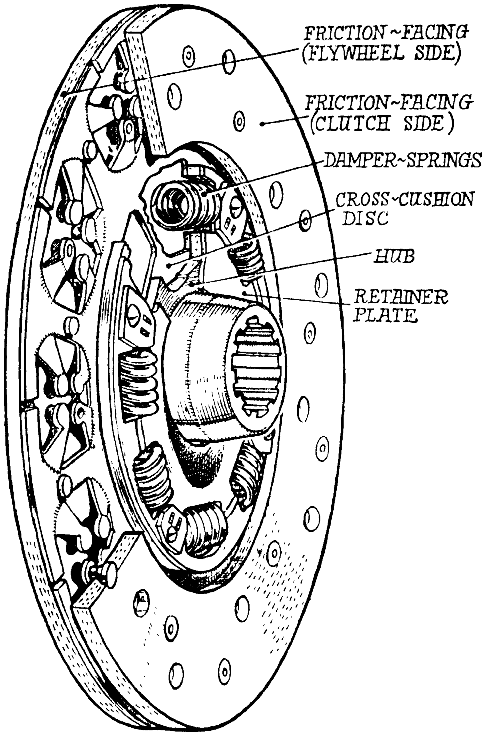

FIG. 49. A BORG AND BECK CLUTCH DISC

This sketch, and that of Fig. 50, are reproduced by kind permission of the Bureau of Information on Nickel compounded (Fig. 50). An adjustment in the pedal gear provides an offset for lining wear.

The Gearbox. The notable decrease in the size and weight of the gearbox needed to cope with a given engine output, which has occurred during the past ten years, provides an illustration of the importance of a rigid mounting to gear life and performances. Shorter casings involve the use of

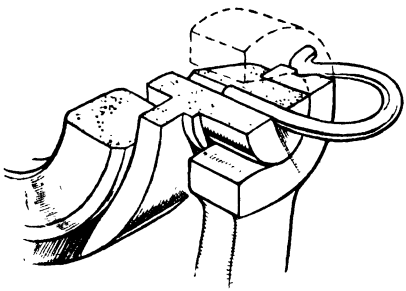

FIG. 50. CLUTCH THRUST PAD, OF CARBON AND GRAPHITE COMPOUNDED

narrower gears but have advantages in respect of rigidity for the support of bearings and a narrow span for the gear-shafts themselves. Hence the available tooth-widths are efficiently utilized, and the relatively narrow gears work as well as the wider units of the larger and lengthier gear-boxes of the past. On the other hand, as regards gear diameters and shaft centres, the smaller the dimensions the greater are the forces (between the teeth) which tend to separate the gears, deflect the shafts, and load the bearings.

The controlling factor remains the tooth-surface loading which can be permitted without running into trouble with undue wear, flaking or pitting of the contact surfaces. When calculating on a "mean torque" basis, the permissible magnitude of this loading depends upon the ratio of maximum to mean engine torque; thus higher loads can, in general, be used for a car with six or more cylinders than in a four-cylinder model. Road performance also enters into the problem, the indirect gears of a small car being used for a much larger proportion of the total running mileage than those of a large car with a favourable "litres per ton-mile" figure on the direct drive (see p. 54).

The "wear" viewpoint is usually more important than tooth-strength calculations because the flexibility and low inertia of a motor-car transmission safeguard the teeth from impulsive overloads. Considerable "winding up" occurs between the engine and the tyre contacts, and this saves the gears from shock, even on sudden clutch engagement. Fatigue fractures may sometimes occur, as a gear tooth is loaded as a cantilever with the maximum stress at the sudden change in section (at the root); hence it is important that a large fillet and smooth surface finish should be provided here.

The compactness of the lay-out also depends upon the skill with which the designer can "pack" the various components. In four-speed gearboxes his task is rendered difficult by the fashion for providing synchromesh engagement for second, third and top gears. A much simpler arrangement results from the three-speed combination with one double-ended synchromesh unit for the top and middle gears.

The choice of gear ratios was mentioned (in connexion with road performance) in Chapter III. Here we are concerned with their effect upon first-gear sizes and therefore upon the maximum depth of the casing. Thus in an American high-performance car the bottom-gear ratio is often of the order of $2\frac{1}{2}$ to 1; in a small British car it may have to be as low as 4 to 1.

Typical lay-outs for two well-known British cars are illustrated. Of these, an Austin four-speed box is shown, with the engine, on page 68 (Fig. 33). A double-ended synchro. slider provides engagement for top and third

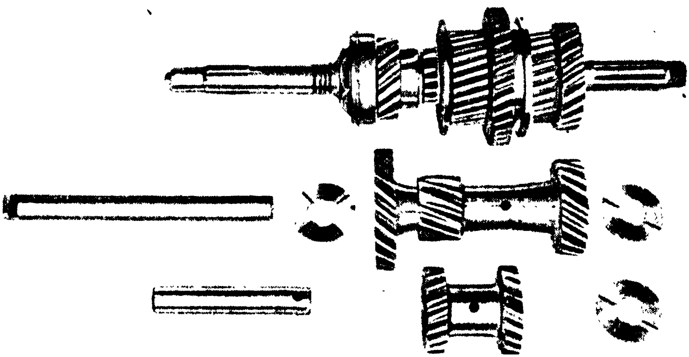

FIG. 51. GEARS AND SHAFTS OF THE VAUXHALL THREE-SPEED GEARBOX (10 H.P. AND 12 H.P. MODELS)

speeds. Another sliding member engages the second-speed mainshaft gear through synchro. cones or, if moved rearwards, provides a sliding-tooth pick-up for first speed. The spiral gear provided for the speedometer drive will be noticed at the rear end of the mainshaft.

In the Vauxhall three-speed design (10 h.p. and 12 h.p. models), a double-ended synchro. sleeve on the mainshaft engages top and second speeds (Fig. 51). A gear mounted on this sleeve gives a sliding-tooth engagement for first and reverse speeds. All the gear teeth, and the splines, are of helical shape.

Another consideration is to reduce the height of the box above the mainshaft to a minimum, so as to avoid a large "hump" in the forward end of a low floor. In conjunction with the recent tendency to shift the gear lever from the box to the steering column, designers have reduced the height of the box by fitting the selector gear at the side instead of at the top.

The shafts of a gearbox, as designed to provide sufficient stiffness, will usually prove adequate to handle the torque to which they are subjected, though the minimum sections should be checked by stress calculations as a safeguard. The type of spline usually employed to provide accurate location for sliding gears results in sharp changes of section from which fatigue cracks may spread unless the stress is reasonable for the material used. For the "layshaft," many designers now employ a cluster of gears turning on a stationary spindle (as in Fig. 51), with plain or needle-roller bearings interposed. Either type of bearing may also be used for those gears, mounted on the mainshaft, which run in constant mesh with the layshaft gears. Relative motion here is never at high velocity (because gear and shaft turn in the same direction, except on reverse drive), and when the gear and bearing are loaded the relative movement ceases. The primary shaft and the tail end of the mainshaft are usually carried in ball bearings.

Synchromesh units can broadly be divided into two classes—those which provide a definite baulking action to prevent premature dog engagement, and a simpler type in which such engagement is restricted only by spring-loaded balls which may be forced aside by a hasty driver. The inner cones are usually made of case-hardened steel, with mating members (of bronze) in which radial slots are cut for the escape of lubricant. Cone angles and surface finish are both critical in their effect upon a smooth and consistent synchronizing action.

The Final Drive and Differential. The design of the conventional spiral-bevel gearing in the back axle has become

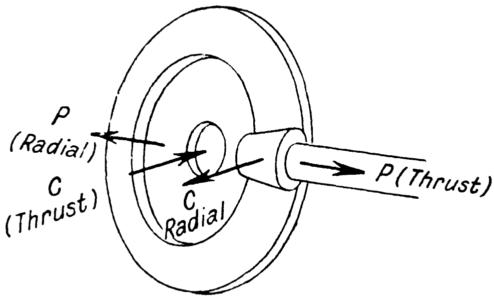

FIG. 52. PINION AND CROWN-WHEEL THRUSTS (P AND C) ARE RESPECTIVELY EQUAL TO THE RADIAL LOAD ON THE OPPOSITE COMPONENT

so specialized that the chassis engineer ordinarily limits his decision to the choice of gear ratio and (possibly) the numbers of teeth needed to obtain that ratio. He is, however, actively concerned with devising a light yet rigid mounting for the gears, and this is a matter of paramount importance to quiet running and durability. As deflections in the mounting, pinion shaft and crown wheel cannot be entirely obviated, the designer also has the task of ensuring that they shall occur in directions least likely to be troublesome. He is responsible for the selection of ball and roller bearings of suitable capacity for the radial and thrust loads involved.

In straight-tooth bevels the pinion and crown wheel, in conveying torque, tend to be forced apart. Hence, in addition to direct radial loading (from torque reaction) the bearings have to carry axial thrust loads. In spiral-bevel gears the teeth have a screw-in or screw-out action, according to the "hand" of the spiral and the direction of rotation. This is superimposed upon the normal bevel thrust. It will be appreciated from Fig. 52 that the end thrust on the pinion (*P*) is equal and opposite to the radial load on the crown wheel bearings; similarly the thrust tending to move the crown wheel laterally (*C*) is equal and opposite to the radial load on the pinion shaft.

These forces can be calculated from knowledge of the maximum bottom-gear (or reverse-gear) torque imposed upon the drive, and the pitch-circle diameters, cone dimensions, face widths, pitch angle, spiral angle, etc., of the gears. The maximum thrust, for a given torque, is the same for a left-hand spiral pinion turning right-handedly (car moving forwards) as for a right-hand spiral pinion turning left-handedly (car in reverse). Minimum thrust occurs under the opposite combination of conditions in each case.

In an overhung pinion the bearings must be reasonably well spaced (in order to reduce radial loading) and this adds to the size and weight of the nose-piece of the axle case. The alternative straddle mounting results in a lighter and more compact design.

Hypoid gears, introduced mainly to drop the level of the drive-line, have been proved to give a very strong, quiet and durable tooth-form and so enable the sizes of the gears and casing to be reduced. A typical example of American practice (the Oldsmobile) is shown in Fig. 53, in which the offset is $1\frac{5}{8}$ in. Bearing loads are higher than in equivalent spiral bevel gears, and the pronounced sliding action of the teeth makes it imperative to use some form of "Excess Pressure" (E.P.) lubricant.

Differential pinion gears may be either two or four in number; the two-pinion lay-out, with single (instead of

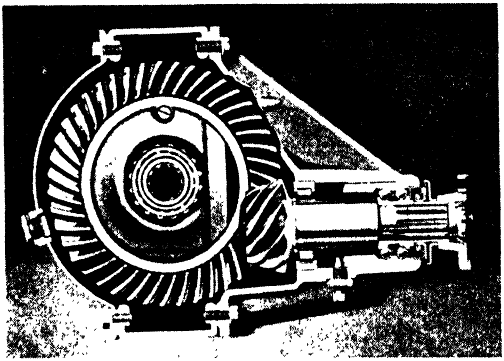

FIG. 53. THE OLDSMOBILE HYPOID-BEVEL FINAL DRIVE, SHOWING GEARS AND BEARINGS

cruciform) spindle, makes it possible to use a one-piece housing for the differential, with saving in weight and cost. For four-pinion differentials the housing is made in halves, bolted together. The side gears with which the pinions engage are frequently formed integral with the drive-shafts. The design of these shafts is separately described (under "Axles") in Chapter VII, where further illustrations of axle gearing will be found. As the differential turns *en masse* during straight-ahead running, and receives very little relative motion on normal main-road curves, satisfactory results are often obtainable from plain steel-to-steel thrust and journal bearings. However, many makers provide thin bronze thrust-washers behind the pinions and side gears to minimize wear and to avoid any risk of "picking up" and scoring.

CHAPTER VI

SUSPENSION AND STEERING SYSTEMS

THAT all parts of a car are to some extent interdependent in their design and functions will have become evident from the earlier chapters of this book. Nowhere is this fact more strongly emphasized, however, than in the behaviour of the suspension and steering systems. Thus, riding characteristics are not merely a matter of designing one spring for the front-end load and another to carry the rear-end load; the sprung and unsprung masses, the position of the centre of gravity, the distribution of the masses, the wheelbase and track, rigidity of the structure, frictional and hydraulic damping, tyre size and design, methods of mounting springs and shock-absorbers—all these are factors which affect the results obtained.

Then the "handling" of the car, by which we mean its behaviour in a horizontal plane under the action of various disturbing and correcting forces, cannot be determined merely by designing a satisfactory steering-gear. This is because handling characteristics are profoundly affected by the suspension system; so much so, that any change which alters riding and rolling qualities will also affect handling in some degree. Particularly is this true of independent suspensions in which the wheels are "mounted off the car" (instead of being paired on an axle) and are therefore tilted when the vehicle rolls.

Having presented this general picture, we can proceed to consider the main factors with which the designer is concerned. But however close the attention which a single component may demand, its connexion with the vehicle as a whole should always be kept in mind.

The Road Springs. Three general types of suspension unit are now in common use: the leaf spring, employing a stack of plates loaded by bending; the coil spring; and the simple torsion bar. Of these the first has the advantage of being able to behave as a structural member as well as a spring; thus in the usual semi-elliptic form it can provide the sole connexion between axle and car, serving to carry tractive

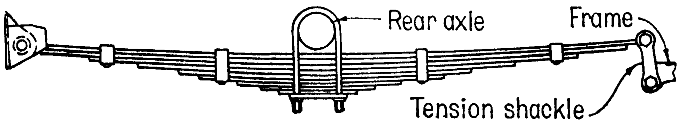

FIG. 54. REAR SEMI-ELLIPTIC SPRING, WITH TENSION SHACKLE (CADILLAC)

and braking forces, and resisting torque reactions. Another inherent advantage is in the damping quality of inter-leaf friction, if properly controlled.

As suspension units, the coil spring and the torsion bar have each the advantage of lightness; for equivalent performance the weight of spring steel required in either of these components is about two-fifths of that needed in an equivalent thin-leaf spring of modern design. But neither of them can function without the assistance of some system of radius arms to carry the road wheel and to convey the load.

As general illustrations for this chapter, various typical applications of the leaf spring, coil spring, and torsion bar are shown in Figs. 54 to 59. These refer to the following cars—

The rear spring used in current Cadillac chassis is shown

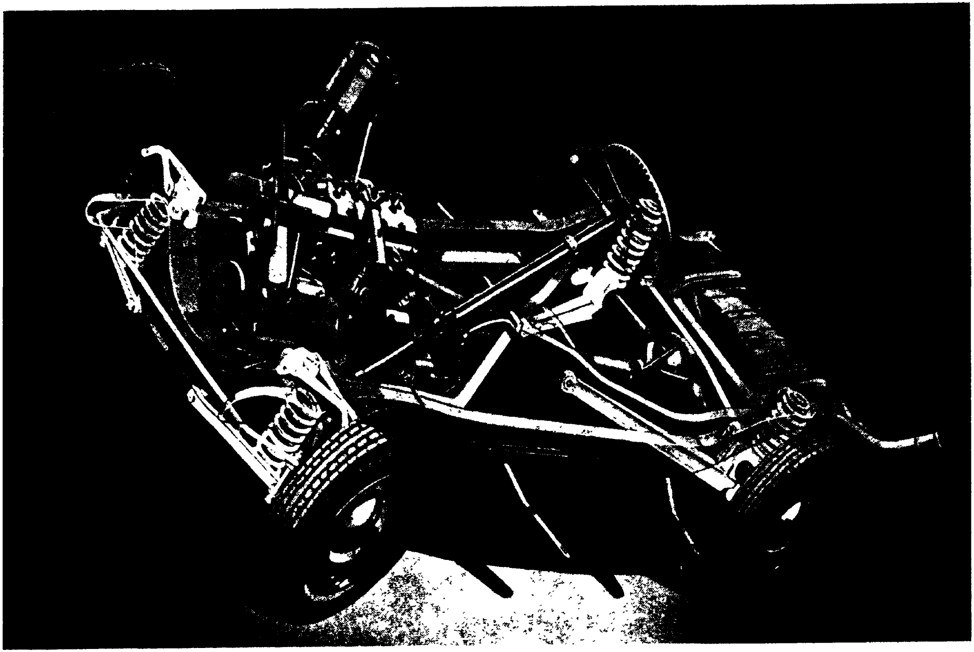

Fig. 55. The Oldsmobile "Series 60" Chassis, showing the Use of Coil Springs at Front and Rear

in Fig. 54. This is slightly cambered under normal load and is fitted with a tension shackle at the rear end; shackle and front eye work on Harris rubber bushings. An interesting contrast is provided by the Oldsmobile lay-out (Fig. 55), utilizing coil springs at the front and rear. The back axle is positioned by radius arms, insulated with rubber at both ends. Each steering head is carried by a pair of transverse

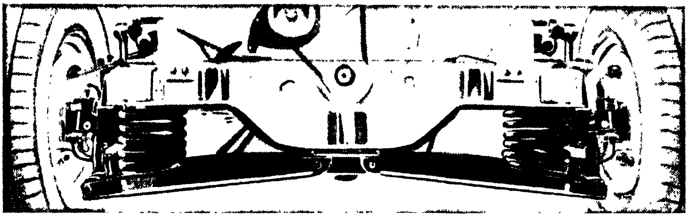

FIG. 56. GIRLING-TYPE INDEPENDENT FRONT SUSPENSION AS USED ON THE 2½-LITRE DAIMLER CHASSIS

links of which the upper units are directly mounted on the shock-absorber spindles.

