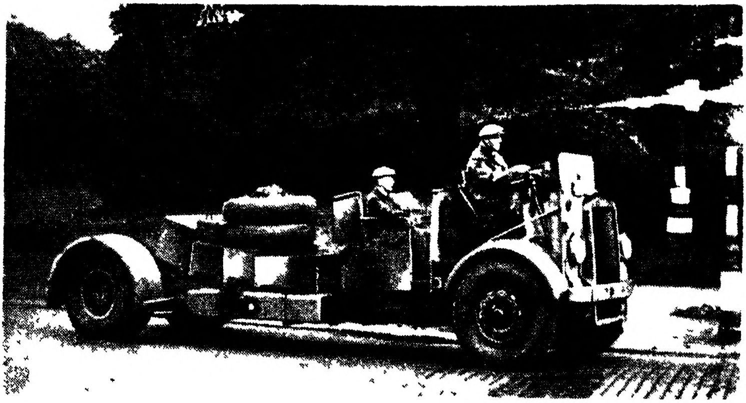
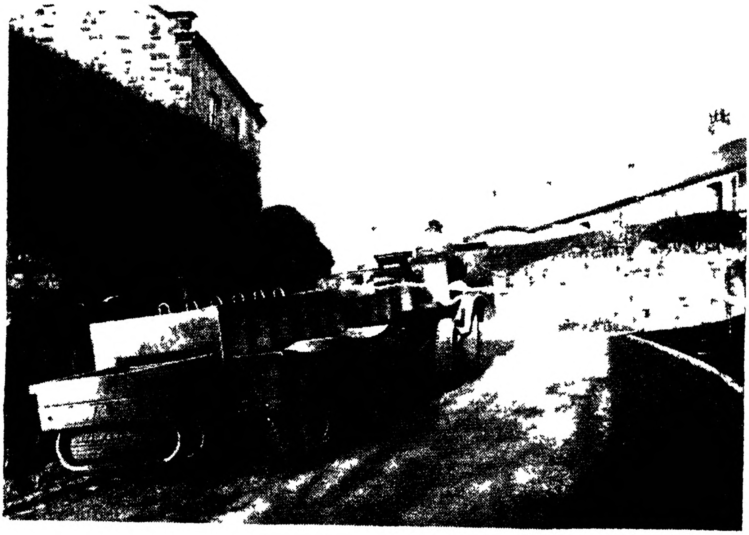
# AUTOMOBILE CHASSIS DESIGN

# ROAD TESTING



Testing of chassis under excess load, showing method of weight disposition, with test engineer in position to observe stopping distance, fuel consumption, etc., by means of up to-date measuring instruments mounted on special bulkhead



Eight-wheeled heavy goods chassis undergoing test for hill-climbing ability under over-load conditions with evenly distributed test weights, on a severe gradient of approximately one in four

# AUTOMOBILE CHASSIS DESIGN

By

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## PREFACE

THIS book is primarily designed as a text for the study of the fundamentals and technique involved in the design of an automobile. It purposely refrains from any discussion on the prime mover as there are already many excellent works presenting in all aspects both the oil and petrol engine. The information is compiled as a result of the author's work and experience at home and abroad, and, whilst some of the conclusions and inferences may be questioned, it is hoped that the technician who studies the subject in the light of research and who is not bound by accepted practice, will find little with which he is in disagreement.

Much of the subject matter is primarily evolved in connection with heavy-vehicle design, but, being of a general character, it is also applicable to the lighter field of car design. For many years the author has felt the need for a standard work which presents the subject of automobile design to the technician whose work it is to produce the vehicle in its final form and which is not entirely devoted to the study of any specialised individual unit.

If, therefore, it is considered that some sections of the book are not treated in equal detail it is hoped that sufficient information is contained therein to enable the student or worker intelligently to deal with the main components of a motor vehicle; and, if such proves the case, the purpose of the book has been fulfilled.

In conclusion, acknowledgement is due by the author to the directors of Leyland Motors, Ltd., for their kindness in permitting the use of many photographs, drawings and diagrams by which the text is illustrated.

*Southport,*
*January,* 1948.

R. DEAN-AVERNS.

## ABBREVIATIONS USED

*(As recommended by the British Standards Institution)*

| | |
|---|---|
| | Degrees |
| | Per |
| b.h.p. | Brake horse power |
| B.M. | Bending moment |
| c.g. | Centre of gravity |
| ft | Foot |
| ft. lb | Foot pounds |
| ft. lb/in$^2$ | Foot pounds per square inch |
| ft. lb/sec | Foot pounds per second |
| ft/min | Feet per minute |
| ft$^3$/min | Cubic feet per minute |
| ft/sec | Feet per second |
| ft/sec/sec | Feet per second per second |
| gal/min | Gallons per minute |
| h.p. | Horse power |
| in | Inch |
| in$^3$ | Cubic inch |
| in. lb | Inch pounds |
| lb | Pound |
| lb/ft | Pounds per foot |
| lb/in | Pounds per inch |
| lb/in$^2$ | Pounds per square inch |
| lb/in$^3$ | Pounds per cubic inch |
| lb/ton | Pounds per ton |
| m/h | Miles per hour |
| p.s.v. | Public service vehicle |
| r/min | Revolutions per minute |
| r/sec | Revolutions per second |
| S.F. | Shear force |
| sq. ft/min | Square feet per minute |
| sq. in | Square inch |
| ton/in$^2$ | Tons per square inch |

# AUTOMOBILE CHASSIS DESIGN

## CHAPTER 1

## VEHICLE PERFORMANCE

IN making calculations of vehicle performance, it should be realised that there are many sources from which potential inaccuracies in basic data might emanate. The effects are fortunately such that the results from general formulae are satisfactory for all practical purposes, the certain refinements in calculations being more often of academic interest.

We do not take into account gyroscopic effect of large tyres and wheels, although such effects are briefly referred to later, nor the effect of inertia of rotating parts upon vehicle acceleration or deceleration. There are many such points which may eventually form the basis for adjustments to final calculations. Certain preliminary calculations are necessary before any attempt is made to lay the chassis down in the design office for scheme and detailed treatment. These are in the main connected with the power available to overcome resistance to motion forces to a degree which produces acceleration, together with the forces which are required to produce deceleration or retardation.

It is assumed that sufficient data is available regarding the desired performance to enable the vehicle essentials to be evaluated. First, estimations of vehicle gross weight, either by comparative data from known models, or, in the case of a heavy goods or public service vehicle, by adopting the maximum legal gross weight allowable. The maximum vehicle speed and the climbable gradient must also be known. From these three conditions the power necessary for propulsion, the overall gear ratio, and the gearbox ratios may be determined. As a rough preliminary for the heavier-type service vehicle power at the rate of 7–8 b.h.p. per ton minimum forms a useful basis for check purposes. In the private-car class, factors other than essential power control engine specification. The following comparisons show the wide range of power-weight ratio variation, so that no reliable constants can be laid down for private-car design—

BRITISH PRACTICE

| | | | |
|---|---|---|---|
| Car No. 1— | 23 h.p. at 4,000 r/min | gross weight | 1,800 lb |
| 2— | 32 h.p. at 4,000 r/min | ,, ,, | 2,100 lb |
| 3— | 52 h.p. at 4,000 r/min | ,, ,, | 3,000 lb |
| 4— | 65 h.p. at 3,800 r/min | ,, ,, | 3,250 lb |
| 5— | 100 h.p. at 3,800 r/min | ,, ,, | 3,400 lb |

Compare these figures with—

Car No. 1— 78 h.p. at 3,500 r/min gross weight 3,000 lb
2—132 h.p. at 3,400 r/min ,, ,, 3,500 lb
3—175 h.p. at 3,400 r/min ,, ,, 5,700 lb

In the assessment of the more detailed performance it would appear necessary to consider this from two aspects: The first as emanating from power functions such as the torque or driving effort at the road wheels, which is the result of preliminary calculations; and the second from the aspect of efficiency of assembly and general layout, typified for instance in the correct layout of steering in combination perhaps with either suspension or braking—the front axle and the selection of correct wheel layout.