An alternative coil-spring design, used in the 2½ litre Daimler chassis, is shown in Fig. 56, the lower links being steel pressings. Fig. 57 shows the application of a leaf spring to an independent front suspension (Hillman Fourteen) in conjunction with transverse links. The highly individual independent front suspension used by Vauxhall (Fig. 58) employs a torsion bar and tube to carry the load, the rate being modified by a very stiff coil spring and toggle. Thus a low rate is obtained in the middle range of movement, stiffening towards either extreme as the toggle loading diminishes. A conventional front suspension, but with an interesting application of the torsional type of stabilizer, is used in the

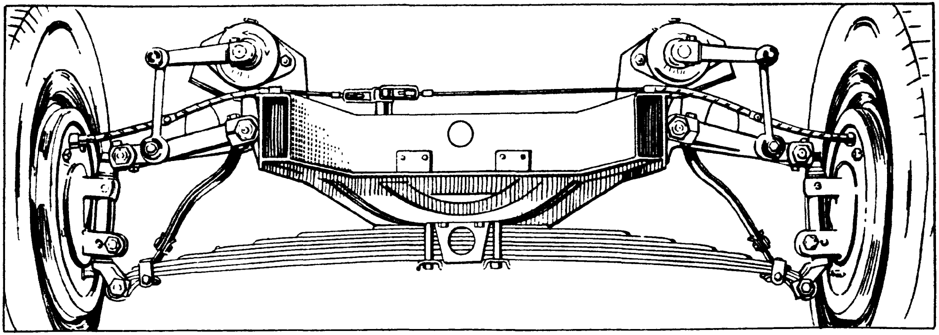

Fig. 57. Independent Front Suspension of the Hillman Fourteen, using a Transverse Leaf Spring and Radius Arms

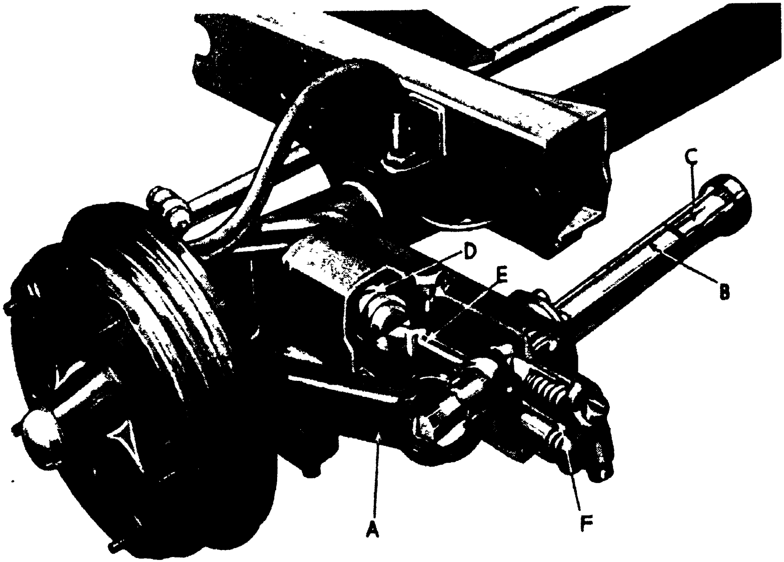

FIG. 58. VAUXHALL TYPE OF INDEPENDENT FRONT SUSPENSION, WITH TORSION BAR AND TUBE, AND TOGGLE SPRING

Morris Ten (Fig. 59). The stabilizer is arranged to control axle "wind-up" under brake-torque reaction.

Spring Deflections and Rates. The riding qualities of a suspension system are often discussed in terms of spring

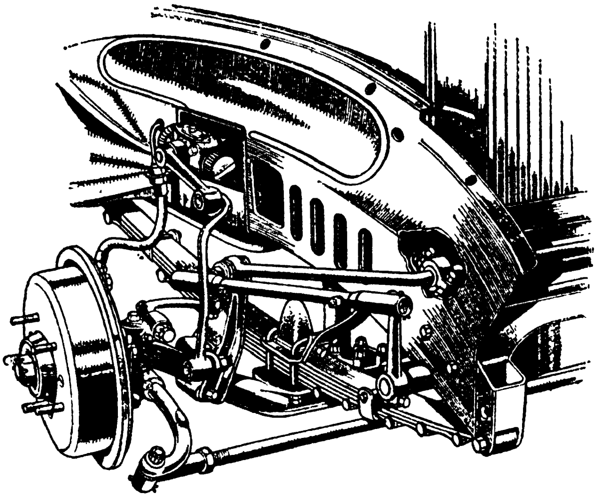

FIG. 59. TORSIONAL STABILIZER, ALSO ACTING AS BRAKE REACTION LINKAGE, IN THE MORRIS TEN FRONT SUSPENSION

(Reproduced by courtesy of "The Motor")

deflections or spring rates, without any very clear idea of the real meaning of these factors. Actually, the two ends of a car should not be discussed individually and without relation to other characteristics of the vehicle, but the reason for this statement can best be grasped by starting with a simple suspension unit carrying a load: such as the coil spring or the simple beam representing a leaf spring (Fig. 60). In each case the load is denoted by W (lb.) and the weight of the spring itself is disregarded.

Suppose that we were to make an experiment by slowly increasing and decreasing the load over a small range of

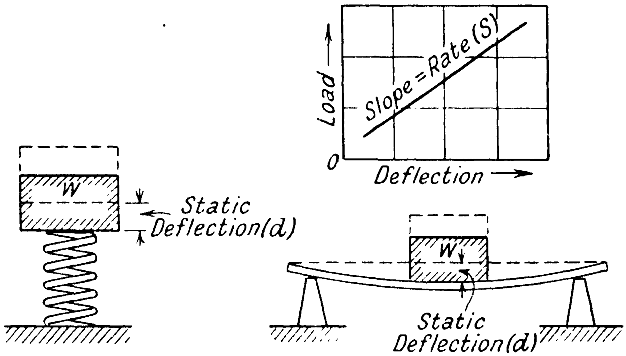

FIG. 60. SIMPLE SPRINGS, AND LOAD-DEFLECTION GRAPH

values, measuring the movement produced in each case. Load and deflection could then be plotted and would result in a straight-line graph (Fig. 60). The slope of the graph is known as the rate of the spring and is often expressed in lb. per inch; we will denote this value by S.

Now suppose that we were to cause the spring to vibrate by the sudden application and removal of an outside force. The system of mass and spring would then vibrate until damped to a standstill by air disturbance and the internal friction in the spring material. The manner of this vibration can be expressed either as the time of one complete oscillation or (more usually) by the frequency, i.e. the number of oscillations per minute. There is a very simple relationship between this frequency (f), the spring rate (S) and the weight (W), for free harmonic vibration, thus—

$$f = 187{\cdot}5\sqrt{S/W}.$$

Now we have already said that the rate (S) is obtained from the ratio between the load and the deflection (d, inches) produced by that load. Hence if the rate happens to be constant from the free position through to the loaded position, it is clear that—

$$S = W/d, \text{ and therefore } d = W/S.$$

Consequently the formula for frequency can be rewritten solely in terms of deflection—

$$f = 187{\cdot}5\sqrt{1/d}.$$

Thus, suppose that we aim at a certain frequency in designing a simple "beam" spring, centrally loaded (Fig. 60)—say 80 cycles per min. Then the required deflection will be given by—

$$d = (187{\cdot}5/f)^2 = (187{\cdot}5/80)^2 = 5{\cdot}3 \text{ in.}$$

In order to calculate the cross-section and length of spring which will give this deflection under a known normal load, the usual beam-deflection formula could be used. The most desirable choice of the two variables (length and cross-section) would depend upon the stress which the material could carry with safety.

An elaborated version of this process underlies all spring design. Working stresses vary with the material, the workmanship and the duty; for good thin-leaf springs they may range up to 95,000 lb. per sq. in. at normal load, or 150,000 lb. per sq. in. at "full bump"; that is to say, with the spring deflected as far as the rubber bump stop will allow.

Variable Rate Effects. Like most simple relationships, the formula for frequency in terms of deflection has a limited application and must be carefully used to avoid misconceptions. The most important limitation lies in the fact that the rate of a spring as actually used on a car shows

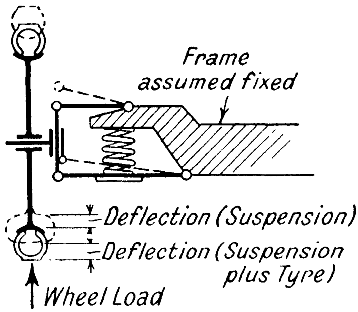

FIG. 61. DIAGRAM EXPLAINING DEFLECTIONS OF SUSPENSION ALONE, AND SUSPENSION PLUS TYRE

very considerable variations through the range of spring movement. The load on the spring also varies according to the number of passengers, etc., that may be carried. Hence the clearest method for comparing suspension units on a frequency or deflection basis is to define the calculated deflection as being the load/rate ratio at some particular load. Thus we might take our "normal load" as the weight supported by the spring concerned with the car carrying three people (450 lb.; see p. 19). Furthermore, our graph of loads plotted against deflections must be concerned with vertical wheel movement relative to the car—not car movement relative to road, because that would embrace tyre deflection as well as suspension movement (Fig. 61).

There are many reasons for rate variations. If a simple spring is tested with the ends resting on trolleys for free movement (Fig. 62A), the changing camber will widen the distance separating the trolleys as the load increases.

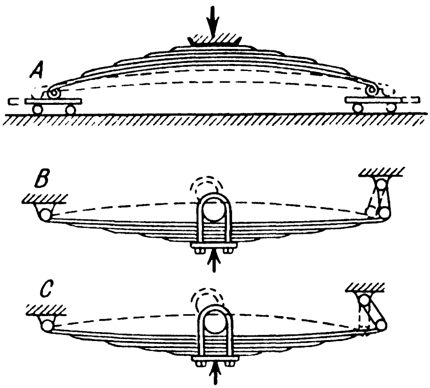

Fig. 62. Three Sketches which show the Influence of Span (A) and Shackle (B and C) upon Rate

Hence the rate of the spring will decrease until it becomes flat; beyond this point the span will shorten and the rate will increase again. In many modern cars the spring is approximately flat at normal load. There are also considerable variations due to the use of shackles, a leaf spring being normally carried by a pivot-pin at one end and a swinging shackle at the other end. Two contrasted diagrams (Fig. 62B and C) serve to indicate how the angularity of the shackle, by throwing a compressive stress into the top leaf, can affect the rate considerably. The vertical shackle (*B*), though good for rate effects, would require a stop to prevent "turning under."

In coil-spring suspensions, variations of rate will occur owing to various factors such as the distortion of the spring and the angles of the links carrying the wheel. Thus in

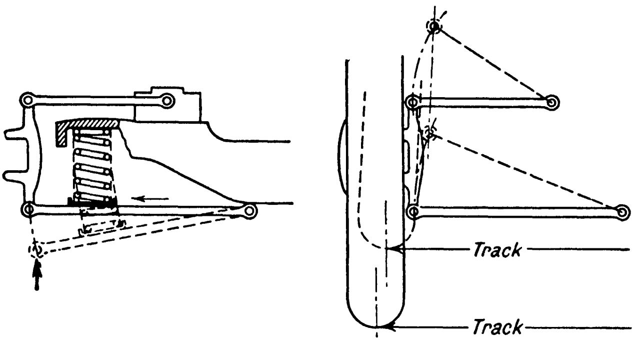

FIG. 63. RATE IS ALSO INFLUENCED BY SPRING DISTORTION (*left*) AND TRACK VARIATIONS (*right*)

Fig. 63 (*left*), the spring is "square" with its abutments only in one position in the range; in all other positions it is subjected to a lateral bending action which affects its performance. And as the wheel rises (relative to the car) the track may vary (Fig. 63, *right*). This alters the leverage exerted by the load on the spring, and so affects suspension rate as measured between wheel and car.

A hypothetical load-deflection curve for a suspension unit is shown in Fig. 64, from the slope of which it will be seen that the rate stiffens up gradually towards full deflection and towards rebound. The rate "at normal load"

(146·6 lb. per in., in this case) can be calculated by drawing tangents to the curves at normal load and then taking the

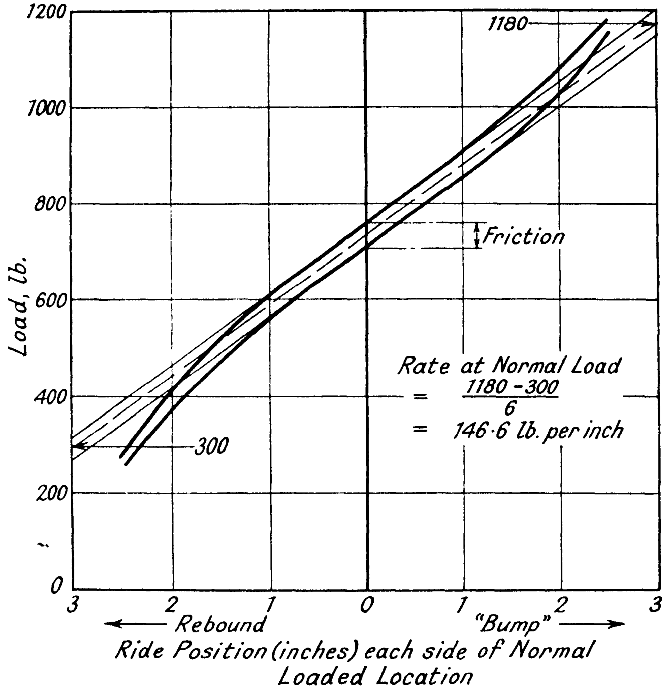

FIG. 64. A GRAPH SHOWING HOW THE RATE AT NORMAL LOAD IS OBTAINED FROM LOAD-DEFLECTION CURVES

mean slope, as indicated. The reason that a loop is shown is that the readings were taken both with increasing load and with decreasing load; the depth of the loop indicates the friction present in the system

The results of applying this method of analysis to the suspension systems of a large number of American cars were given by Mr. Maurice Olley in a paper which he presented before the Institution of Automobile Engineers in 1938. This showed that in the medium-priced range of cars (mainly with independent front suspension) the deflections commonly ranged from 9 in. at the front to 6 in. at the rear. There is no comparable analysis of small British cars, but approximately equal front and rear deflections, ranging from 3½ to 5 in., are probably common.

Effects of Car Size. As human beings desire equal standards of comfort, no matter what size of vehicle they may occupy, the big discrepancy in practice (noted above) may well be questioned. The reasons must be sought in the fact that a small car cannot be simply a scaled-down edition of a big car: (1) because the space required by the occupants does not decrease proportionately; (2) because it travels the same roads and therefore needs the same ground clearance.

Whenever the sprung mass of a car experiences acceleration it tends to "move on its springs" relative to the axles; the chief examples are brake dive (caused by negative acceleration in line of travel), rearing on get-away (due to positive acceleration in line of travel) and rolling on corners (lateral acceleration due to change of direction). Now the forces which disturb the sprung mass in this way act through its centre of gravity. Apart from a few exclusive sports models, the height of this centre shows remarkably little variation in popular cars of all sizes, due mainly to the practical requirements which determine ground clearance and passenger space. An average figure is about 25 in. Yet the wheelbase and track, and the spring spacings that go with them, necessarily become much reduced in the small car. In other words the "base" on which it stands is smaller, but the height at which the disturbing forces are situated remains much the same (Fig. 65).

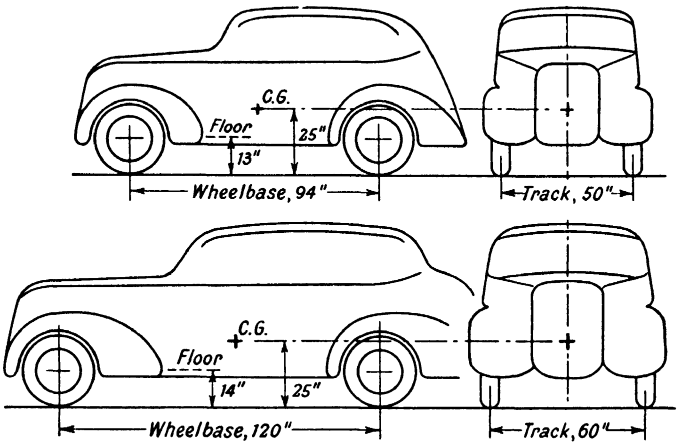

FIG. 65. COMPARATIVE OUTLINE DIAGRAMS OF LARGE AND SMALL CARS, SHOWING SIMILARITY OF C.G. HEIGHTS AND ROAD CLEARANCES

For these reasons the suspension rates permissible on, say, a 10 h.p. car may often be slightly less than those which can be allowed in a car of twice the weight. Hence the spring deflections of the small car (sprung weight divided by rate, at normal load) may have to be only 45 per cent of those used in the big car. These are merely broad figures, but suffice to indicate the general trend. Thus if the deflections were in the ratio 9 : 4 the frequencies would be proportional to $\sqrt{\frac{1}{9}}$ and $\sqrt{\frac{1}{4}}$, or 0·33 : 0·5; the small car will move much more rapidly on its springs than the big car, but with a smaller amplitude.

Tests made by many different research workers on human beings agree broadly in showing that the lower the frequency the greater is the amplitude of the motion that can be called "comfortable" by the majority. From 120 cycles per min. towards low frequencies of 70 or less the tolerable amplitude increases very rapidly, becoming 8 in. or more. In the opposite direction, as the frequency goes up, the amplitude which the "victim" can reasonably withstand falls away until, above 250 cycles per min., a figure as small as $\frac{1}{10}$ in. may be voted uncomfortable.

Pitch and Bounce. Of major importance to the degree of comfort actually experienced on the road is the tendency of the car to pitch, as distinct from "riding flat." During the past five years weight distributions have been moving towards the present approximation to 50–50, fore and aft (car laden), and front-spring rates have been progressively reduced. At the time of writing, front and rear spring deflections are often approximately equal in European cars, whereas formerly, with stiff (high-rate) front springs and low front-end weights, the deflection was often very small at the forward end.

One of the principal advantages secured from a proper use of independent front suspension is that it enables large deflections to be employed without prejudice to stability and handling qualities. The paths of the wheels relative to the car are fully controlled (given rigidity in linkage and the car structure). For a given spring deflection, the roll stability is much increased, in comparison with a conventional axle-leaf-spring suspension, because the "base" is the full track of the wheels instead of the spacing of the springs themselves (Fig. 66).

The actual motion of a car on its springs is a complex combination of pitch and bounce. These have been analysed mathematically by Mr. H. S. Rowell and Professor Guest, to whose published writings the student is referred

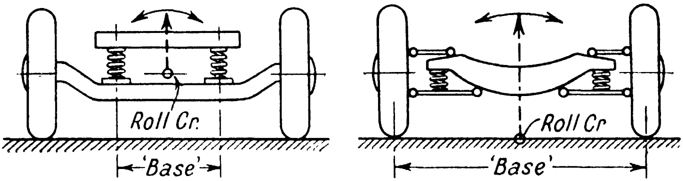

FIG. 66. "BASE" AND ROLL CENTRE: FOR CONVENTIONAL FRONT SUSPENSION (*left*) AND INDEPENDENT FRONT SUSPENSION (*right*)

for detailed information. Here we can give only a brief outline.

Viewing two cars of similar size in elevation (Fig. 67), we will suppose that in one of them the main masses are more widely separated than in the other, but that the centres of gravity fall in the same relationship. Hence wheel weights, spring rates, and deflections could be identical, front and rear. Yet the ride would not be by any means the same in the two vehicles and we need some way of expressing this difference.

Clearly there is a flywheel effect, about a transverse axis, which is greater for car A than for car B. In the diagrams (Fig. 67), the sprung mass of each vehicle is represented by two equal weights (shaded circles) equidistant from the centre of gravity (C.G.) The centre which we ought to take has been demonstrated to be a little below the C.G. of the sprung mass, but it will be simpler to disregard this discrepancy. Then, if the radius of gyration of the masses about the C.G., in the longitudinal vertical plane, be denoted by k, the behaviour of the car is strongly influenced

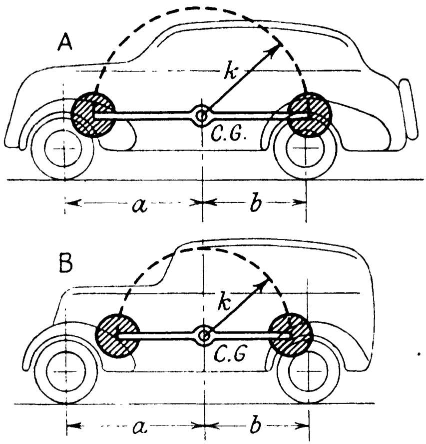

FIG. 67. RADIUS OF GYRATION (k) FOR A CAR WITH WIDELY SPACED MASSES (*top*) AND MORE COMPACT MASSES (*bottom*)

by the ratio k^2/ab. The dimensions (a and b) are the distances separating the front and rear axles (respectively) from the C.G., as shown. From a knowledge of these factors and the spring rates, the centres and frequencies of pitch and bounce motions can be found.

As this ratio (k^2/ab) approaches unity, the increasing flywheel effect will obviously slow down the pitch frequency of the car. Against this advantage, handling may be prejudiced because the complementary flywheel effect (in a horizontal plane) is also increased. So there is a limit to the degree of ride improvement attainable in this way. To quote a familiar example, a large weight of luggage piled up on an overhung rear platform will usually improve ride and make handling more difficult. It is perhaps only fair to add that there is a secondary effect on handling when rear tyres are overloaded, as we shall see later.

In well-sprung modern cars the ratio (k^2/ab) will often approximate to 0·8, and it is interesting to note that, when front and rear spring deflections are equal, or nearly so, the ratio: (pitch frequency)/(bounce frequency) is equal to (k^2/ab). This fairly close agreement of frequencies is a desirable result for comfortable riding, and is more easily attained in small cars than in larger vehicles; a fact which helps to diminish the handicap (previously mentioned) from which the small car suffers.

Damping Devices. In writing of the oscillations of a simple spring (earlier in this chapter), resistances to motion were disregarded. These resistances take two forms in motor-car suspensions, namely (*a*) the fluid damping produced by hydraulic shock-absorbers, and (*b*) the friction produced between various parts of the mechanism such as spring leaves, shackle and pivot pins, radius-arm supports, etc.

Front damping should, in general, be held to the minimum value consistent with reasonable control of suspension movements, and can always be considerably less than rear damping. Modern practice inclines towards designing the front suspension to work with as little friction as may be practicable, most of the control being vested in the shock-absorbers.

Some makers have adopted the same principle in rear suspensions, using coil springs and radius arms in place of leaf springs. The majority continue to employ leaf springs in the "Hotchkiss drive" manner, believing that the inherent friction in these springs has a valuable damping effect upon torsional reactions, "ride" movements and motion of the unsprung masses.

The use of hydraulic shock-absorbers is almost universal, most of these being now of the double-acting type. Moderate movements (in speed and amplitude) cause oil to be pumped through an orifice with a damping effect which increases with velocity. When a certain pressure is reached, a valve opens against a spring to provide a larger passage. This pressure is predetermined by the valve size and spring strength; as a rule it is smaller in the "bump" direction than on the rebound stroke.

When damping is proportional to velocity, and is small in comparison with the disturbing forces, the frequency of the system is very nearly the same as for the free condition. Following an impulse, the vibrations will decrease in amplitude in every successive cycle, in accordance with the law of geometrical progression.

Constant damping, of the kind produced by dry friction, is independent of velocity. Its value for small movements is as great as its restraint upon large and rapid movements, for which reason overmuch dry friction in a suspension will result in harsh riding at low road speeds, especially on bus-rippled or "cobbly" surfaces. It does not affect the frequency, and the reduction in amplitude per cycle (following an impulse) is in arithmetical progression. Furthermore, the system will finally "stick" above or below its normal riding position, according to whether the vehicle is on the downswing, or the upswing when friction checks further motion.

Damping is of great importance as a check upon the violent movements which occur when the frequency of a disturbing force becomes equal to that of the system to which it is applied. This is called a condition of resonance. When the frequency of the disturbing force is low, the forced vibration which it produces will not (as a rule) be serious. But as its frequency approaches that of the natural vibration of a system, the forced vibration becomes violent until, in the absence of any damping, the amplitude will become infinite. At still higher frequencies (of the disturbing force) the amplitude dies away rapidly. There is a rapid change in phasing at the point of resonance: below it, a damped (forced) vibration lags only a little behind the fluctuations of the disturbing force; above it, the phase difference becomes half a cycle, i.e. while the force acts in one direction the system is moving in the opposite direction.

Forced vibrations and resonance are of considerable importance when studying problems such as front-wheel wobble (often effectively checked by steering-head friction), transmission-line vibrations, and the "response" of different parts of the car structure to the disturbances which they receive. But the car as a whole "rides the road" in the manner set by its suspension frequencies and does not experience a ride enforced by road-surface contour. The main function of friction (and of the shock-absorbers) is to damp these ride motions, together with the movements of the unsprung masses which occur at much higher frequencies (500 to 600 cycles per minute).

The advantage of dry friction is that it operates "from the word Go." The build-up of damping with amplitude, which is such a valuable feature of hydraulic shock-absorbers, is likewise a limitation in that they provide no check until the motion is well started; neither can they prevent very small impulses from producing movement. So the two kinds of damping are complementary.

In order to get the best results, hydraulic shock-absorbers must be rigidly mounted and rigidly coupled; otherwise much damping will be lost in the movement allowed by the yield of the mounting and linkage. Various examples will be noted in the illustrations of modern suspension systems on previous pages (Figs. 54 to 59).

Tyre Characteristics. The modern tyre, with its cord construction, has very little inherent frictional damping as compared with the old canvas cover; for the same reason it is more durable, absorbs less energy in rolling resistance, and has a high "enveloping power" over small obstacles.

Suppose that wheel movements are excited by a disturbing force of gradually increasing frequency; for example, by driving over closely-spaced road ripples at a rising speed. For some time (perhaps up to a frequency of 500 cycles per min.) the *wheel* will vibrate with gradually increasing amplitude *between* the tyre and the suspension system, moving with it the unsprung mass of the axle and attachments.

Then a resonant condition is reached and the wheel will begin to hop more and more violently, until the tyre may leave the ground altogether. The frequency of the hop slows down, but the amplitude continues to increase until the hop suddenly ceases at, perhaps, 400 cycles per min. Its persistence or otherwise depends upon the damping available—hydraulic, and plain friction. Damping may also be introduced between tyre and road if the suspension is such as to cause "lateral scrub" when hop occurs.

Like a spring, the inflated tyre has a "rate" expressed as the ratio: load/deflection. Plotted against inflation pressure, the rate for a given tyre size follows a straight-line law. To quote an example, a 5·00 in. by 16 in. tyre on a 3 in. rim, inflated to 28 lb. per sq. in., has a rate of 1145 lb. per in.

The overall "ride" rate (S_0) is a combination of tyre rate (S_T) and suspension rate (S). To visualize this it is only necessary to remember that the reciprocal of "rate" is deflection per lb. of load. Now the total deflection (per lb.) as between car and road is the sum of the tyre and suspension deflections; consequently—

$$\frac{1}{S_0} = \frac{1}{S_T} + \frac{1}{S} = \frac{S + S_T}{S_T S}.$$

$$\text{Hence} \qquad S_0 = \frac{S_T S}{S + S_T}.$$

For example, if the suspension rate is 140 lb. per in. and the tyre rate 1145 lb. per in., then the overall rate is—

$$S_0 = (1145 \times 140) \div (1145 + 140) = 125 \text{ lb. per in.}$$

Tyre Action in "Handling." As the control of a car in the horizontal plane depends upon what is loosely called "tyre adhesion" (on the road), the problems of handling logically commence with a study of tyre behaviour. This has been elucidated by much recent research work.

If a rolling tyre, carrying a load as when on a car wheel, is subjected to a sideways force (F, Fig. 68, *left*), the carcass becomes distorted in its contact with the road and the tyre will commence to follow a path (B) which is no longer in the plane of the wheel (A). The divergence between the lines A and B is called the "slip-angle." Note that this is not produced by any wholesale sliding of the tyre tread on the road; even a small lateral force will produce a divergence and, by moving in this way, the tyre is able to provide an equal and opposite force. So conditions become stabilized at a slip-angle which depends upon lateral force, load, and inflation pressure.

The lateral reactive force acting upon the tyre at the

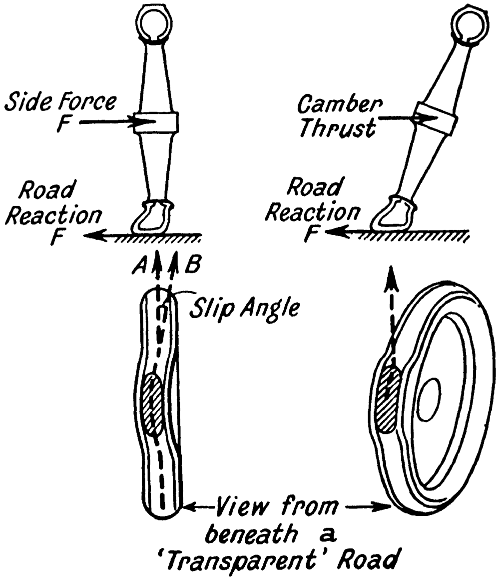

FIG. 68. TYRE RESISTANCE MADE POSSIBLE BY SLIP-ANGLE (*left*); THRUST CAUSED BY CAMBER (*right*)

road surface, when it is rolling at a certain slip-angle, is normally located ahead of the centre of tyre-tread contact and therefore exerts a considerable torque in a direction tending to "straighten" the wheel. This torque provides a powerful restoring effect when the force causing the slip-angle motion is removed, the tyre returning to its true path.

A tyre can also produce lateral thrust if it is on a cambered wheel, i.e. a wheel sloped as in Fig. 68 (*right*). This also produces carcass and tread distortion as the tyre rolls, such that it tends to creep in the direction shown. If prevented, the tyre produces a lateral force (in opposition to the "control") which has been termed "camber thrust."