The first results in vehicle movement, whilst the second supplies the refinements of movement. In this chapter, therefore, it may sometimes be found necessary to discuss detail before an appreciation is possible of the true problems involved, as the solutions are oft times far more elusive, but, with regard to the first aspect, most of the assessment may be made from empirical formulae and known data, which at all events permits performance to be gauged from a comparative view point. Consider first the category concerned with vehicle movement.

## VEHICLE MOVEMENT

The mechanics of a moving vehicle are confined primarily to simple calculations from accepted formulae. In considering a wheeled vehicle three main factors are concerned—

(a) Rolling and frictional resistance.
(b) Gradient resistance.
(c) Air or wind resistance.

Rolling resistance varies considerably with the type of road surface as indicated in the following table—

*Table* 1

| Road Surface | Resistance lb/ton | Road Surface | Resistance lb/ton |
|---|---|---|---|
| Railroad | 10 | Soft Macadam | 97 |
| Good Asphalt | 15 | Well-rolled Gravel | 57 |
| Medium Asphalt | 22 | Small Cobbles | 60 |
| Poor Asphalt | 29 | Medium Cobbles | 130 |
| Wood Paving | 30 | Large Cobbles | 240 |
| Granite Sets | 35 | Hard Dry Clay | 100 |
| Best Macadam | 45 | Sand Road | 360 |
| Ordinary Macadam | 50–60 | Loose Sand | 560 |

An average figure which appears to give a little in hand for all general purposes is 50 lb/ton. This, of course, does not apply to specialised military or agricultural vehicles. It is interesting to note that rolling resistance for cord tyres is approximately 33 per cent less than that for fabric tyres, and the figures are practically constant for speeds of 20 to 50 m/h.

Frictional resistance is another variable factor. It includes resistance to motion through transmission losses, such as gear efficiencies, oil churning, tyre adhesion and many other influences. A useful general approximation is—

$$f = 30 + 0{\cdot}012\ W$$

where f = frictional resistance in lb
W = total weight of vehicle.

Transmission losses are usually estimated at 10 per cent in direct gear and 15 to 20 per cent in low gear, so that in calculating performance, tractive effort should be estimated at efficiencies of 90 per cent top gear and 85–80 per cent in lower gears, as an average. This figure includes such losses as occur in oil churning in a closely-designed gearbox and confined rear axle. It also takes into account temperature losses. For private cars these figures are low, 95 to 89 per cent being the usual figures assumed. The friction loss in the tyres is about two-thirds that of the total chassis loss.

## GRADIENT RESISTANCE

Gradient-resistance figures must be added to those for rolling resistances in compiling the summation of total resistance to motion. Gradient resistance is a function of vehicle weight and gradient, and it does not depend on vehicle speed. It may be expressed as $P = \frac{W}{G}$, where G = gradient and W = weight of vehicle or W sin $\theta$ = P if the gradient is expressed in angular dimensions.

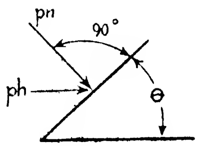


Fig. 1

Wind or air resistance is dependent upon speed and is calculated from the formula—

$$pn = ph\ \frac{2 \sin\theta}{1 + \sin^2\theta}$$

The value varies according to the amount of streamlining effected or angle of inclination of the surface to normal, for instance for a streamlined car a constant of 0·0017, or for a single-deck streamlined coach body 0·0021, may be sufficiently accurate for all practical purposes: 0·0032 is used in connection with double-deck passenger vehicles. These constants form part of the expression ph = $kV^2A$, where V is velocity in ft/sec and A the projected area in sq. ft. Note should be made of the direction of the wind, whether head on or following, as obviously this affects the relative velocity as opposed to the surface (see Fig. 1).

## TRACTIVE RESISTANCE

The sum of these three resistances to motion is termed the tractive resistance of the vehicle and may be expressed by the equation—

$TR = W\left(R + \frac{2,240}{G}\right) + KV^2A$ where TR = tractive resistance

W = vehicle weight in tons
R = rolling resistance lb/ton
G = gradient
V = velocity in ft/sec
A = projected area in sq. ft.

Examination of the power curves of the engine reveals that whereas the torque curve rises to a point representing 1,150 engine r/min and gradually falls with further increase in engine speed, the b.h.p. curve continues to rise throughout the range of engine speed: b.h.p. for the purpose of actual work calculation is of little value, and merely represents the rate at which engine is performing work. It becomes, however, a necessary figure for speed calculations. Torque represents the actual useful work performed, and its value is consequently applicable to work calculations as exemplified in gradient climbing ability (see Fig. 2).

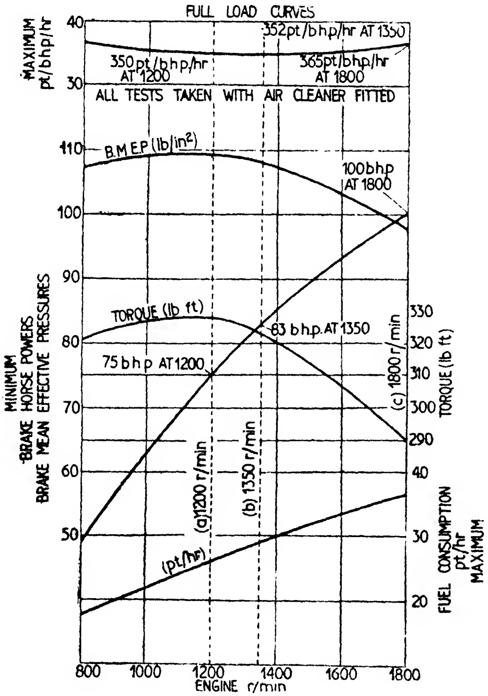



**Fig. 2—Engine performance curves**

In explanation of this it should be noted that torque signifies a turning moment, that is a force acting at a lever arm and is expressed in lb/ft, whilst power is made up of torque and angular velocity. Therefore $\frac{\text{Work}}{\text{Time}}$ = Power, which is the rate of doing work, hence the product of torque and angular velocity is power. It is sometimes necessary to calculate the torque from the horse power and speed of engine revolutions, in which case

$T = \dfrac{33,000 \text{ h.p.}}{6{\cdot}28\text{ n}}$ or $5,250 \times \dfrac{\text{h.p.}}{\text{n}}$ lb/ft, n being the number of engine revolutions per minute.

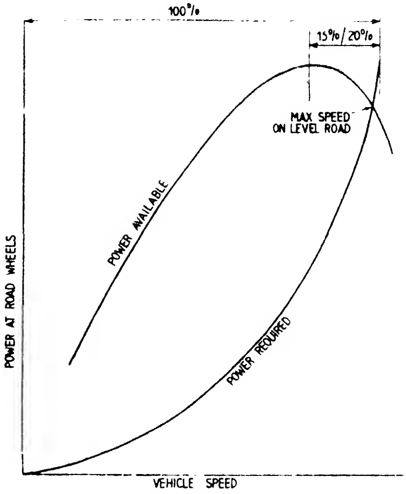



**Fig. 3**

## TRACTIVE EFFORT

To overcome the external forces, effort must be applied in the form of a torque at the road wheels, and the first step is to ascertain what overall gear ratio, that is low gearbox and axle ratio, is necessary to convert the available engine power to useful work.

If e = transmission efficiency (see frictional resistance)
r = overall gear ratio in low gear
D = running radius of tyre
T = engine torque in lb/ft

then tractive effort at wheels is found from the formula

$$T_E = \frac{T \times r \times e}{\frac{D}{12}}.$$

As $T_E$ must at least equal $T_R$

$$\text{then } \frac{T \times e \times r}{\frac{D}{12}} = W\left(R + \frac{2,240}{G}\right) + K V^2 A$$

$$\text{and } r = \frac{\left(W\left(R + \frac{2,240}{G}\right) + K V^2 A\right)\frac{D}{12}}{T \times e}.$$

The extremes in conditions both of the forces opposing motion and those which are required under certain circumstances to overcome such resistances have been established. It now becomes necessary to ascertain requirements for the desired performance in intermediate stages.

Knowing the engine characteristics, it is a comparatively simple matter to arrange the intermediate gearbox speeds. It should be remembered that it is advisable that the peak of power at the road wheels should occur at say 15–20 per cent early on the engine power curve, that is to say the gear ratio is such that the peak occurs at a speed of 15–20 per cent lower than the ultimate speed which is given when the peak is projected on to the power required curve (see Fig. 3). It is desired that the number of speed changes should ensure that the drop in engine speed and consequent vehicle speed is not too great when making the gear shifts, whilst at the same time permitting ease of gear change by the driver.

The main function of the gearbox is to maintain engine speed at the most economical value under all conditions of vehicle motion, so that the optimum value of power output/fuel consumption is achieved. This however is not easy to accomplish with the ordinary gear-type reduction; the ideal would be an infinitely variable transmission. It is known, of course, that the larger the step between two gears the more difficult is the change. The selection of correct ratios is therefore important.

## GEOMETRICAL PROGRESSION

Geometrical progression affords a selection which has many merits, since the vehicle is propelled in gear by a series of engine accelerations and decelerations. If the ratios are in geometrical progression, then the engine speed range is constant throughout all gears. This, of course, is the theoretical ideal.

In practice, however, many other features enter into the final gear-ratio selection; for instance, low-speed gear is often an emergency one, whilst the change down must frequently be quickly effected. Such conditions would be assisted by reducing the speed range of the engine between the preceding ratio and low gear in order that the loss of vehicle speed should be a minimum and the waiting period for synchronisation of the two gear wheel speeds be reduced.

Fig. 4 illustrates a series of curves of engine speed/vehicle speed of gear ratios in geometric progression.

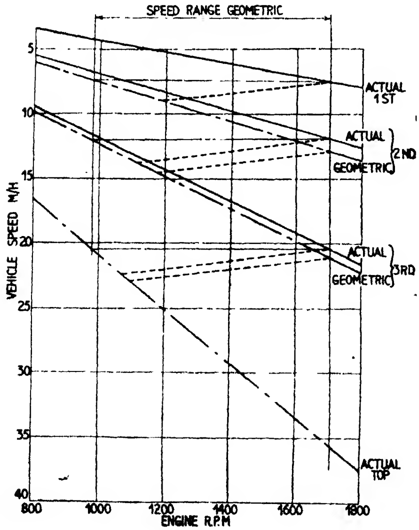



**Fig. 4—Vehicle performance (gears in geometric progression)**

In the example it has been assumed that in order to satisfy known conditions of performance an overall ratio of 25·6 to 1 is necessary. The selected axle ratio to give the desired direct- or top-gear performance is 5·4 to 1. If the gearbox is of the four forward speed type then the low ratio will be 4·74 to 1, direct or top 1 to 1, and the intermediate ratios for geometric progression will be 1·68 to 1 and 2·82 to 1. These are the ratios used in the curves indicated.

Incidentally geometric progression represents a series of quantities in which each term is obtained by multiplying the preceding term by some constant factor called the common ratio, for example, 1, 3, 9, 27, 81, etc., each term being three times that preceding it. For a series of terms $n$ having common ratio $r$ and the first term $a$ the sum

$$S = \frac{a\,(r^n - 1)}{(r - 1)}$$

or, if $r$ is less than 1, a more convenient expression is

$$S = \frac{a\,(1 - r^n)}{(1 - r)}.$$

It is further assumed that the engine is governed to a maximum speed of 1,800 r/min and it is therefore not advisable to allow engine revolutions to reach this figure before changing gear, as power will be cut off suddenly and the vehicle will lose considerable road speed. Moreover, maximum engine torque is developed far below maximum engine revs, and in low gear it is desirable to utilise this power when climbing steep gradients. The engine-speed range permitted by the gear ratios should therefore reach its maximum just above the speed for maximum torque.

From the diagram it will be seen that in first gear the vehicle accelerates from 4¼ m/h at 980 r/min to 7½ m/h at 1,800 r/min on the level road, and the speed range of engine is *constant* over 720 r/min. The gear change is made when the engine revs fall to 980 r/min, from which the next acceleration is to 13 m/h through third gear to 21 m/h, and on to top or direct at 37½ m/h governed speed.

These ratios may need finally adjusting, due to extraneous features, after initial test.

Incidentally, a certain amount of clutch slip is often experienced, when re-engaging takes place, more particularly in the lower gears, with higher engine speed and lower vehicle speed. The effect of this is illustrated by dotted lines on Fig. 4, noting that the vehicle speed has increased over that in the higher gear even before the drive has become positive.

The selection of intermediate gears permits the plotting of tractive-effort curves against road speeds, in each of the four gears selected (see Fig. 6). Previously road speeds have been related to engine speed on the level ground. If the ideal tractive effort curve be plotted on the same ordinates, that is through the point on each of

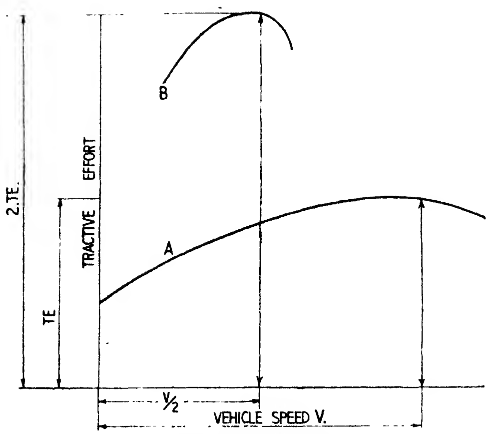



Fig. 5—Ideal tractive-effort curve

the four curves corresponding to the selected engine speed, which indicates the most economical point of working on the engine torque curve, it will be seen that there is a portion of each individual effort curve overlapping the ideal. This means that the gear ratios selected provide in each case for the maximum pulling power at any vehicle speed. The ideal tractive-effort curve (Fig. 5) is a rectangular hyperbola based on ordinates of vehicle speed and tractive effort.

For example, if $T_f$ = tractive force, and
$V$ = vehicle speed ft/min

then from the formula $\dfrac{T_f \times V}{33{,}000}$ = b.h.p. all the data for plotting such a curve can be ascertained.

If, say, 1,000 lb is the tractive force required at the road wheels, and the vehicle speed is 30 m/h or 2,640 ft/min, then—

$\dfrac{1{,}000 \times 2{,}640}{33{,}000}$ = 80 b.h.p. is the power required at the driving wheel to produce the desired vehicle speed.

If the vehicle speed is to be 15 m/h, then the tractive force will be 2,000 lb and for 7½ m/h a corresponding proportional increase.

## GRADIENT PERFORMANCE AND ACCELERATION

We can now consider the gradient performance, and investigate the accelerating properties of the vehicle. Incidentally, it is of course appreciated that if a gearbox ratio is modified so that the total overall ratio between road wheel and engine is double its original ratio, curve A becomes curve B (see Fig. 5), all horizontal dimensions being halved and all vertical distances doubled, as for given engine speeds and doubled total ratio the vehicle speed is halved but the tractive effort is doubled.

Using the same scale as for the tractive effort, we can calculate the total tractive resistance at different gradients. Referring to Fig. 6

it will be seen that the curves obtained cut the tractive efforts at various points.

Take, for example, top-gear tractive effort on no gradient represented by the rolling and air resistance line. At a vehicle speed of 20 m/h there is ample excess effort available, which gradually decreases until at 31 m/h the two curves intersect, showing that the margin of surplus power has been reduced to zero. In other words 31 m/h is the theoretical maximum speed of the vehicle in top gear.

This is not strictly the case for, as will be noted from the combined resistance curves, wind resistance is the main cause—the estimated pressure is of course for the worst possible condition and for ordinary work might be greatly in excess of average. In third gear tractive effort considerably exceeds resistance on the level and the vehicle will use this effort to accelerate from 10 m/h to 27 m/h, at which speed a change to top gear would be made. If, however, a gradient of 1 in 32 is encountered in third gear, the vehicle does not possess so much surplus power, but can still accelerate from 10 m/h to 18½ m/h.