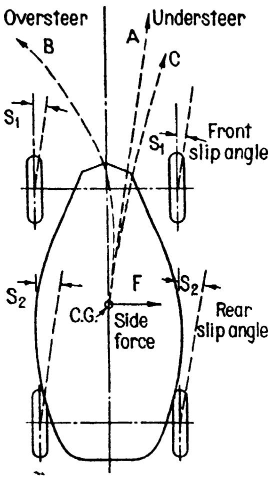

FIG. 69. EXPLAINING THE MEANING OF UNDERSTEER AND OVERSTEER
(See text)

The great importance of these tyre factors lies in the way in which they can be utilized in car design, to obtain the handling characteristics desired. A complete study of this subject would be much too lengthy for inclusion here, the student being referred to papers that have appeared in the journals of the Institution of Automobile Engineers and the Society of Automotive Engineers, particularly those of Mr. Maurice Olley. But the following summary will indicate the general argument.

Directional Stability. The simplest case of handling as affected by tyre slip-angles is that of a car travelling in a straight path (Fig. 69). Suppose that it is acted upon temporarily by a lateral force (F) produced by some disturbing element such as a gusty side wind or a cambered road. In developing lateral resistance, the front and rear tyres drift away from the force along lines which diverge at slip-angles (S_1 and S_2) relative to the wheel planes; the car is initially deflected on to a path such as A.

If the rear slip-angle is greater than the front slip-angle the car will proceed to follow path B, turning towards the lateral force and so producing a centrifugal force which *adds to it*, in effect. This is an unstable condition because the centrifugal force increases the slip-angles and the car turns more and more sharply, unless quickly corrected by the driver. The car is said to "oversteer."

Conversely, if the front slip-angle is greater than the rear slip-angle the car veers away from the disturbing force and follows path C. In so doing it produces a centrifugal force which acts in opposition to the lateral force (F). The car is therefore "self-righting" and is said to "understeer."

As the initial slip-angles produced by the disturbance are independent of speed, whereas centrifugal forces increase in proportion to the square of the speed, it will be clear that the oversteering car becomes more and more dangerous to handle as the velocity increases. Conversely, the understeering car becomes increasingly stable as the speed goes up.

Now in addition to its lateral effect on the car as a whole, the disturbing force will also produce roll. If the car is carried by conventional axles and springs, the effect of roll will depend upon the geometry of the axle attachments and steering-gear, as described later, but the wheels will remain upright on the road.

In a car with independent front suspension, however, the front wheels become cambered when roll occurs (Fig. 68). Camber thrust is then added to the effect of the lateral force, so increasing the front slip-angle. The rear wheels experience no cambering effect and (other things being equal) will have a normal slip-angle. Hence this combination of independent front suspension and a conventional rear axle results in understeer and stability. On the other hand, too pronounced an understeer is not desirable, for which reason the "lean" of the front wheels, which occurs when the car rolls, can be restricted by various tricks in the geometry of the suspension linkage.

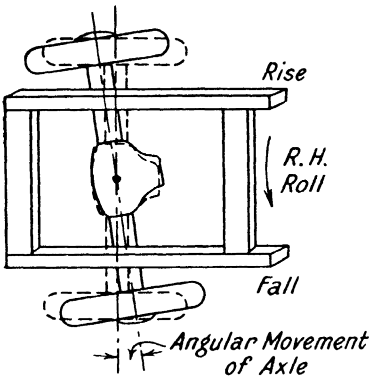

FIG. 70. REAR-AXLE STEERING EFFECT FOR A CAR WHICH IS ROLLING TO THE RIGHT

"Geometrical" Understeer and Oversteer. The connexions between car and axles require careful study to secure a geometry which will produce minimum disturbance when the car "rides or rolls"; or such interference as cannot be avoided should at any rate operate in a safe manner. Two illustrations will explain these points.

In Fig. 70 we have a plan view of a rear axle, supposedly placed under a car which is rolling to the right under a lateral force such as that shown previously (Fig. 69). If the lay-out of the springs is such that their radius-arm effect causes the axle to "skew," the manner in which it moves will determine whether it has an oversteering or understeering effect. Thus, if it takes up the position shown by the full lines, the rear wheels will steer towards the disturbing force and run with reduced slip-angles; this is an understeering condition. An opposite skewing of the axle, when roll occurs, will tend towards oversteer and instability.

A front axle is liable to produce similar effects together with an interference in the steering system. Thus, if a rightward roll causes the axle to skew, and the steering gear is held firmly by the driver, the drag link will swing on the drop-arm ball and will deflect the front wheels to the right or left, according to the position of the link in relation to the front spring. If the deflection is to the right, the front slip-angle will increase and an understeering tendency will be introduced.

The word "tendency" is used advisedly in considering front and rear ends individually, because the final effect on the car as a whole must always depend upon the front and rear slip-angles as a combination.

Independent rear suspensions have been used to a considerable extent on the Continent. Some of them are of the radius-arm type in which the wheels roll with the car. For the same reason that this results in understeer at the front end (by increasing the slip-angles) it will obviously tend towards oversteer if used at the rear end. An alternative independent rear suspension, known as the swing-axle system, is so arranged that the wheels become cambered in a direction opposite to that of the roll and therefore have an understeering effect. Thirdly there is the De Dion axle in which the wheels are connected by a light tube, the drive mechanism being mounted separately. These three rear suspensions are shown diagrammatically in Fig. 71 (centre, top and bottom, respectively).

Apart from wheel-camber effects, slip-angles are strongly influenced by the load which a tyre carries in relation to its inflation pressure. Thus, overloaded (or underinflated) rear tyres will work with abnormal slip-angles

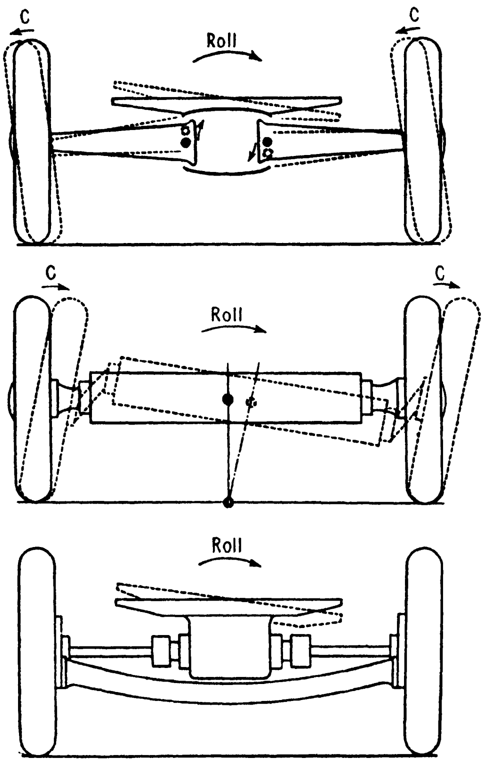

FIG. 71. THREE TYPES OF REAR SUSPENSION: "SWING-AXLE" (*top*); PARALLEL-ACTION (*centre*); DE DION AXLE (*bottom*)

(*Reproduced by kind permission of the "Automobile Engineer"*) and may produce oversteer in a car which normally understeers.

Again, the overturning couple exerted by a lateral force is not necessarily distributed, front and rear, in proportion to the normal static weight distribution. By various expedients—notably the use of a torsional stabilizer—the proportion carried by, say, the front wheels can be greatly increased and that carried by the rear wheels correspondingly reduced. In this way a device primarily introduced as a means for limiting roll (and therefore having a direct effect upon wheel camber in independent front suspensions) is now also employed to correct handling by its effect on roll-couple distribution and slip-angles.

The Steering System. In this chapter on the general principles of suspension and handling, the steering mechanism itself can be only briefly mentioned. Proprietary steering gearboxes of several well-known types are widely used and these mechanisms, and the linkages through which they operate the front wheels, will be familiar to most readers.

From what has already been stated regarding straight-line travel the fact that a car fails to follow the Ackerman-predicted curves, when steered, will be readily appreciated. If it is an "understeerer" it will desire to run straight and will require excess steering movement to make it follow a curve. An oversteerer, on the contrary, will respond quickly but will tend to follow a tightening spiral unless the driver "lets up" at the steering wheel.

Referring to Fig. 72, the Ackerman curve, which is followed only when lateral forces are very small, is determined by the wheelbase (l) and the average front steering angle (A). Thus the mean radius (R) is such that $l/R = \tan A$. This relationship is useful when laying out the steering-gear to give a certain minimum turning circle. At the speeds and radii normally used on the road, the lateral centrifugal force is seldom greater than ¼ the weight of the car. The slip-angles then shift the car away from its

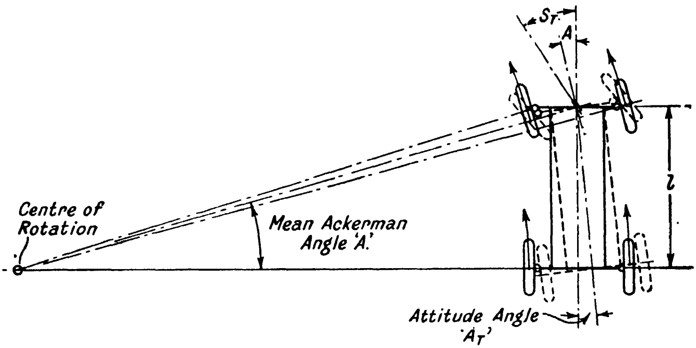

FIG. 72. ACKERMAN POSITION OF A CAR TRAVELLING ON A CIRCULAR PATH (*full lines*) AND POSITION ACTUALLY TAKEN UP (*broken lines*)

Ackerman position on the curve in a manner dependent upon the understeering and oversteering qualities already discussed, as shown (exaggerated) by the broken lines in Fig. 72. The limiting lateral force above which the tyres "let go" and slide, depends upon many factors and is often about 45 per cent of the car weight; occasionally ratios as high as 60 per cent have been recorded.

If a car is being driven around a circle at a set speed, the attitude angle (A_T, Fig. 72) will be equal to the rear slip-angle (S_2), disregarding any angular movement of the back axle under the car. If this attitude angle were zero, the mean front slip-angle (S_1) would be the difference between the mean steering-angle of the front wheels relative to the car (S_T) and the Ackerman angle (A). But as the car as a whole swings through the attitude angle, this is added to the slip-angle at which the front wheels operate. Hence—

$$S_1 = (S_T - A) + A_T$$

Table VIII shows a series of direct experimental readings (for S_T, S_2 and roll) on an understeering car, travelling at various speeds around a set circle as indicated by the corresponding lateral accelerations in the first column. The Ackerman angle is calculated from radius and wheelbase, after which the front slip-angle (S_1) can be derived from the above formula.

TABLE VIII

CAR FOLLOWING A SET CIRCLE AT VARIOUS SPEEDS

(Ackerman angle $4\frac{3}{4}$ degrees approx. (A).)

Lateral Accelera-tion, per cent "g"	Roll Angle, Degrees	Angles, in Degrees			
		Steering, S_T	Difference, $S_T - A$	Rear Slip, S_2	Front Slip, S_1
10	1	$5\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{2}$
20	$2\frac{1}{4}$	6	$1\frac{1}{4}$	$1\frac{3}{4}$	3
30	$3\frac{1}{2}$	$6\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{3}{4}$	$4\frac{1}{2}$
40	$4\frac{3}{4}$	$7\frac{1}{2}$	$2\frac{3}{4}$	$3\frac{3}{4}$	$6\frac{1}{2}$

The mounting of the front wheels requires a study of various angular relationships of which the most important are wheel camber, toe-in, king-pin inclination, castor angle and castor offset. The meaning of these terms is explained by Fig. 73.

As the camber of the wheels gives rise to outward thrusts tending to make them "splay," a compensating effect is

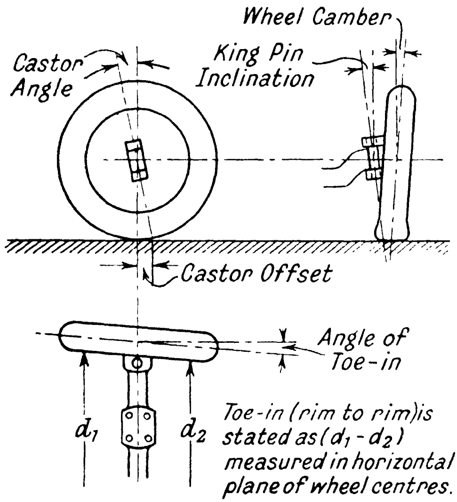

FIG. 73. THE MEANING OF VARIOUS TERMS USED TO DEFINE FRONT-WHEEL MOUNTING, ETC., SHOWN GRAPHICALLY

introduced by "toe-in"; i.e. the wheels are not parallel when viewed in plan. King-pin angles and castor angles give rise to self-straightening torques which augment the torque produced by the tyre itself when it operates with any degree of slip.

The maintenance of these angles depends upon the rigidity of the structure carrying the wheels, and also is in some measure dependent upon correct adjustment. Thus, means are always provided for adjusting toe-in and castor; sometimes camber can also be varied. Many ingenious machines have been developed to assist service stations in this work, which is of the greatest importance to satisfactory handling and to tyre life.

CHAPTER VII

FRONT AND REAR AXLES

ALTHOUGH the front and rear axles of a motor-car have each some special functions to perform, they have also various duties in common which can conveniently be discussed in a single chapter. Thus each axle has to act as a bridge between wheels and supports a proportion of the total weight of the car at the intermediate points where the road springs are attached. In addition to this type of loading, the axle is subjected to forces in a horizontal plane (such as occur when the car is braked, or takes a corner) because it forms the connecting link between the sprung mass and the road wheels. Obviously, every change in the velocity or direction of the car involves reactions between the tyres and the road surface which affect the axles and suspension.

In cars with independent front suspension the wheels are supported directly from the frame, and consequently the structure in question has to perform some of the functions of the absent axle. Designs of this kind were discussed and illustrated in the preceding chapter.

In the design of axles, previous practice must be relied upon to a considerable extent because it is not possible to make allowance for all the conditions of service and to design the unit solely from theoretical reasoning. However, the guiding principles must be understood, and in this chapter we endeavour to present them in a concise form.

The Axle as a Beam. When the car is standing on level ground, each axle carries a share of the total load which, in modern practice, tends to approximate to 50 per cent. It may vary, however, between extremes of 55 per cent front and 45 per cent rear in small cars, up to 40 per cent front and 60 per cent rear in certain large cars.

If the load carried by a pair of wheels, joined by an axle, is W lb. per wheel, and the distance from each road-spring

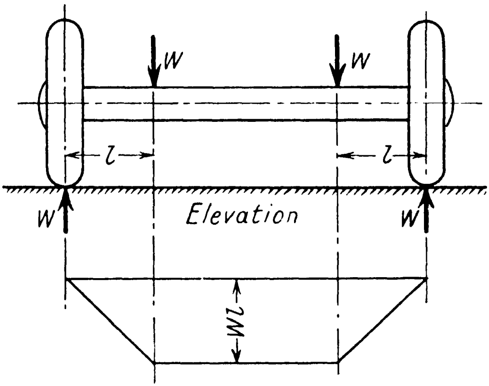

FIG. 74. BENDING MOMENTS DUE TO VERTICAL LOADS ON AN AXLE (VERTICAL PLANE)

to the adjacent wheel is l in., then the central portion of the axle between the springs will be subjected to a uniform bending moment of Wl lb.-in. From the springs outwards the bending moment decreases uniformly to zero, as indicated in Fig. 74. The diagram shows that the bending moment carried by the central part of the axle can be reduced in proportion to any decrease in the length l which practical considerations may allow. Thus in a front axle the separation of the springs is restricted by the need for lateral wheel clearance when steering the car; in rear axles the springs may have to be spaced to suit suspension requirements rather than the axle.

An axle is also loaded as a beam in a horizontal plane by braking forces (front and rear) and by tractive forces (rear

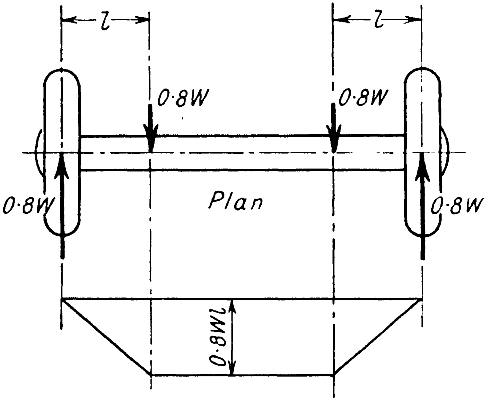

FIG. 75. BENDING MOMENTS DUE TO BRAKING LOADS ON AN AXLE (HORIZONTAL PLANE)

axle only). As explained in Chapter VIII, the braking force which can be applied through a wheel is limited, by tyre adhesion, to a maximum of approximately 80 per cent of the weight carried by the wheel. The greatest horizontal force that can be applied to each end of an axle is therefore equal to $0{\cdot}8W$, producing a bending moment of $0{\cdot}8\ Wl$ lb.-in. in the central portion between the springs (Fig. 75). As tractive forces are subject to the same limitation they need not be separately considered.

A third form of axle loading is produced by the lateral forces which arise when a car is cornering. Here again, the limitation of tyre adhesion applies because the centrifugal force acting upon the car cannot exceed the figure at which the tyres commence to slide outwards. This limit is lower than that applicable to braking and can safely be taken as 0·6 W per wheel. The value attributed to W needs some consideration, however, because it is affected by the action of centrifugal force (C.F.) upon the car as a whole.

The reasoning is similar to that employed in connexion with braking forces and their effect upon the front/rear weight distribution (see page 184). In this case we are dealing with a lateral force through the centre of gravity which, acting in conjunction with the restraining forces applied by tyre adhesion, produces a couple tending to transfer a weight (w) from the inner pair of wheels to the outer pair.

Referring to Fig. 76, if the height of the centre of gravity is h in., the track T in., and the total weight of the car M lb., then, as the overturning couple must be equal to the righting couple, we have—

$$0{\cdot}6\,Mh = wT,$$

and therefore,

$$w = 0{\cdot}6\,Mh/T.$$

This result gives the total weight transferred when the centrifugal force reaches a maximum such that the tyres are about to slide. It does not tell us how the "transfer" is distributed front/rear, but merely indicates that the load on the outer pair of wheels becomes ($\frac{1}{2}M + w$) while the load on the inner pair becomes ($\frac{1}{2}M - w$). It is often considered sufficiently accurate to use the static weight distribution for this purpose, although the distribution of roll couple, as influenced by suspension stiffness (or a stabilizer) should be allowed for, if known. On the static basis, if a car normally carries 45 per cent of the load on the front axle, it is often assumed that under ultimate cornering conditions the outer front-wheel load is 0·45 ($\frac{1}{2}M + w$) and that the inner front-wheel load will then be 0·45 ($\frac{1}{2}M - w$).

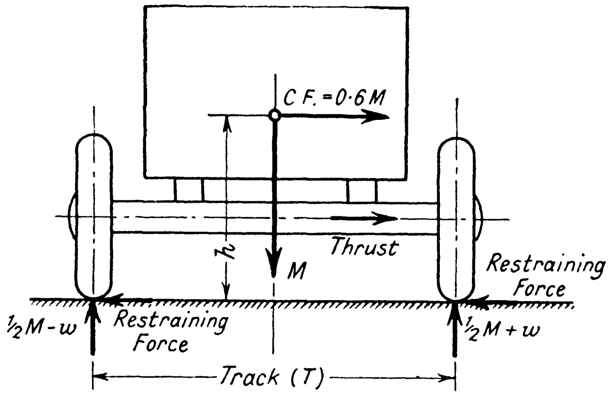

FIG. 76. TRANSFER OF WEIGHT DUE TO THE OVERTURNING COUPLE, ON A CORNER

Rear-wheel loads are estimated in the same way. It is also assumed that each wheel carries a side-thrust, at road level, equal to 60 per cent of the load carried by the wheel and that this is equal to the end-thrust on the hub bearings. These forces form a couple with a leverage equal to the rolling radius of the tyre.

Before proceeding to consider these hub loads and stresses, the torsional loads applied to the ends of an axle, by braking, must first be described. The maximum braking torque per wheel is 0·8 Wr (see page 175) and this tends to drag the shoes and backplate in the direction of the rotating drum. The backplate is anchored to the axle, which, in turn, is prevented from revolving by its attachment to the road spring. Consequently, the end of the axle projecting beyond the spring carries the full brake torque.

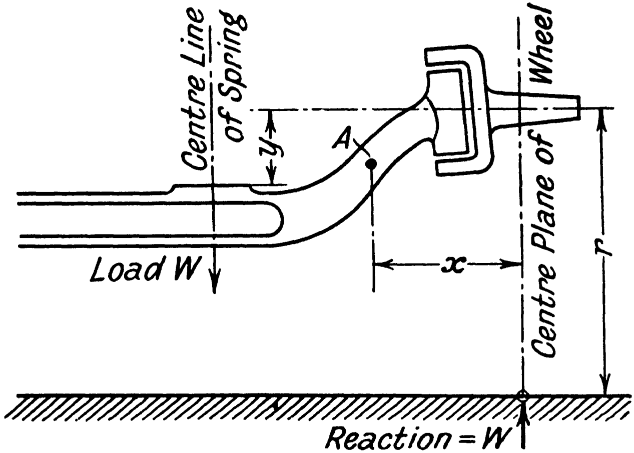

FIG. 77. EFFECT OF "CRANKING" (OF THE FRONT AXLE) UPON BRAKE TORQUE

In the case of the front axle, the ends are cranked upwards and therefore the horizontal braking forces obtain a leverage which results in a torque contrary to that produced by the drag on the backplate. It is not, however, a uniform torque, but one which increases from zero at the extreme end to a maximum at the spring attachment. If the height of the wheel-hub centre line above the spring seat is y in. (Fig. 77), the maximum torque due to this degree of cranking will be $0{\cdot}8\ Wy$.

The "I-beam" section of a front axle, and the hollow circular section of a rear-axle casing, can be checked over to ascertain the maximum stresses produced by all these forms of loading, using the customary formulæ for bending and torsion. Such calculations are usually supplemented by full-scale bending tests which enable the deflections of the axle to be checked under load.

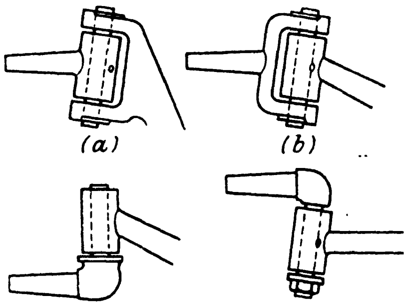
(a) (b) (c) (d)

FIG. 78. FOUR TYPES OF STEERING KNUCKLE

The Stub Axles. In the case of a front axle, provision for steering is made by "hinges" at each end of the axle beam. These are classified as the Elliott, reversed Elliott, Lemoine and reversed Lemoine (Fig. 78 *a*, *b*, *c*, and *d* respectively). Of these the reversed Elliott (*b*) is almost universally employed, so that the other types are mainly of historical interest. A boss forged at each end of the axle beam is drilled to take a king-pin on which swivels the fork of the stub axle. Bronze bushes surround the pin and, to take the end-thrust, either a plain washer or a ball-bearing may be used.

The loads carried by the stub axle and bearings can be estimated as follows. The stub axle carries the direct wheel load and is also subjected to severe stresses when cornering. Referring to Fig. 79, the wheel load is assumed to be W lb. (as before) and acts along the centre-plane of the wheel at a distance of d in. from the king-pin axis. The separation of the king-pin bushes is D in. and the load carried by each of them is w lb. Then, by equating moments, we find that

$$w = Wd/D.$$

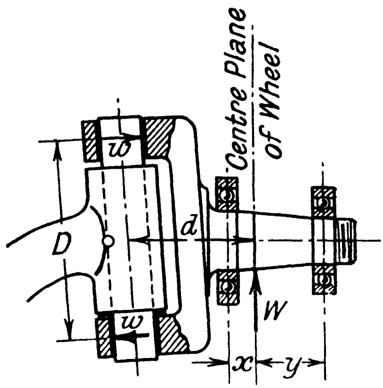

FIG. 79. FORCES ACTING ON A STUB AXLE AND STEERING KNUCKLE

It is obviously advantageous to reduce the overhang (d) to a minimum. To do so will also reduce the bending moment produced by the load on the stub axle. However, the maximum bending moment which a stub axle may carry occurs when cornering and, although still in a vertical plane, is reversed in direction as compared with that due to normal loading.

Under static conditions the hub bearings each carry a part of the load (W) which is in inverse proportion to their distances (x and y) from the centre plane of the wheel; thus the inner bearing carries by far the greater proportion. Under cornering conditions this distribution becomes very different. First, the load on the outer wheel is greatly increased. Secondly, the lateral force exerted on the stub axle, and the restraining force, at tyre contact, produce a canting effect on the wheel which gives rise to forces that are superimposed upon the normal radial loading of the bearings (Fig. 80).

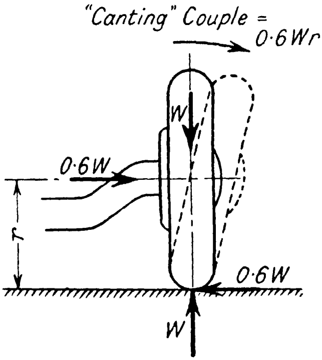

FIG. 80. CANTING COUPLE APPLIED TO A WHEEL WHEN CORNERING

Still using the symbol W for the wheel load, but remembering its greatly augmented value for the outer front wheel on a corner, the maximum lateral force will be 0·6 W and the canting couple is therefore 0·6 Wr lb.-in. This produces equal and opposite auxiliary loads which add to the normal load on the inner bearing of the pair and act in a direction opposite to that of the normal load carried by the outer bearing of the pair. The amount of each auxiliary load (L) is found by dividing the canting couple by the distance separating the two bearings. It is therefore given by—

$$L = 0{\cdot}6\ Wr/(x + y).$$

Reverting to the stub axle, the bending moment in a vertical plane due to cornering must be equal to the canting couple on the wheel (0·6 Wr) and acts in opposition to the direct bending moment produced by the load (Wd) in the case of the outer wheel of the pair. For the inner wheel the two bending effects are in the same direction, so that, although the vertical load is much smaller, the net bending moment may in some cases be larger than that experienced by the stub axle of the outer wheel.

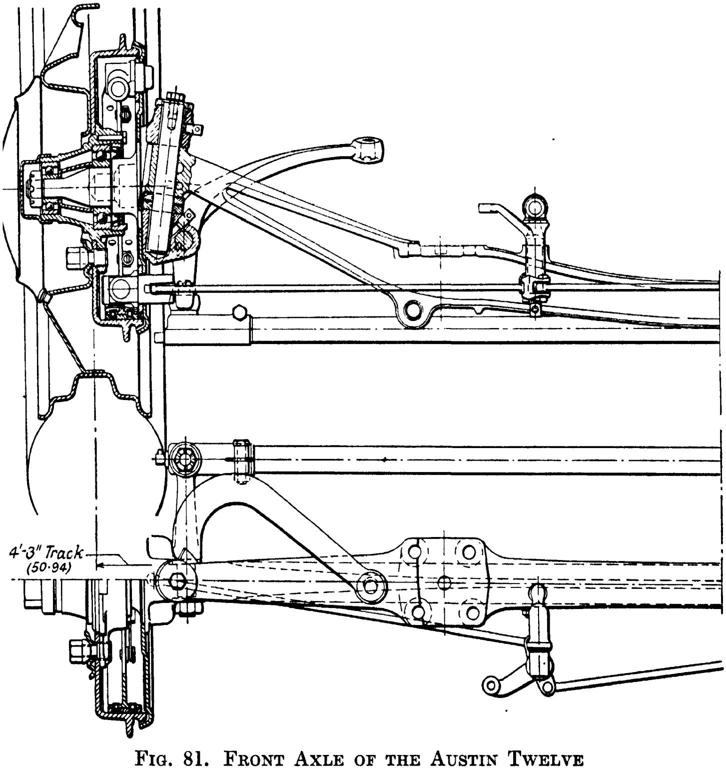

FIG. 81. FRONT AXLE OF THE AUSTIN TWELVE

Before concluding this study of front-axle design, two examples from current practice will be helpful. A drawing of the Austin Twelve axle is shown in Fig. 81; this carries a load of 615 lb. per wheel on a track of 51 in.; the separation of the springs is 22½ in. Each hub runs on two radial ball-bearings of which the inner unit is located endwise to take all the end-thrust.

An alternative design is shown in Fig. 82, this being the Standard Ten front axle used for a car which is now supplemented by an independently sprung chassis. It will be noticed that in this instance the hub is carried by tapered roller bearings. The track is 44⅛ in., the spring separation 21½ in., and the wheel load 546 lb. The "I-section" beam merges into an oval section at the axle ends, well adapted to resist brake torque.

The Rear-axle Structure. The casing of the rear axle carries loads similar to those already described for a front-axle beam and, in addition, supports and encloses the mechanism which conveys the drive from the propeller shaft to the rear wheels. The casing is now often made from pressings which are welded or riveted together, and its design is largely empirical, i.e. is based upon previous practice and experimental test results rather than upon elaborate stress calculations. By proceeding on these lines a remarkable decrease in the weight, complexity, and cost of the rear axle has been effected during the past ten years.

The final drive and differential gears are separately treated in Chapter V (Transmissions), so that here we need mention only the fact that they are usually mounted as an assembly carried by ball or roller bearings. These bearings are in turn supported by pedestals formed in a flanged casting, which is secured to the axle casing by bolts or studs.

The bending moment produced by the downward load

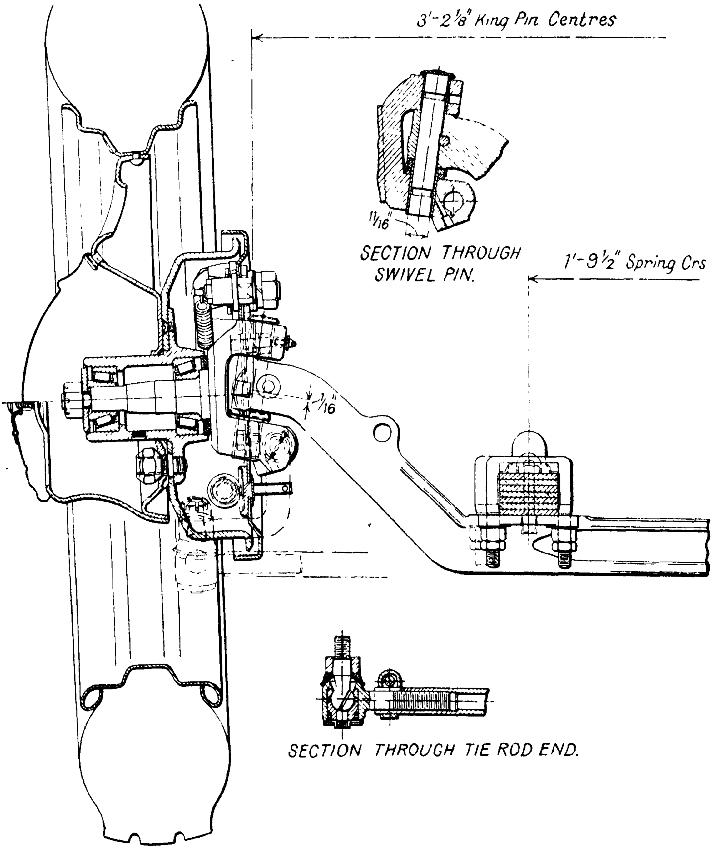

FIG. 82. FRONT AXLE OF THE STANDARD TEN

on the axle casing depends upon the positions of the road springs in relation to the wheels (see page 147). In the popular system known as "Hotchkiss drive" the springs also carry tractive and braking forces, so that bending moments in a horizontal plane can be determined as for a

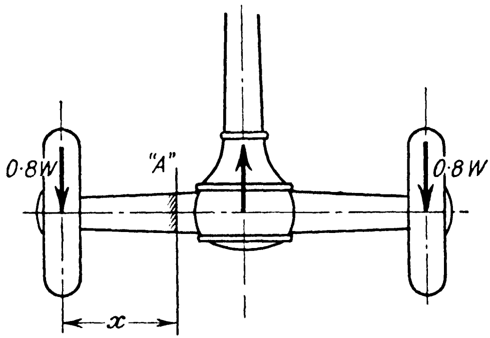

FIG. 83. LOADING OF A TORQUE-TUBE-AXLE SYSTEM BY BRAKING FORCES

front axle. In the alternative torque-tube system, horizontal forces applied through the hub bearings are transmitted to the chassis through the tube and therefore produce a very considerable bending moment adjacent to the centre of the casing (Fig. 83) which is of the "cantilever" type. At any section such as "A" the bending moment due to braking may reach a value of $0{\cdot}8\ Wx$.