Fig. 6 also shows the vehicle speed at which the best pulling power of the engine is exerted in each gear. It is read from each tractive-effort curve at the point of intersection with the ideal tractive-effort curve.

Top gear 21 m/h surplus effort over rolling and air resistance of 250 lb

Third gear 13½ m/h surplus effort over rolling and air resistance of 950 lb

Second gear 8¼ m/h surplus effort over rolling and air resistance of 2,200 lb

First gear 5 m/h surplus effort over rolling and air resistance of 3,950 lb.

The climbable gradient at each of these optimum points is easily estimated from the formula—

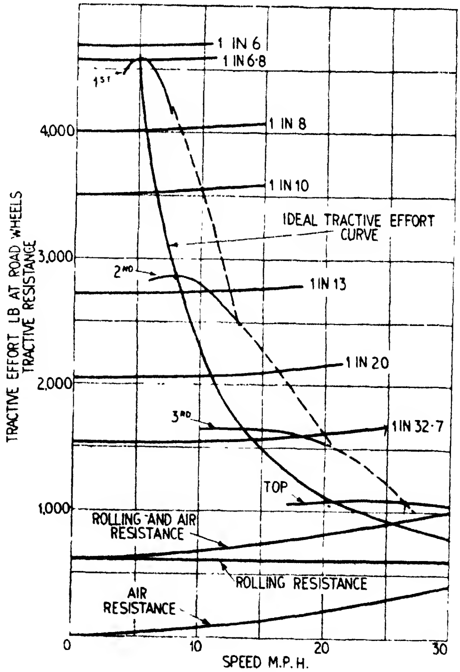



**Fig. 6—Vehicle gradient performance in each gear**

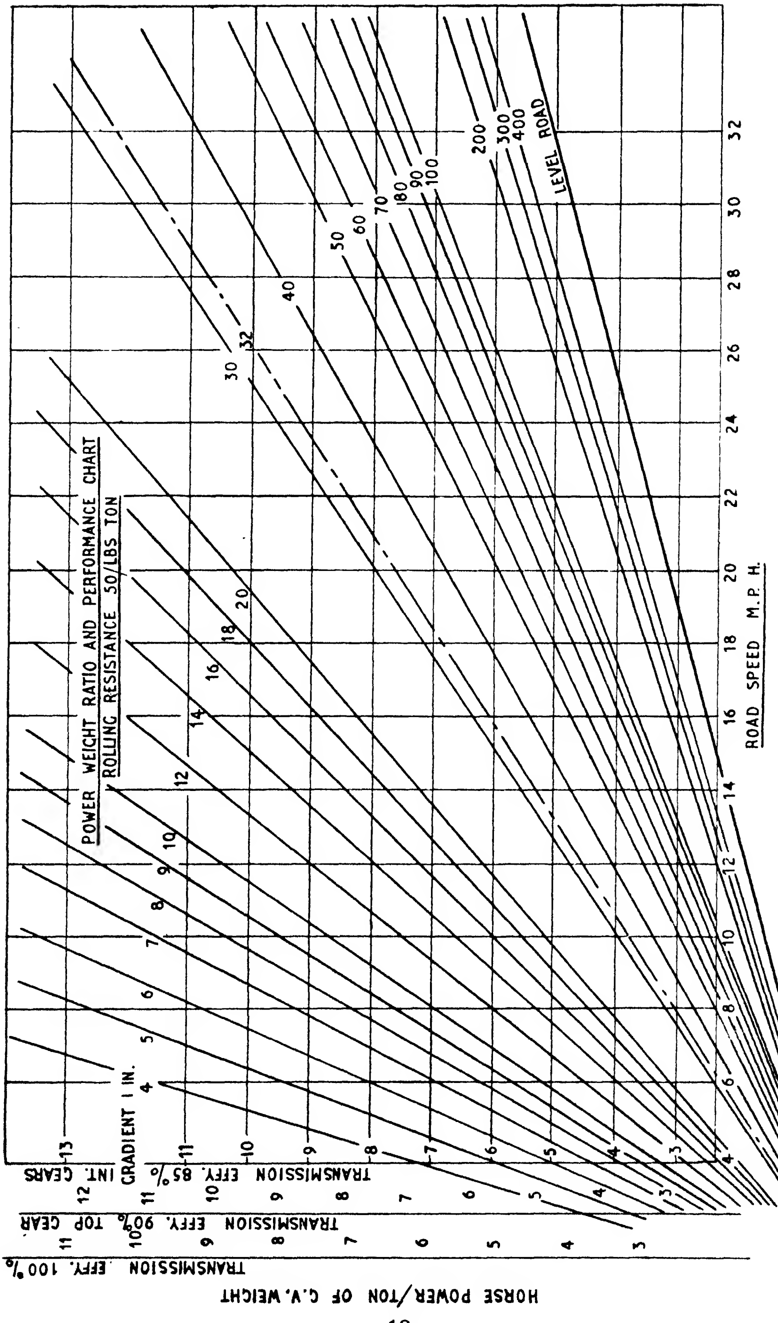



Fig. 7—Vehicle gradient performance chart
(Power/weight——Road speed)

$$P = \frac{W}{G} \text{ or } G = \frac{W}{P}, \text{ where } G = \text{gradient}, \; W = \text{vehicle weight}, \; P = \text{excess power.}$$

For the purpose of ascertaining acceleration characteristics of the vehicle it must be borne in mind that the limiting factor is the adhesion between the tyre and road, for if excess tractive effort is put through the wheels, the latter will spin and adhesion will be lost. The coefficient of adhesion varies and depends upon the surface of the road, and to some extent upon the rear-axle weight. If we divide the driving force at the wheels by the vehicle weight we arrive at a factor $K = \frac{T_E}{W}$. $T_E$ of course varies in each gear and is a maximum at the point of maximum torque on the engine-performance curve. This factor may be used in calculating vehicle acceleration thus—

If $T_E$ = tractive effort in lb/in
$W$ = mass of vehicle lb
$f$ = acceleration (32·2 ft/sec/sec)

$$T_E = mf \text{ or numerically } T_E = \frac{W}{32{\cdot}2} f$$

therefore $f = \frac{32{\cdot}2 T_E}{W}$, but $K = \frac{T_E}{W}$ and substituting we get

$$f = 32{\cdot}2 \; K \text{ ft/sec/sec.}$$

This expression, however, bears no relation to the coefficient of adhesion between tyre and road, and deals only with vehicle gross weight. It has been stated that the maximum effort which can be exerted at the ground without wheel skid will depend upon the weight upon the rear axle × coefficient of adhesion, hence $T_E = \omega\mu$ where $\omega$ = axle weight gross and $\mu$ = coefficient of adhesion (usually 0·7) for normal surfaces,

by substitution $\omega\mu = \frac{Wf}{32{\cdot}2}$ and $f = \frac{32{\cdot}2\omega\mu}{W}$.

## STABILITY AND CENTRE OF GRAVITY

We have dealt briefly with the mechanics of a vehicle moving in a straight line. There are other factors to be considered in assessing its performance: location of the centre of gravity (c.g.) with relation to cornering, its steering and road-holding capacity, braking efficiency, and the effects of braking upon steering. However, these factors may be discussed briefly under their respective sections, as they do not directly affect calculations made in the initial design stage of a vehicle, which primarily are estimates regarding power available and power required. One point, nevertheless, should be noted and concerns the centre of gravity of the complete vehicle. It is important that this centre should not be incorrectly disposed, and a few notes here emphasising that fact may not be out of place at this stage. The height of the c.g. from ground affects its stability when rounding curves. The vehicle is acted upon by a centrifugal force, which is denoted by P (see Fig. 8).

$$P = \frac{Wv^2}{gr} \text{ and } P\frac{h}{B} = \frac{W}{2} \text{ or } \frac{2Ph}{B} = W$$

The centrifugal force applied when the vehicle overturns is given by the formula

$\frac{Wv^2}{gr} = \frac{WB}{2h}$ from which the maximum velocity is obtained

$$V = \sqrt{\frac{32{\cdot}2\ rB}{2h}}$$

where r = radius of bend (ft)
V = velocity (ft/sec)
W = weight of the vehicle.

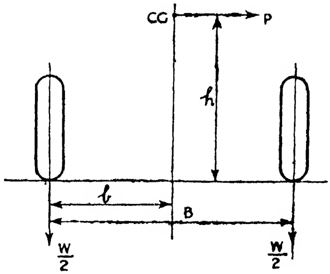



Fig. 8

Certain other known features should not be lost to sight in connection with the c.g. location as related to brakes, which affect the motion of a four-wheeled vehicle—

(1) There is a considerable falling off in the effectiveness of rear brakes as the centre of gravity moves forward and a similar though smaller effect with front brakes when c.g. is moved to the rear.
(2) With combinations of both rear and front brakes, the variations in stopping distance caused by horizontal movement of the c.g. is comparatively small.
(3) There is an optimum position of c.g. wherein front and rear brakes are equally effective so far as stopping distance is concerned.

So far as deviation from line of travel is concerned—

(1) Locked rear wheels tend to produce deviation whether used alone or in combination with other brakes.
    (a) The tendency becomes more marked as c.g. is moved forward.
    (b) As opposed to the effect of rear brakes in producing deviation, the steadying effect of front brakes is marked.
    (c) With the forward c.g. and the front brakes in use, the rear brakes have only a slight effect on deviation as well as on stopping distances.
    (d) To ensure minimum deviation whatever the braking combination, the c.g. must be near the rear wheels.
    (e) *The optimum position for c.g.* appears to be one-third wheelbase from rear-axle centre line.
(2) *The height of the centre of gravity* has but little effect on stopping distance. The effect of increased height is felt more

**Fig. 9—Tilting test of double-decker bus. Note body angle and chassis angle due to spring deflection**

with rear-wheel brakes only, probably due to the reduction of rear axle load when braking.

(3) The most marked effect of *transverse movement of c.g.* is in diagonal braking. When the right-hand front and left-hand rear wheels are locked, moving the c.g. so as to throw weight on to the right-hand wheels causes stopping distance to diminish to the extent of 15 per cent. With left-hand front and right-hand rear effect on stopping distance is much less.

Where three wheels are locked, the effect of moving the load away from the free wheel is to reduce the stopping distance, probably because a more heavily loaded wheel produces a greater retarding force when locked. Similarly when the c.g. is moved towards the freely running wheel, stopping distance increases.

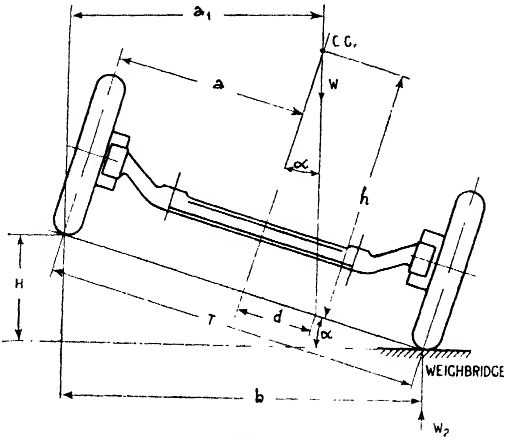



Fig. 10

*To find the Centre of Gravity of a Vehicle*

Data—

W = total weight of vehicle

$W_1$ = weight on near-side wheels, vehicle on level ground

$W_2$ = weight on near-side wheels, when vehicle is tilted to a known angle.

Referring to Fig. 10—

$$h = \frac{d}{\tan \alpha}$$

where $\alpha = \sin^{-1} \frac{H}{T}$.

$$a_1 = \frac{bW_2}{W}$$

where $b = T \cos \alpha$ substituting, $a_1 = \frac{T \cos \alpha \, W_2}{W}$.

$$d = \frac{a_1}{\cos \alpha} - a, \text{ substituting for } a_1, \; d = \frac{T \cos \alpha \, W_2}{\cos \alpha} - \frac{TW_1}{W}$$

where $a = \frac{TW_1}{W}$.

Therefore $d = \frac{T}{W}(W_2 - W_1)$

$$h = \frac{T}{W \tan \alpha}(W_2 - W_1)$$

## VEHICLE FLOTATION

Cross-country operation presents many differing problems as provision is necessary against sinkage on all types of terrain. Correct flotation of the vehicle in mud, clay, sand and snow demands consideration of the most suitable type of vehicle, the ground contact area of the tyre, its loading, and, above all, traction must be given first consideration.

It should be appreciated that correct flotation on one type of terrain is not necessarily correct for another, neither does it follow that adequate traction on one surface will be successful on one of dissimilar characteristics. For instance, snow differs fundamentally from sand, and sand can hardly be compared with mud. Snow is compressible, although the amount of penetration by the tyre depends upon the tyre load and the consistency of the snow. A ground pressure of about 2 or 3 lb/in$^2$ is all that can be allowed if the tyre is to compress a bearing surface for supporting the load without sinking too deeply and stalling through lack of ground clearance.

Sand, on the other hand (except when wet), does not compress so readily. It resembles very minute round shot and the particles are displaced upon contact with the tyre so that some roll to each side of the tyre whilst the remainder immediately under the tyre compacts slightly. When traversing such soft sand tyre pressures are of paramount importance, as upon pressure depends ground-contact area and thus intensity of loading, which if much in excess of 10 lb/in$^2$ will probably permit sinkage to a depth at which, although the engine power is sufficient to spin the wheels, forward traction ceases, due to the build up in front of the driven wheels with the consequent increase in rolling and gradient resistance.

Sand operation generally requires a tyre size with rating for road operation of 75 per cent more than that required for load carrying. The tread design is not of great importance except in so far as it effects tyre flexibility. A tyre should be chosen having the minimum number of plies and should be used as single equipment. Twin tyres on a driving axle considerably reduce sand performance, which from the author's experience may measure 30 per cent reduction.

Reverting to inflation pressures, tractive ability is improved by reduced pressure whilst rolling resistance is reduced. Reference to Fig. 11 illustrates the effect upon ground-contact area under reduced inflation pressures, whilst it will be noted that the lower pressure avoids the wedge action of the tyre and allows the tread to become concave, trapping the sand to improve adhesion and sustain the full amount of what compressibility the sand possesses.

Reference to Fig. 12 shows how both tractive effort and rolling resistance are affected by this question of tyre pressures. The variations as shown in both cases are true for all tyre sizes and it should be noted that as deflection increases with decreased pressure, tractive effort increased rapidly, whilst rolling resistance approaches the minimum until any further deflation takes place when it results in work being done on the tyre itself and thus the curve begins to rise rapidly. The tyre size chosen is greatly in excess of that required for road work with similar loads and has a pressure rating of 50 lb for normal work.

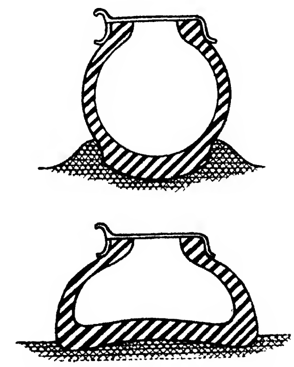


**Fig. 11—Effect of reduced inflation pressure**

In the particular case and for comparative purposes, the following percentage deflections on hard roads correspond to the actual tyre inflation pressures—

50 lb pressure equals 25 per cent deflection
33 lb pressure equals 30 per cent deflection
22 lb pressure equals 40 per cent deflection
14 lb pressure equals 50 per cent deflection.

Another comparison of actual vehicle performance will illustrate the effect of tyre pressures. A vehicle fully laden to tyre capacity for road work and fitted with 10·00 × 20 single tyres and inflation pressure of, say, 70 lb would hardly be capable of movement in sand dunes—certainly not able to climb any gradient. In other words its flotation is practically nil. Should the tyre pressures be reduced, say to 40 per cent of their rated pressures, everything else remaining unaltered, it is found that the vehicle has a moderate sand performance and in the example in mind a 14 per cent gradient could be climbed with a trailer tow load of 2,500 lb.