Drive-shafts and Bearings. Three ways in which the shafts and bearings can be arranged in relation to the casing are shown in Fig. 84, these being known as (*a*) semi-floating, (*b*) three-quarter floating, and (*c*) fully floating. Of these the third is used only in heavy vehicles, for which it has the fundamental advantage that the drive-shaft (S) carries torque alone, all other forces being conveyed from the hub (H) to the axle casing (C) directly by the hub bearings.

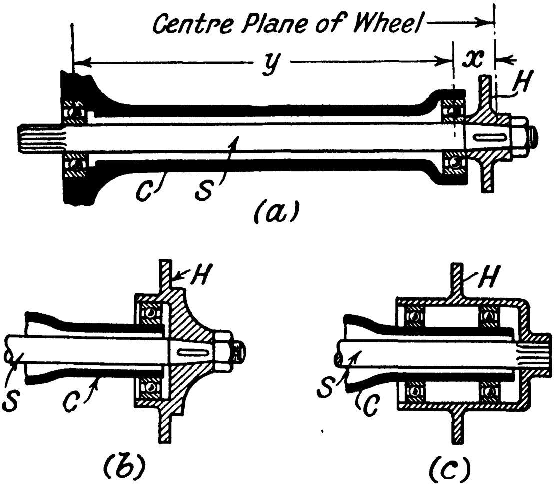

FIG. 84. THREE TYPES OF REAR-AXLE LAY-OUT

Owing to its lightness and simplicity, the semi-floating axle is very widely used, although it necessitates a drive-shaft strong enough to carry bending loads as well as torque. Thus, forces acting in the plane of the wheel, due to traction, braking or the downward weight of the car, exert a leverage equal to the overhang beyond the outer bearing (x, Fig. 84 (a)) and so tend to bend the shaft. They also produce a reaction on the inner bearing which we will denote by p (lb.). Then, if the wheel load is W lb. and the distance separating the bearings is y in., moments taken about the outer bearing show that—

$$py = Wx,$$

and therefore—

$$p = Wx/y.$$

The load on the outer bearing (P) is therefore given by—

$$P = W + p.$$

The bending moment in the shaft produced by the load is simply (Wx) under normal static conditions, but becomes greatly modified by the alteration in wheel loads, together with the creation of transverse forces, which occur when the car is driven on a curve. These loads and forces can be calculated as for a front stub axle and bend the drive-shafts in a vertical plane with a leverage equal to the rolling radius of the tyres. In the case of the outer wheel (on a curve) this bending moment acts in opposition to that produced by the downward load, while for the inner wheel it augments the bending moment due to the downward load. The figures can be calculated for any given case, and the higher stress so found can then be combined with the maximum shear stress in the shaft due to drive torque, by the usual Combined Stress formulæ.

There are two ways of ascertaining the greatest torque which the shaft may have to carry. One is to multiply the maximum engine torque by the overall bottom-gear ratio, halving the result to obtain the torque carried by each drive-shaft (neglecting transmission losses). It will be appreciated that the differential ensures an equal division of torque between the two shafts. An alternative plan is to calculate back from the limit set by tyre adhesion, taking the greatest torque as being that which will cause wheel-spin on a good, dry surface. This torque will be simply 60 per cent of the wheel load multiplied by the rolling radius of the tyre.

As the drive-shafts carry shock loads and are subject to fatigue, good materials and a high factor of safety are needed as an assurance against failure. Any sudden change of section should be avoided because it will create high local stresses and may initiate a fatigue failure. A particularly critical section in the shaft is adjacent to the outer bearing. The connexion between shaft and hub is usually by taper and key, secured by a castellated nut with split pin.

In a fully floating design, each drive-shaft is splined to the wheel hub and its section is calculated on torque alone. The loads carried by the hub bearings—direct, and when cornering or braking—are calculated as for front-hub bearings.

In the intermediate three-quarter floating axle the outer bearing is set almost in the plane of the wheel and so carries radial forces due to vertical load, braking and traction without calling upon the drive-shaft to withstand any appreciable bending effect. Transverse forces which tend to cant the wheel when cornering do, however, apply a bending moment to the drive-shaft and therefore modify the loading on both inner and outer bearings in much the same way as in a semi-floating design.

End Loads when Cornering. The restraining forces which act transversely between the tyres and the road when the car is cornering not only tend to cant the wheels but also produce end-thrusts forcing the outer wheel towards the car and the inner wheel away from the car. In a fully floating axle these end-thrusts can be taken by the hub bearings, which are mounted directly on the axle casing.

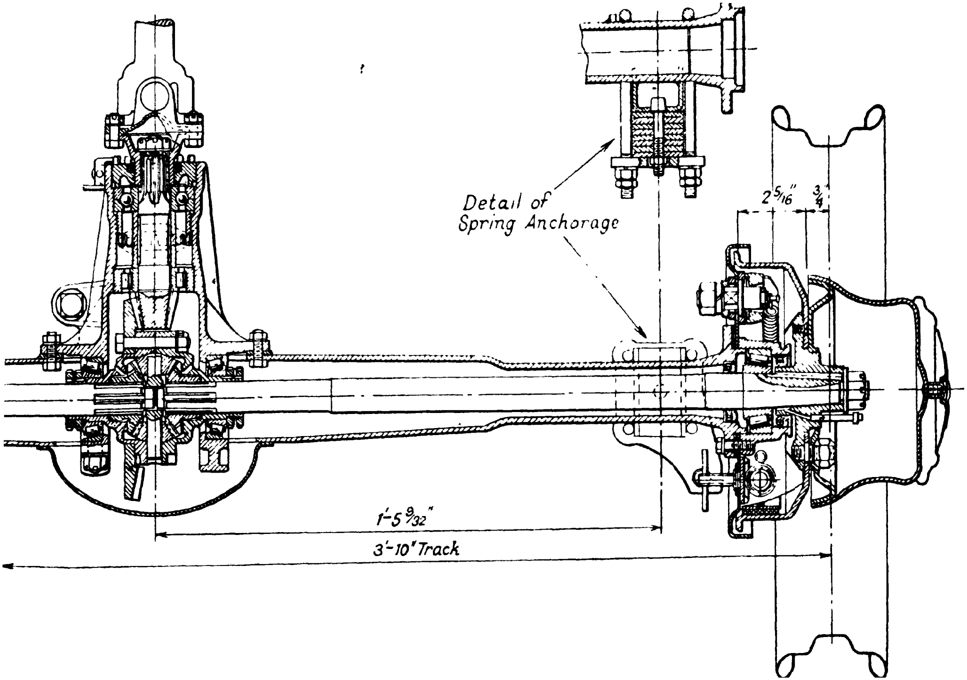

Fig. 85. Semi-floating Rear Axle of the Standard Ten

In other types of axle a variety of methods for taking end-thrust are used. Thus, in some cases, thrusts are taken by the differential assembly bearings; in others, by the hub bearings adjacent to the wheels. Hub bearings may be located and designed to withstand thrust in two directions or in one direction only; in the latter event, a thrust pad is provided between the inner ends of the drive-shafts so that an endwise force applied to one wheel can be carried through the shafts to the hub bearing of the opposite wheel.

A good example of this last-mentioned practice is the semi-floating rear axle of the Standard Ten (Fig. 85) in which tapered-roller hub and differential bearings are used. It will be seen that the drive-shafts butt together at the centre; incidentally, they are made from nickel-chrome molybdenum steel. This axle carries a total load of 1540 lb. with the car fully laden, the wheel track being 46 in.

An example of the alternative three-quarter floating type of rear axle is shown in Fig. 86, this being the design used in the Austin Twelve. Each hub is carried by a single ball-bearing mounted on the end of the axle casing, almost exactly in line with the central plane of the wheel; the hub is secured to the drive-shaft by a taper and key. The differential side gears are formed in one piece with each drive-shaft and butt against the spider at the centre. End-thrusts on the hubs can therefore be conveyed through the shafts, in either direction, to the large bearings which carry the differential assembly and which are adjustable for location by means of ring nuts (*S*).

Oil seals are very important in rear-axle design because lubricant carried at the centre of the casing will always work its way along the rotating drive-shafts to the outer ends. It must be prevented from going farther because it will otherwise reach the interior of the rear brake-drums.

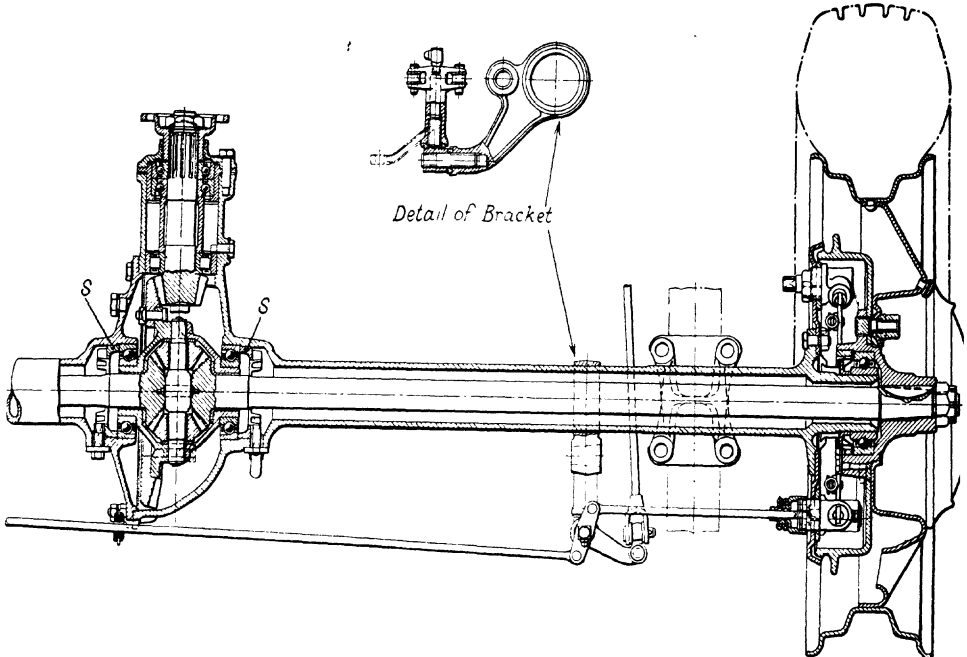

FIG. 86. THREE-QUARTER FLOATING REAR AXLE OF THE AUSTIN TWELVE

The sealing rings used for this purpose by Standard and Austin can be seen just outside the hub bearing in Fig. 85 and on the inside of the hub bearing in Fig. 86; similar arrangements are made in most modern axles. A seal must also be provided where the pinion shaft emerges from the "nose" of the axle centre.

CHAPTER VIII

BRAKES AND BRAKE PERFORMANCE

EVER since the use of four-wheel brakes became general, in 1925, the trend of design has been consistently directed towards simplification, a reduction in the energy lost between the pedal and the brake shoes, and the elimination of parts which, by developing play or by lack of rigidity, were apt to cause "lost motion" in the system. It is also notable that whereas many car manufacturers were at one time responsible for the design and making of their own individual brake gear, the majority now equip their cars with one or other of three leading proprietary systems: namely, Lockheed, Girling, and Bendix.

Under modern traffic conditions a high performance is expected of the braking system, and it must also be capable of giving consistent results over fairly long mileages without adjustment. The response obtained from the brakes should be proportional to the effort exerted at the pedal in order that the driver may be able to control his vehicle accurately. Another requirement is that the effort needed to give an emergency stop should be well within the physical capabilities of a driver of average physique. Finally, the brakes must not show excessive "fade" on a long descent, or during a quick succession of stops in traffic; this implies that the effective braking area should be adequate for the weight and performance of the car concerned, and that cooling of the drums should not be impeded by the position of the surrounding parts.

Brake Performance. Before proceeding to consider questions of design, the performance expected of the service brakes must first be laid down. The ultimate retarding effect obtainable is limited by the horizontal forces which tyre adhesion will withstand before the wheels cease to roll and commence to slide. Apart from exceptional cases, this limit is reached when the total retarding force amounts to about 80 per cent of the running weight of the car to which it is applied. The deceleration obtained will then be $0{\cdot}8\ g$, i.e. 80 per cent of that produced by gravity, or 25·76 ft. per sec. per sec.

It has become quite a common practice to speak of this percentage as "brake efficiency." Thus brakes are said to be 80 per cent efficient if they provide a maximum deceleration, on a good, level, dry road, of $0{\cdot}8\ g$. Although this is quite a convenient plan it may give rise to misconceptions if not properly understood. For example, on a slippery road the best of brakes might not be capable of giving more than 20 per cent, but as this would be due to lack of tyre adhesion it is scarcely logical to say that the "brake efficiency" was only 20 per cent. However, as this way of expressing deceleration has become so widely used it will be employed in this chapter.

The relationship between speed, stopping distance, and efficiency can easily be derived from the usual formula for the kinetic energy of a moving body, rewritten in the following approximate form, in which W is car weight (lb.) and V is car speed (m.p.h.)—

$$\text{Energy in ft.-lb.} = WV^2/30.$$

Assuming that the brakes provide a uniform retarding force (F lb.) throughout the stopping period, and that the stopping distance is represented by D (measured in ft.; see Fig. 87), then the product (FD) is the work done by the brakes and must be equal to the kinetic energy dissipated in bringing the car to a standstill. Equating these results and substituting braking efficiency (E) for the ratio F/W, expressed as a percentage, we get the result—

$$D = V^2/0{\cdot}3E$$

This shows that for any given efficiency (or deceleration) the braking distance is independent of the weight of the car, and is proportional to the square of the speed. Furthermore, the distance is inversely proportional to any change

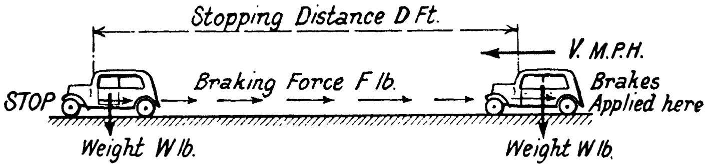

FIG. 87. THE WORK DONE IN STOPPING A CAR IS EQUAL TO THE KINETIC ENERGY DISSIPATED

of efficiency. Thus, if E is 80 per cent, the corresponding stopping distance will be 37·5 ft. from 30 m.p.h., 150 ft. from 60 m.p.h., and so on; if the efficiency were halved these distances would each be doubled. Curves of stopping distance plotted against car speed for various efficiencies are shown in Fig. 88.