If, however, tyre equipment be revised and 14·00 × 20 tyres fitted, having a rated load twice that of the 10·00 × 20 (that means that the tyres are only 50 per cent laden), a further modification to pressures is permissible and the tyres may be inflated to 40 per cent their rated load, resulting in a similar reduction to that with the 10·00—which in this case is 15 lb—then the vehicle will be capable of climbing a gradient of 30 per cent with a corresponding increase in tractive effort to the extent of that necessary to trail a 5,000 lb load.

Having briefly considered tyre equipment, it remains to decide upon the number of driven wheels desirable. For soft terrain the more driven wheels provided the better will be the vehicle performance. This may be illustrated to greater effect if a very heavy vehicle is considered, as gross laden weight of the lighter types permits normal sand duties to be carried out by tyre adjustment, although it must not be overlooked that the advantages gained on the larger vehicle are equally applicable to the light weight. In the vehicles in mind the gross laden weights were 40 tons distributed evenly over all tyres, the measured rolling resistance of the front wheel drive vehicle was 11,100 lb, 24 × 32 tyres, inflation pressure 50 lb. This vehicle was capable of movement on the level with a drawbar pull of 2,500 lb, but its gradient performance left much to be desired.

The front driving wheels in this case had to overcome not only the rolling resistance of the trailing axle but in order to exert the 2,500 lb drawbar pull, they had to be capable of producing a tractive effort of 8,050 lb (that is 2,500 + 5,550).

When four-wheel drive had been fitted the front drive still exerted its effort of 8,050 lb and two driving axles would produce 16,100 lb, whilst part of the rolling resistance of the rear driving wheels would be eliminated since they become driven. The vehicle then climbed a gradient of 15 per cent and could produce a drawbar pull in the region of 16,000 lb on level sand.

Further reduction in inflation pressure from 50 lb to 20 lb in all tyres increased the drawbar pull to 20,000 lb.

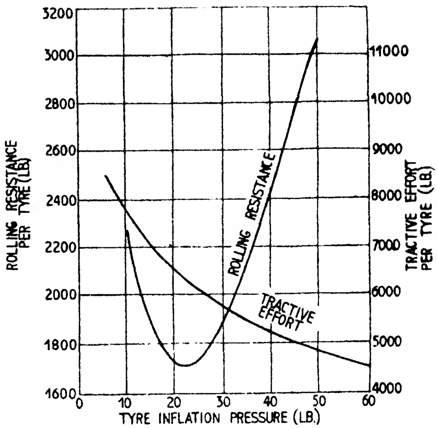



Fig. 12—Effect of tyre pressure on performance

Flotation in mud may possibly not be accomplished merely by tyre adjustments as mud differs somewhat from sand since it does not possess a great degree of compressibility, and a tyre passing over such a surface will sink until it reaches a solid bottom and sufficient contact area to give load support. Generally a mud surface has a hard subsoil underneath, but in marshy districts it sometimes consists of top growth, underneath which is an almost non-negotiable morass. In this latter case traction is almost impossible except with the multi-wheeled drive and low ground pressures.

Mud generally is more easily displaced than sand, hence it is often found that even with reduced tyre pressures the tyre does not become deformed and thus it loses many of the advantages of deflation. Much research has been carried out on suitable tread design, but if the ground is adhesive chains or the like are very necessary, and ground clearance is of primary importance. The use of a large *diameter* tyre is preferable to a tyre of large section since the arc of sinkage is longer with the large diameter than that of the smaller but larger-section tyre, or alternatively to obtain the necessary contact area the smaller tyre must sink deeper.

Traction in snow is equally difficult and variable, so variable in fact that a tyre designed with a suitable tread pattern may give excellent results in snow at temperatures of 0° F, whereas in temperatures nearer to freezing point tractive ability almost disappears, for under pressure the snow forms ice and the tread grip is lost: hence the use of chains in snow. As in the case of mud, a larger diameter tyre is necessary, large enough to produce a ground pressure of 2–3 lb/in$^2$ on hard ground. Remember that when the tyre sinks, as in the case of mud, the flanks of the tyre do not contribute to any appreciable extent area which in practice reduces intensity of ground pressure. It is important to consider that as any type of terrain where the coefficient of traction is low or where penetration or sinkage is above normal, the performance of the vehicle is limited by the amount of work the tyres can convert into tractive effort; thus for sand, snow and mud operation all-wheel drive is necessary as, regardless of engine power, forward movement may be impossible if all tyres do not contribute their share of tractive effort.

## ROLLING RESISTANCE

The standard or generally accepted figures for rolling resistance are, of course, applicable to vehicles moving on made roads, but the problem of ascertaining rolling resistance applicable to specialised vehicles, as instanced in the four-wheel drive gun tractor for service overseas, assumes a much more formidable aspect. The problem of placing the wheel on the ground so that the surface may not be displaced sufficiently to stall the vehicle completely is one which cannot be fully dealt with in these notes. The theory is based upon wheel sinkage, and as a guide to heavy-vehicle designers the following observations will be of assistance.

There are certain limitations with regard to type of vehicle for functioning on different types of terrain, for instance the multi-wheeled low-pressure tyred vehicle is probably superior to the track-laying vehicle for certain surfaces, and vice versa under opposite conditions. Success or otherwise in performance is contributed to by careful consideration of rolling resistance, which in turn may be affected by one or more of the following points—

(a) Input to axle shall be balanced and not exceeded at the ground no matter what coefficient of adhesion is encountered; the engine power and axle ratio to be so arranged as to provide tractive effort at the wheels not in excess of the requirements of gross weight tractive adhesion and gradient.

(b) A distribution of axle weights which will prevent stalling or alternatively wheel spin when transference of engine torque from one axle to the other is occasioned by various adhesion factors as between one axle and the other.

(c) A proportion between track and wheelbase which will permit transverse articulation of one wheel without unduly reducing the load on the wheels of the other axle through excessive frame rigidity.

All the foregoing points, however, cannot be treated individually, those concerning torque input, tractive output and axle loading being entirely bound one to the other. In applying the theory that input to wheels must not exceed required output (bearing in mind coefficients of adhesion) it is realised that a compromise must be found for effort required on gradients having soft surfaces. It is believed that engine torque varies continually between the two driving axles in direct proportion to the weight of the individual axles and the ground surface coefficient; for instance, if the front and rear axles are both similarly laden, but two wheels of one axle are on soft ground whilst the other axle is traversing a hard surface, and, say, the adhesion factors are 0·3 and 0·6 respectively, then the rear axle will assume twice the effort of that in the front axle. This is accounted for by the fact that the product of tractive effort and adhesion balances according to axle load, up to a point where tractive effort exceeds adhesion factors when the wheels will spin.

A further illustration could be shown if the front-axle adhesion is reduced to zero or the axle lifted clear of the ground through obstruction, all the torque being then utilised by the rear axle; for, although the engine is producing maximum output, the only work done is that at the rear, since the forward axle is offering no opposite reaction at the ground. It is this reaction which allows the available torque to be converted to useful work, so that the front axle absorbs only sufficient energy to move rotating parts.

The basic fact to bear in mind is that—

$$\text{Coefficient of Tractive Adhesion} = \frac{\text{Propulsive thrust}}{\text{Driving axle weight}}.$$

If, therefore, both axles are on similar surfaces, then weight distribution is of little importance, but the one condition which makes careful balance of weight a necessity is that which appertains when the vehicle is negotiating a gradient demanding maximum torque output, the vehicle climbing the hill and not descending, whilst the surface is such that each axle is on ground of which the surfaces possess different adhesion factors.

It is appreciated, of course, that it would be possible to destroy the benefit of this condition if the percentage of rear-axle weight were exaggerated; as, for instance, when front-axle tractive effort multiplied by adhesion would be inadequate to move the chassis since the rear would sink and thus materially increase another militating factor—that of rolling resistance.