In actual fact the retarding force does not remain uniform during a stopping period, particularly from the higher speeds where the energy dissipated creates enough heat to produce a certain amount of fade before the car comes to a stop. The driver may compensate for this by progressively increasing the pedal load during deceleration, but if the load is held constant the efficiency tends to fall off. Another

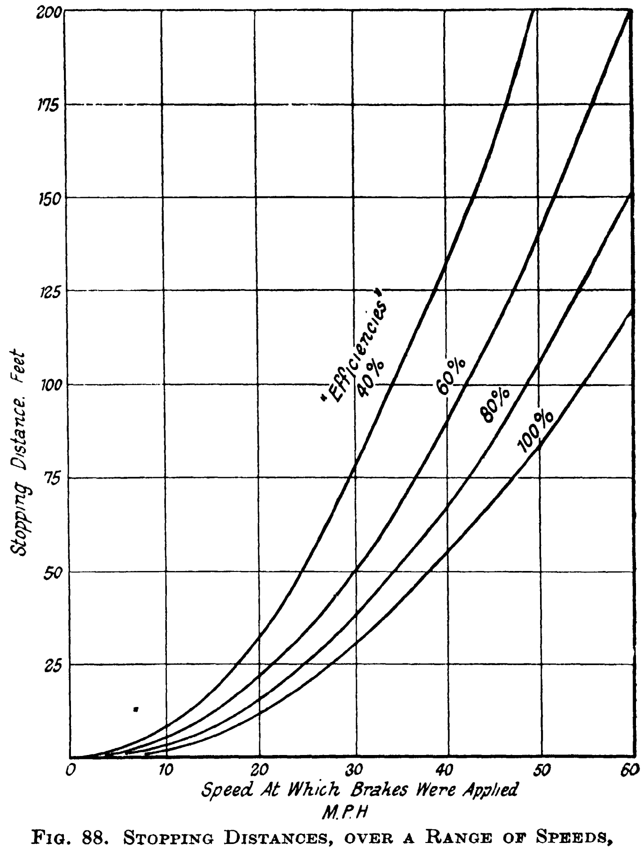

FIG. 88. STOPPING DISTANCES, OVER A RANGE OF SPEEDS, PLOTTED FOR VARIOUS BRAKE EFFICIENCIES

practical consideration, which becomes important when estimating brake performance in connexion with accident cases, is that an appreciable reaction time elapses between the instant at which the need for braking is realized and the instant at which the brakes become fully applied. This period may vary between half a second and slightly more than one second, according to the capabilities of the driver concerned, and represents a considerable distance on the road at high speeds.

Braking on Gradients. A convenience of the "efficiency" method of describing brake performance is that it can easily be applied to braking on gradients. This follows from the fact that the ratio between car weight and the force parallel to the slope (caused by gravity) is equal to the ratio between the height and hypotenuse of a triangle representing the gradient (see p. 49). For example, on a slope of 1 in 5 the force required of the brakes is $\frac{1}{5}$ the weight of the car. This can equally well be termed a 20 per cent gradient, and a car can therefore be held from rolling down the slope by brakes exerting an "efficiency" of 20 per cent.

It follows that for emergency braking on a gradient the deceleration actually available for stopping can be derived by adding, or subtracting, the percentage gradient to, or from, the brake efficiency available on the level. Thus, brakes capable of giving a maximum of 80 per cent will provide a deceleration equivalent to an "apparent efficiency" of 100 per cent when ascending a 20 per cent gradient, or will give 60 per cent when the car is descending the same slope. From this "apparent efficiency" the stopping distance on any gradient can be calculated, using the formula already given.

Relating Pedal Load and Deceleration. An important factor in brake performance is the relationship between deceleration and pedal load. For test purposes this can be ascertained by interposing a pressure-recording device between the foot and the brake pedal, so that a series of known loads can be applied while the corresponding decelerations are read from a meter of the inertia type.

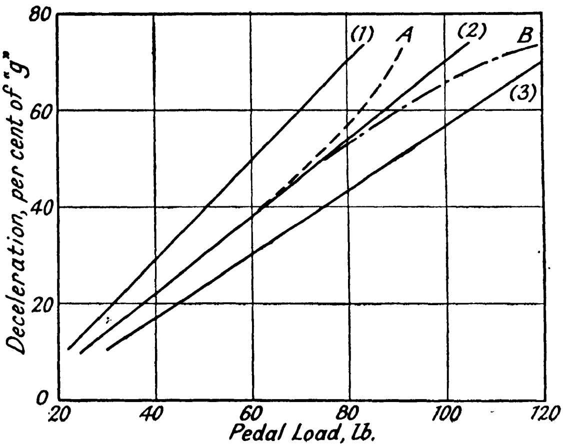

FIG. 89. DECELERATION (PER CENT) PLOTTED AGAINST PEDAL LOAD

In passing, it is worth noting that attempts to measure stopping distance directly do not give reliable results owing to the difficulty of recording the position on the road, and the exact speed of the vehicle, at the instant of brake application. The meter is independent of both these factors.

The graph in Fig. 89 shows deceleration expressed as a percentage of g (brake efficiency would be a confusing term n this case) plotted against pedal load in lb.; a straight-ine relationship, marked No. 1, is of the kind popular in small cars, and it will be seen that a load of 80 lb. gives a igure of 70 per cent. Some makers would consider that a driver can gauge the extent of brake application more accurately, and is less likely to brake excessively in an emergency, if the pedal load is graded in accordance with line No. 2. For heavy cars, still higher pedal loads are often required, as in line No. 3.

If there is a deviation from the straight-line law towards the upper end it is important to notice the nature of this tendency. The alternatives, in relation to line No. 2, are shown by curves A and B in Fig. 89. Curve A is a dangerous condition because the driver gets, in effect, more braking than he had bargained for and will find it difficult to avoid locking the wheels under emergency conditions. Curve B is a safe (though not wholly desirable) condition, which implies that more effort than expected will be needed to get the higher rates of deceleration.

Lining Area and Drum Size. Average figures for the total brake-lining area required for cars of various weights are useful, but must be employed with discretion. It is true that the kinetic energy dissipated under any given stopping condition is proportional to car weight, and therefore, in order to keep within the limitations required for satisfactory lining service, the lining area should be proportioned in accordance with the weight to be handled. In most cars the ratio works out at some figure between 25 and 30 lb. of car weight per square inch of brake lining employed, the weight taken for this calculation being kerb weight plus a 450 lb. allowance for passengers.

This simple ratio takes no account of the disposition of the lining which varies in effectiveness according to the arc of shoe contact; in most cases a matter of about 90 degrees. If the arc is lengthened, the lining area increases proportionately, but in actual fact it has been added in places where it can do relatively little work and consequently cannot make much difference to the rate of lining wear. A second defect of the simple weight/area ratio is that no allowance is made for the fact that the brakes have a heavier duty in a fast car than in a slower vehicle. For this reason the following calculation can usefully be made.

The basis is the rate at which energy can be dissipated per square inch of lining without reaching undesirable temperatures. Messrs. Raybestos give this as 50,000 ft.-lb. per min. From the kinetic energy formula it is easy to derive the following relationship in which E is brake efficiency and V the maximum speed (m.p.h.) of the car concerned (the ratio K is car weight divided by total lining area, expressed in lb. per sq. in.)—

$$K = 113{,}750/EV.$$

If we take 65 per cent as a reasonable value for E for a safe emergency stop from high speeds and assume that the maximum speed for which the car is designed (V) is 60 m.p.h., then the value of K works out at 29·2 lb. per sq. in. Under similar conditions a 90 m.p.h. car should require 50 per cent more lining area or a ratio (K) of 19·5 lb. per sq. in. These figures are useful for comparative purposes.

It is obvious that for given arcs of contact the lining area required in a braking system can be obtained either by using small and wide drums or from large and narrow drums; in theory it is a matter of indifference what proportions of diameter and width are selected. In practice, however, the choice of drum diameter is often governed by the space available, remembering that 16 in. wheel rims are now commonly used on almost all sizes of car. There is also the fact that a large diameter drum of sufficiently rigid design (to avoid distortion) is heavier than a drum of small diameter and larger effective width. Popular internal drum diameters are 8 in. for small cars weighing less than 2000 lb., 9 in. for most medium-sized cars, and 10 in. or more for larger vehicles.

Iron drums have largely superseded the pressed-steel type, mainly because they can more easily be made in rigid sections using external ribbing. It is important to avoid distortion because this prevents the lining from working efficiently over the full area of contact and also because it may result in "grab." For smooth braking the internal working surface of the drum must be circular within fine limits and must not run out of truth. The latter condition depends upon the mounting and attachment of the hub and wheel as well as the accuracy with which the drum itself is made. In a few modern cars the drum and hub are formed in one piece from malleable iron, but the more usual practice is to make them as separate components.

Braking Forces and Leverages. A study of the leverage in the system between the brake pedal and the drums is necessary in order to find the conditions required to establish reasonable figures for pedal load in relation to deceleration obtained. A wheel and braking unit are shown diagrammatically in Fig. 90, the ratio between the retarding force (F) produced by the brakes and the weight (W) carried by the tyre being approximately equivalent to the term "efficiency" which we have been applying to the car as a whole. The brake torque exerted on the wheel will be the product (Fr), where r is the rolling radius of the tyre in inches. Consequently, if we require an F/W ratio of 80 per cent, the brake torque needed to give this result is easily calculated from known values of weight and radius. Dividing this torque by the internal radius of the brake drum will give the total tangential force which must be produced by the linings at their working surfaces.

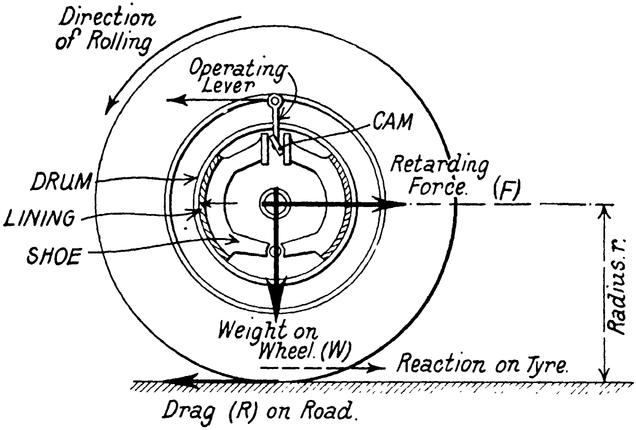

FIG. 90. DIAGRAMMATIC VIEW OF A BRAKED WHEEL, SHOWING THE PRINCIPAL FORCES ACTING ON THE WHEEL, AXLE AND ROAD

In the simple type of brake shown in Fig. 90 the shoes move outwards to an equal extent when the cam is turned, and their combined effect upon the drum can be calculated with sufficient accuracy by disregarding servo action and by assuming that the ratio : (tangential force)/(load on shoes) is equal to the coefficient of friction between linings and drums. An example will show how the calculation is made.

Assume that the wheel load (W) is 625 lb. and that we require a retarding force equal to 80 per cent of this load, namely, 500 lb. Then, if the internal brake drum diameter is 8 in. and the rolling radius of the tyre is 12 in., the tangential force at the working surface of the drum is given by—

$$\text{Force} = 500 \times 12 \div 4 = 1500 \text{ lb.}$$

The coefficient of friction depends upon the brake lining employed, a common figure being 0·4. Dividing the tangential force by this figure gives the total radial load required on the brake shoes, namely, 3750 lb.

It follows that if the total leverage between brake pedal and the working surfaces is 150 to 1, the effort needed at the pedal for each braking unit will be 25 lb., i.e. a total effort of 100 lb. for a four-wheel braking system. As the shoes themselves provide a leverage of 2 to 1, we shall need a ratio of 75 to 1 between pedal and shoe tips, provided by all the levers in the system, including the pedal and the shoe expanders.

This calculation takes no account of the losses in the system which must in actual practice be overcome by augmented pedal effort. One of the advantages of simplification is that mechanical friction is reduced and, with it, the strength of the pull-off springs required to retract the shoes when the pedal is released. Modern mechanical systems (and also the hydraulic system) have reduced losses to such a point that the energy delivered to the shoes is usually not less than 90 per cent of the energy applied to the pedal.

The limitation on the leverage which can be used is provided by two factors: the minimum practicable brake-shoe clearance and the maximum permissible pedal travel. Thus, apart from the actual travel required to bring the shoes into contact (from the "off" position), with the system adjusted to minimum clearances, a little extra travel may be needed to stretch the operating mechanism before the shoes become fully loaded. To reduce this "extra," the system should be designed as rigidly as possible by the

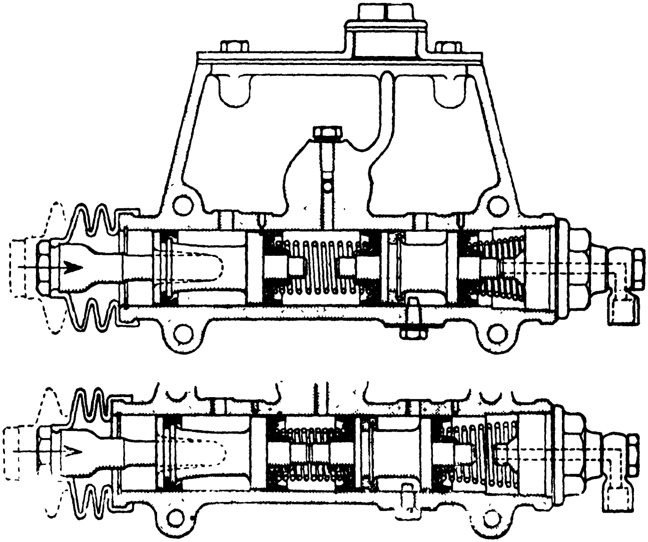

FIG. 91. THE LOCKHEED TANDEM MASTER CYLINDER "CLOSES OFF" ONE OR OTHER OF TWO BRAKE LINES, SHOULD LEAKAGE OF FLUID OCCUR

elimination of long cross-shafts and similar parts. Beyond the operating range there must be enough remaining space between pedal and toe board to allow for a reasonable amount of lining wear before brake adjustment becomes necessary.

The *distribution* of leverage is also an important point in the design of mechanical systems. By concentrating the maximum effect at the shoe-expanders, the leverage in the rest of the system can be reduced and, with it, the stresses and liability to stretch. Furthermore, the interference caused by axle movement as the road springs deflect and react is minimized when the travel of the operating gear is increased. In hydraulic systems this interference is eliminated because flexible hose connexions are used between car and axles. Leverage becomes a matter of the relative sizes of the master cylinder and the wheel cylinders.

"Compensation" is the name given to devices which distribute the pedal effort equally between the four shoe-expanders. Wherever compensation is provided between two parts the failure of one of them will render the other inoperative unless safeguards are arranged. In mechanical systems the safeguard takes the form of stops which limit the travel of a swinging lever or similar device. In an hydraulic system a tandem master cylinder may be employed (Fig. 91), so that if leakage occurs in one part it becomes automatically closed off to allow hydraulic pressure to be maintained on the other part.

Self-applying Brake Shoes. In a two-shoe brake the drag of the rotating drum is such as to carry the leading shoe towards the pivot while the trailing shoe tends to be moved away from the pivot. The diagram in Fig. 92 illustrates an imaginary experiment on a leading shoe which is pressed against the drum by a measured force F. The tangential force (P) is assumed to be just sufficient to keep the drum turning at a steady speed and we will also assume that it acts at the radius of the friction surface. It will then be found that the ratio P/F is higher than the known coefficient of friction of the lining in question.

The reason for this state of affairs is that the force F is being augmented by a force created by the drag of the drum and its moment about the shoe pivot. If we could measure this supplementary force (f) we should find that the ratio $P/(F + f)$ would be equal to the true coefficient of friction.