It might be suggested that a heavy front-axle load would be beneficial in such cases and, whilst it is appreciated that the ideal would be a variable loading between front and rear axles, a more important feature emerges which definitely rules out this desirability.

Assume therefore that a heavy front-axle load is provided and the vehicle encounters soft ground with its front axle, for example a ditch, then the extra loading would tend to produce cavitating of the wheels and rolling resistance would increase to an extent which would overbalance power input to the wheels, whilst the remaining tractive effort at the rear axle would add further rolling resistance to the front axle by virtue of its driving effort which would in such a case be exerted downwards.

It is thought, therefore, that as light a front-axle load as possible should be arranged for such conditions, always bearing in mind that a too-heavy rear axle would also be disastrous under other circumstances.

The effect of increased rolling resistance due to cavitation may be appreciated by study of the following approximate assumptions. It is necessary to have some knowledge of the conditions of terrain to be traversed and accordingly what is known as a " soil factor " is incorporated. These factors for the following ground conditions are—

| | |
|---|---|
| Fresh snow | 1 lb/in³ |
| Soft mud or quicksand | 1 to 2 lb/in³ |
| Freshly-ploughed loam | 4 to 7 lb/in³ |
| Dry sand | 20 lb/in³ |

and they cover a wide diversity of conditions.

Referring to Fig. 13, if G = ratio of rolling resistance to vertical load

$$G = \frac{P}{W} = \frac{Fh^2}{ZV}$$

or approximately $G = \frac{7}{8}\left(\frac{W}{KFD^2}\right)^{\frac{1}{3}}$ where K = soil factor.

For ground angles, or tyre sinkage $\theta$ up to 40°

$$G = \text{approximately } \frac{3}{4} \tan\left(\frac{\theta}{2}\right)$$

$$h = \frac{D}{2} \text{ versine } \theta$$

and $V = \frac{\pi D^2 F}{4}\left(\frac{\theta}{360} - \frac{\sin 2\theta}{4\pi}\right)$.

A tyre which is loaded to produce an angle $\theta$ of 30°, which is equivalent to a rut of approximately 6 or 7 in of wheel diameter, has a rolling resistance of 180 lb/ton in dry sand, whereas with a 14·00 in × 20 in tyre the rolling resistance increases to 360/560 lb/ton.

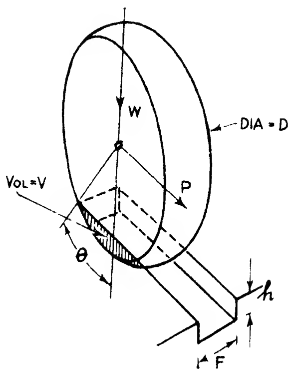



Fig. 13—Wheel cavitation

The power absorbed in overcoming rolling resistances of such values is considerable and it is considered necessary to provide at least 10 h.p. per ton as the minimum which should be available particularly for track-laying vehicles.

There is yet another factor which enters into the question of axle loading which is more apparent and of greater consequence on steep gradients. It concerns longitudinal torque reaction. The effect of this reaction is to reduce the load on the front wheels and more forcibly depress the rear, or in other words to add to the dead weight on the rear. The total load, of course, remains unaltered, but it will be seen that with the same ground-adhesion conditions as previously mentioned, that is uneven values for both axles, this feature militates somewhat against front-axle utility.

It is not, however, of such importance when all wheels are encountering the same adhesion coefficient. The tow load, whilst imposing additional work on the engine, has but little effect on axle loading; it does, however, increase the rolling resistance of the complete vehicle and in consequence further increases the required tractive effort. It is on account of this load that wheel spin occurs under certain conditions even after the foregoing theories have been incorporated in the design, since if the engine is to deliver sufficient output to overcome both rolling resistance of the tow and weight and adhesion of the tractor, the fact that only the tractor wheels absorb this power means that under some conditions the engine is too powerful by an amount of torque equal to the figure required to overcome the rolling resistance of the gun or tow load.

Hence it is possible for the tractor to negotiate clay comfortably without tow load, although with the latter wheel spin is experienced. It is definitely not a function of engine capacity, but entirely one of balance between power tractor weight and adhesion. It is, of course, appreciated that both longitudinal and transverse torque reaction exist, but it is important to note that both affect the chassis in different ways.

Transverse torque reaction is transmitted via the springs to the frame. In consequence two important features enter into the problem of its absorption; first that the track of the chassis should be as wide as possible in order to reduce the actual reaction effect at the ground, and secondly that the periodicity of the suspension springs must be unequal, or in other words they must not synchronise.

Of prior importance is the track, so that reaction effect will not depress the tyre to too great an extent and thus continually vary its running radius, promoting a similar inverse fluctuation in radius of the opposite tyre. The effect of uneven torque is therefore to establish a varying load through the tyre to the spring and consequent high-frequency oscillation. This is magnified as engine speed falls (torque rising), the road wheels assuming a greater load during depression so that axle weight and adhesion become unbalanced with torque input. This appears to suggest that again the light front load is preferable.

It will be realised that the position of spring attachment to the frame is of importance, as also is the degree of frame rigidity, as the longitudinal torque reaction, which is of considerable magnitude, provides an aggravating addition to the bounce phenomena (when once such a condition commences) directly through the frame in proportion to the wheelbase of the vehicle: this reaction, however, is transmitted to the wheels and does not emanate from that source as in the case of the transverse reaction. It follows that such a source, whilst not altering the total load on the vehicle, redistributes the load between front and rear axles since the front axle tends to lift under its influence. The rear springs therefore become depressed and relieved alternately, with consequent alteration to tractive adhesion. The ultimate resultant torque reaction from transverse and longitudinal forces has the tendency to twist the frame diagonally.

## EFFECT OF CENTRE OF GRAVITY

From investigations carried out, in which the effects of combined transverse and longitudinal torque reaction on each individual

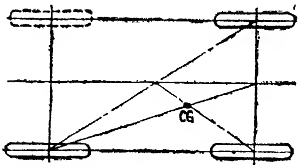



Fig. 14a—Rear axle control

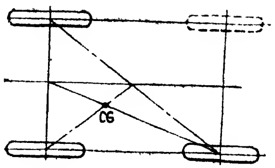



Fig. 14b—Front axle control

wheel were measured, it was noted that the rear nearside wheel load is reduced by approximately the same percentage as the front offside is increased, whilst the front nearside loses a similar amount to that gained by the offside rear; and it will be appreciated that any reduction in either track or wheelbase would increase these transfer percentages. Since the wheelbase is more or less determined by other considerations, it is desirable to widen the track to the utmost possible. Hence the desirability of establishing a definite relationship of track to wheelbase.

In absorbing these reactions, the suspension-spring disposition (transversely and longitudinally) is very important, particularly in relation to the frame, which is the medium through which engine torque reaction is transmitted. " Patter " depends to a certain degree upon the position of the centre of gravity of the whole vehicle in relation to the points of reaction of the springs; for example, if the centre of gravity of the vehicle falls in such a position that it is possible for it to be balanced by the two front wheels and one rear wheel (plan view) then any lurching which occurs at one side of the front axle results in the body following that movement. In a case where the c.g. is such that the body could be supported by two rear wheels and one front (plan view) then the motion of the vehicle follows relatively the motion of the rear axle.

It will be appreciated, therefore, that tendency to patter at the front end of the vehicle is aggravated where the vehicle centre of gravity falls within the front-axle control (Fig. 14). " Patter " is a particular form of high-frequency periodicity, caused probably by the coincidence of the spring period with that of the chassis and body; and, since periodicity is a function of deflection, it would appear that patter can be reduced by increase in deflection. Consider, then, the effect of spring location and attachment upon any forces tending to cause lurch towards the front axle, of a vehicle whose approximate centre of gravity conforms to that of Fig. 14b. The type of chassis to which Fig. 15a refers is typified in Fig. 14b, wherein most of the load is at the forward end. Engine-torque reaction acts upon the frame side members in a downwards direction on one side and upwards at the other side. The coil-suspension springs are inside the frame member (plan view) and in consequence are not located in so favourable a position to resist side movement or sway of the frame as if they had been more widely spaced. Moreover, the torque reaction is applied at a position along the frame much in front of the point of spring attachment so that any reaction tends to tip one side downwards and forwards at one corner, with only one point of resistance. The frame and suspension loads are therefore unbalanced, whilst at the same time frame movement due to this unbalance is transmitted to the coil spring, setting up continual compression and lengthening with consequent increase and decrease in its applied load to the axle, or in this case, torque tube.