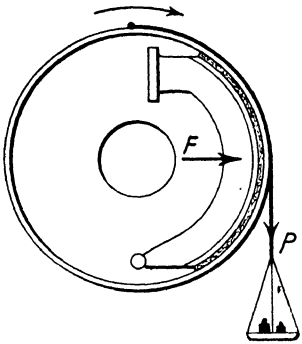

FIG. 92. SIMPLE EXPERIMENT TO SHOW BRAKING EFFECT OF A LEADING SHOE

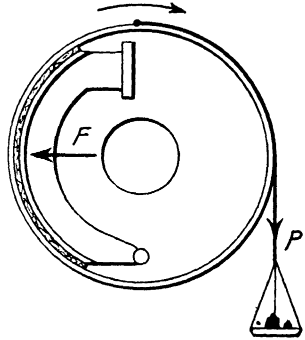

FIG. 93. AN EXPERIMENT SIMILAR TO THAT OF FIG. 92, BUT APPLIED TO A TRAILING SHOE

By repeating the experiment upon a trailing shoe (Fig. 93) we should find that the applied force F was being diminished by the drag of the drum to an extent f, with the result that the coefficient of friction would be equal to $P/(F - f)$. This is a general statement which needs modification in special cases but illustrates a broad principle.

Another and more precise way of analysing the self-applying effect of the leading shoe is shown in Fig. 94. Here we imagine that the lining is divided up into a number of small sections. Concerning section A, which is pressed against the drum by a shoe pivoted as shown, the forces which it experiences are the radial reaction of the drum (F) and a tangential drag (P). The ratio between these forces is determined by the coefficient of friction and controls the inclination of their resultant. It is clear that in the case

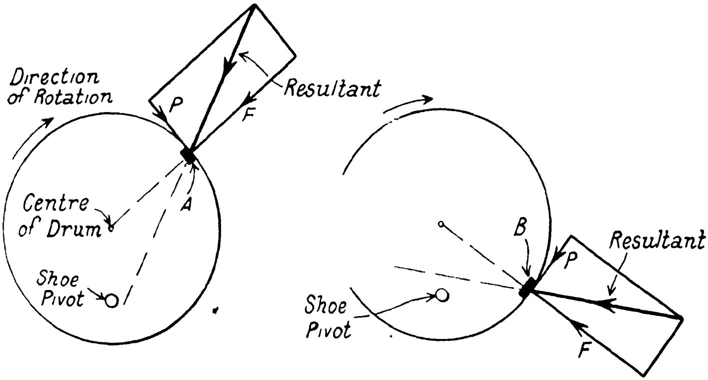

FIG. 94. SELF-APPLYING EFFECTS CAN BE STUDIED BY DIVIDING THE LINING INTO SMALL AREAS, SUCH AS THOSE MARKED A AND B

shown the resultant acts to the right of the shoe pivot and therefore has a self-applying effect upon the lining at A. We can also see that this effect will increase if (a) the coefficient is increased; (b) the location of the shoe pivot is raised; (c) the shoe is lengthened, so that the piece of lining under consideration is brought nearer to the vertical centre line.

The second diagram in Fig. 94 shows the forces acting upon a piece of lining (B) at the other end of the shoe, where the resultant has a "self-freeing" effect. The action of the shoe as a whole will depend upon an integration of the moments exerted by all the small sections into which we may assume the lining to be divided.

As already indicated, a simple two-shoe brake has a

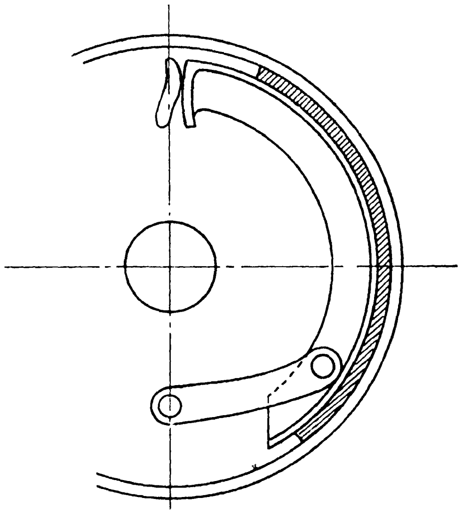

FIG. 95. LINK-MOUNTED BRAKE SHOE AS USED BY THE VAUXHALL CO.

negligible servo effect. In order to obtain any advantage from the self-applying property just discussed, the leading shoe must be so mounted that it can "float," and the cam (or expander) must be able to "follow up." Although a servo action can provide valuable assistance, and enables the leverage of a braking system to be reduced, there are limitations to the extent to which it can be employed. One of these is the risk of "grabbing," which is minimized in certain designs by allowing the leading shoe a certain degree of freedom instead of making it work in a fixed relationship with the drum. Examples are the link-mounted shoe used by Vauxhall (Fig. 95) and the Lockheed slotted shoe (Fig. 96). Then again, some servo systems have a "self-freeing" action in reverse, so that a brake-system leverage adequate for forward travel may require an uncomfortably large pedal effort when the car has to be held against running back down a steep gradient.

For smooth brake application it is essential to keep the shoes square with the drum. This implies that the backplate on which the assembly is mounted should have considerable rigidity and that the shoes should be retained against accurately machined surfaces on the backplate by means of spring-loaded links or similar devices. It will also be appreciated that eccentricity or elliptical distortion of the brake drum may provide a cause of grabbing.

In order to meet legal and practical requirements the brake pedal must be supplemented by an alternative means of operation which almost invariably takes the form of a hand lever interconnected with the normal service system and working either all four brakes or just the rear pair. A lost-motion device, such as a slotted link, enables the pedal to be depressed without moving the hand lever, and vice versa. The interconnexion must be so arranged that the failure of any one part of the system will still leave two brakes under the control of the driver. In hydraulic systems, to meet this condition, the hand lever must be mechanically connected to at least two pairs of brake shoes.

A ratchet and pawl are usually provided, to enable the handbrake to be "set," although in some recent cars the

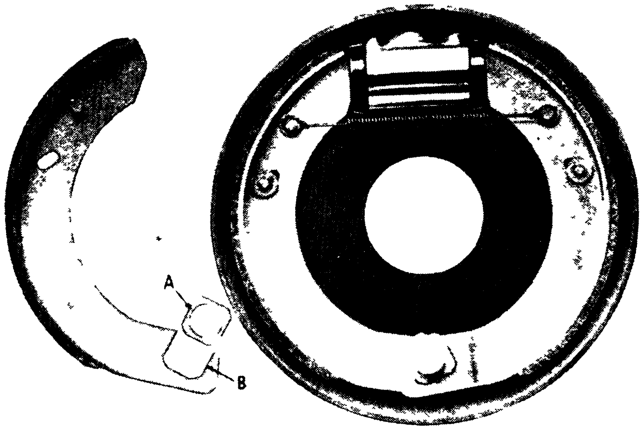

FIG. 96. LOCKHEED SLOTTED SHOE WITH TRUNNION MOUNTING

holding device takes the form of a self-locking frictional control. A modern trend is to mount the hand lever beneath the instrument panel so as to leave the floor space unobstructed.

Weight Transfer When Braking. The static loads on the front and rear wheels of a car become altered by any

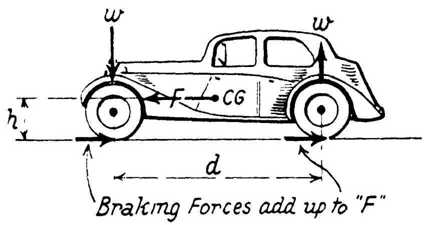

FIG. 97. BRAKING FORCES ALTER THE FRONT/REAR WEIGHT DISTRIBUTION BY TRANSFERRING A LOAD (w) FORWARDS

positive or negative change of speed. Thus, when the brakes are applied, the retarding forces act upon the car as a whole through tyre contacts at road level while an equal and opposite force, created by the disinclination of the car to lose speed, acts through the centre of gravity at a height (h, Fig. 97) which is usually 24 in. to 25 in. in cars of all sizes. By equating moments it is clear that if w represents the weight added to the front wheels (and subtracted from the rear wheels), F the retarding force, and d the wheelbase, then—

$$w = Fh/d.$$

The effect is considerable, as can be seen from an example. Take a car weighing 3000 lb. and brakes giving an emergency application of 80 per cent "efficiency." Then the total retarding force (F) is 2400 lb. and, if the ratio h/d

is 1/5, the weight transferred (w) will be 480 lb. Consequently, if the normal weight distribution front/rear is 50/50, emergency braking will increase the front-wheel load from 1500 lb. to 1980 lb. and will decrease the rear-wheel load from 1500 lb. to 1020 lb.

Now the distribution of braking effort, front/rear, can be arranged in any desired proportion (by the leverages in the system) and for the utmost effectiveness under emergency conditions, on a dry road, it should correspond with the weight distribution created by these conditions. On the other hand, it is seldom that the brakes are called upon to give decelerations exceeding 30 per cent of gravity and the transfer of weight is then much smaller; on slippery roads an even lower limit may apply. Consequently, the designer must compromise by selecting a distribution tending to give a more nearly equal rate of lining wear in the front and rear brakes under normal conditions and lessening the risk of locking the front wheels on a slippery surface. Many makers use a 55/45 distribution; a few use 60/40, and others employ 50/50.

Proprietary Braking Systems. Space will not permit of more than a brief reference to the leading proprietary systems, which will, however, suffice to illustrate some of the principles enunciated in this chapter.

The braking units of the Lockheed hydraulic system are shown in Fig. 98, the pipe lines being connected to a master cylinder (not shown) in which there is a plunger operated by the pedal. This plunger, together with the opposed pistons in each of the wheel cylinders, is sealed by cups of a rubber composition impervious to the special fluid used in the system. There is a reservoir to maintain the supply of fluid which is connected to the master cylinder. The way in which the hand brake is hooked up to the rear brake

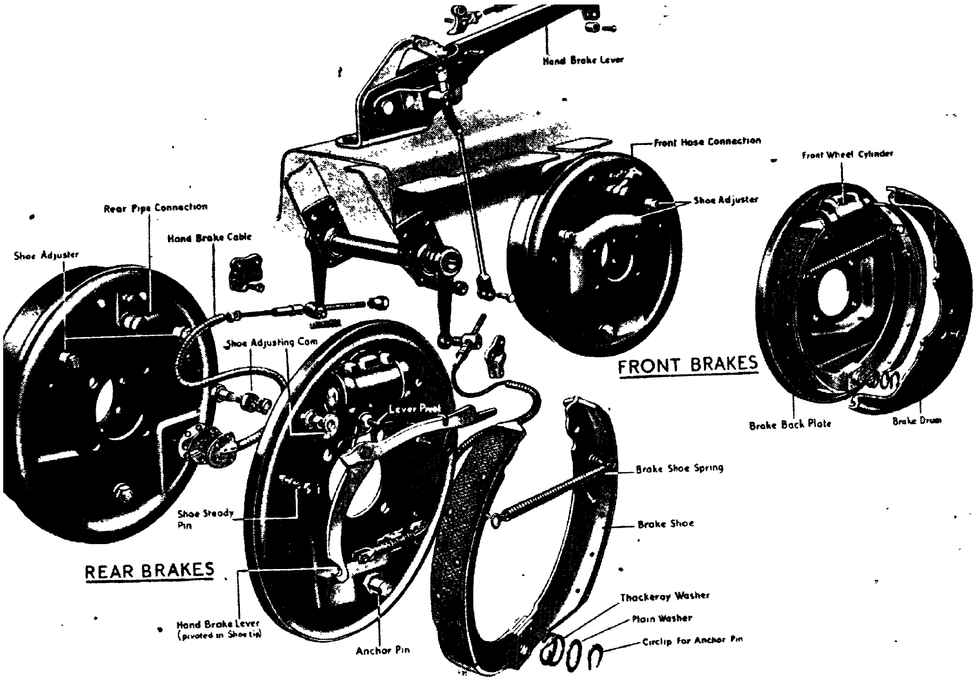

Fig. 98. General Lay-out of a Lockheed Hydraulic Braking System (Less Master Cylinder) and the Hand-brake Connexions

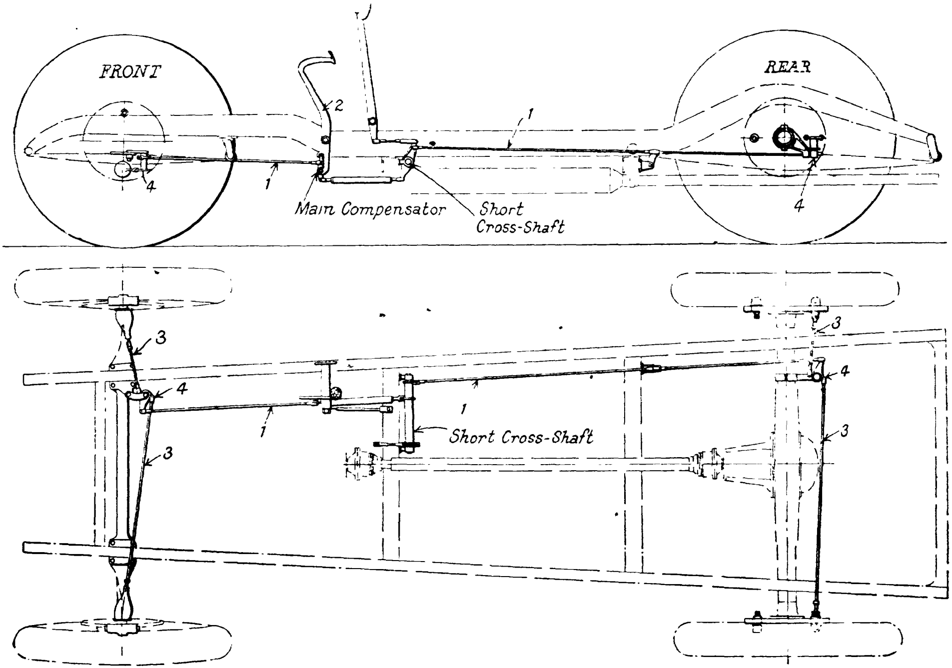

Fig. 99. Chassis Drawings of a Girling Mechanical Braking System showing Rods (1, 3) and Axle-mounted Compensators (4)

shoes will be noticed, also the "snail cams" which enable the shoe clearances to be adjusted.

In the Girling mechanical system (Fig. 99) the pedal operates shoe-expanders through a simple lay-out of tension

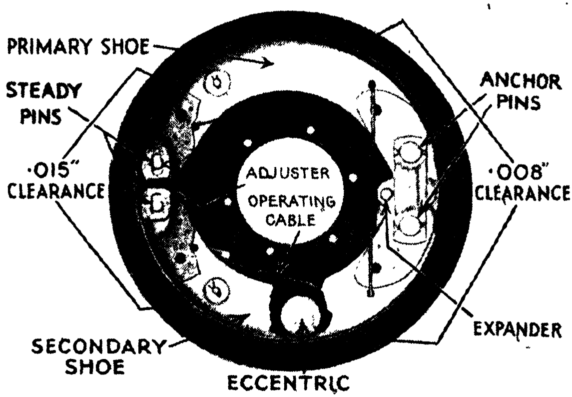

FIG. 100. COUPLED SHOES OF THE BENDIX DUO-SERVO BRAKE; FOR FORWARD TRAVEL THE DRUM TURNS ANTI-CLOCKWISE

rods and bell cranks arranged to provide compensation. Each shoe-expander consists of a cone operated by a pull rod and acting upon plungers through the medium of rollers. These plungers actuate the tips of the brake shoes. Clearances are adjusted by an expander which separates the opposite (pivoted) ends of the shoes.

The original Bendix brake provides an excellent example of the servo principle. The two shoes (Fig. 100) are connected by a link (which incorporates an adjuster) instead of being mounted upon fixed pivots. At their opposite ends the shoes are actuated by a floating expander, the mechanism being so arranged that the drag of the drum enables the primary (leading) shoe, with its self-applying characteristic to augment the load on the secondary (trailing) shoe. The anchor-pins do not restrict this movement, as they pass through slots formed in the shoe webs. The Bendix mechanism must not be confused with the more recent Bendix-Cowdrey brake, in which the servo principle is retained but a wedge-type expander is used to operate the shoes. A still later development is the "two-leading-shoe" brake, embodying an ingenious linkage between the shoes. At the time of writing the use of this device is limited to commercial vehicles.

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