Now, bearing in mind that the wheel already has its own driving mechanism and consequent torque reactions to absorb, it experiences a succession of loads increasing and decreasing alternately according to the number of impulses per revolution of the engine (4- or 6-cylinder) at exactly the same time as the torque-reaction fluctuation occurs in the wheel-driving mechanisms, so that as the pressure on the wheel is relieved and increased thus the required amount of input varies to the wheel.

The total torque must, however, be absorbed over all the drives and in consequence the excess from one wheel is distributed to the others. A general disturbance occurs, particularly when the adhesion factor is low, and a continual hunting of input torque commences through each of the four drives. One or even two wheels, at some period wherein maximum torque is required to negotiate a gradient whereupon adhesion is low, will be submitted to an excess torque and immediately is/are subjected to an increased torque reaction in consequence. Running radius of tyre becomes another variable quantity and generally a vicious circle is set up.

The adhesion factor does not vary, but the wheel loading, tyre radius, and spring loads are never constant. A condition thus develops wherein all these loads seek to synchronise, and a period is set up between the springs and the wheels with consequent promotion of patter.

## EFFECT OF VEHICLE SUSPENSION

If the chassis is considered as a unit mounted on four coil springs spaced close together as in Fig. 15a a general idea of the effect of hunting torque will be appreciated, particularly as all reactions are each partly absorbed at the approximate centre of the chassis (Fig. 15a). Were these springs situated at wider centres longitudinally and outside the frame transversely, they would afford a much greater opportunity for absorption of the loads. The ideal longitudinal position is immediately above the axle centres, as at these points the reaction moments are a minimum, whilst deflection is not so critical in preserving frame stability.

An example of this type is illustrated by Fig. 16, in which the springs possess a much more desirable location both transversely and longitudinally than Fig. 15a since they are immediately above the axis from which torque reaction emanates. Moreover, the arrangement permits wider transverse spacing with consequent greater possibility of increased deflection and also periodicity adjustment. This type of suspension incidentally (and apart from the question of weight distribution) possesses an inherent defect with regard to ground clearance which, compared with Figs. 15a or 15b, amounts to serious proportions. The axle centre is fixed to the frame and consequently does not rise with the road wheel.

During articulation, ground clearance is reduced by the amount of wheel rise, or in other words ground clearance is inversely proportional to wheel rise; further, since the axle arms are pivoted near the " banjo " portion, the wheel track is continually subjected to variation, and this is particularly undesirable from a wheel-loading aspect, when it is recalled that the torque reaction moment is varying accordingly.

A coil spring, however, is without one of the most important characteristics—that of leaf friction—necessary to prevent at the outset any increase in frequency of possible oscillations. The suspension springs will absorb engine-torque reaction, via the frame, and will transmit the impulse to the wheels. The springs thus become loaded through an additional source. To reduce the magnitude of this load they should be placed at as wide a centre as possible from the source of oscillation. The friction between the leaves of the laminated spring may assist in preventing the development of a period, whilst the fact that they possess transverse rigidity is of considerable assistance.

A useful idea might be the introduction of progressive springs wherein each progression has a different periodicity and it could be arranged so that (since the first progression is very light in comparison to chassis weight) any addition to the dead loads of chassis weight bring the second progression into action, resulting in the use of a certain proportion of friction from the first leaf of that portion of the spring. As previously stated, oscillations are therefore prevented from developing a period. Moreover from a glance at Figs. 15b and 16 in elevation it will immediately be apparent that in addition to the advantage possessed by wide transverse location any reactions from the wheels will be transmitted to the frame through points widely spaced longitudinally.

Reverting to the normal road vehicle, it will be realised how important is the suspension in connection with the general performance and, having very briefly discussed the effect of spring location, we may consider another aspect which is intimately bound up with the quality of riding and which must therefore be related to performance.

In considering vehicle suspension we are not mainly concerned with spring rates, but primarily with deflection. The main motions of the vehicle are pitch and bounce, pitch being a rotary oscillation about a centre within the wheelbase and bounce, a rotary oscillation outside the wheelbase. The position of these centres is important and they are fixed not by actual deflection of front and rear ends but by the relative deflections of the two ends and the polar moment

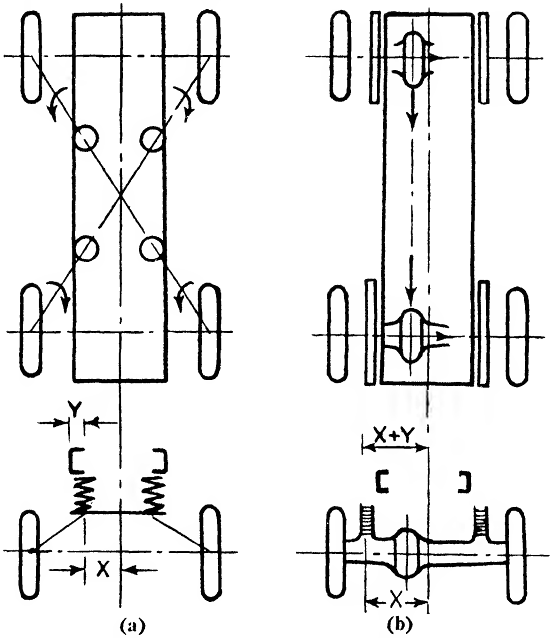



Fig. 15—Effect of spring location

of the vehicle. These centres have certain characteristics, two of which are that the motion of each is a simple harmonic whatever the car displacement, and that additional load at either centre does not affect the frequency at that centre, but reduces the frequency at the other. In order, therefore, to produce pitch, load must be applied at the bounce centre and vice versa. Since the centres are close enough to the wheel centres, the front loads mainly cause pitch and rear loads bounce, the faster motion being started first during vehicle motion. The front therefore drops as the rear rises, whilst this results in the maximum angular motion as produced by the vehicle passing over a single obstruction. Softening the front springs tends to move both centres forward and the pitch rapidly slows ; on the other hand bounce is reduced slowly.

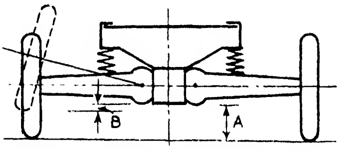



Fig. 16—Effect of spring location

Continuation of the front-spring softening eventually eliminates the excitation of the pure pitch or bounce from loads at either end of the vehicle. The bounce centre is already at infinity and reappears behind the vehicle, and as the pitch centre moves ahead of the centre of gravity it approaches nearer the rear, so that instead of the pitch frequency slowing down rapidly it is the bounce which possesses this quality. Excess softening of the front springs in relation to those of the rear causes the front end to move forward without apparent vertical displacement, whilst the rear end of the vehicle executes individual movement vertically.

On average cars, therefore, it would appear that springing should be arranged so that under maximum load the rear deflections should be equal to the front deflections at their lightest load. The heavier

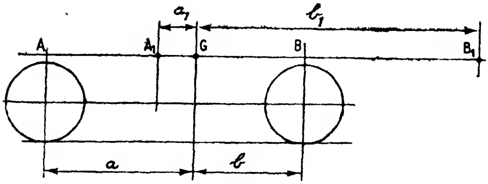



Fig. 17

vehicle requires that both springs should have equal deflections only at maximum load. It has been generally considered that if the ratio between the radius of gyration K of the vehicle squared, and the product of the two parts of the wheelbase either side of the centre of gravity (a and b) is equal to 1, then the best vehicle suspension has been obtained. However, this does not eliminate vertical interference kicks or horizontal vibrations and later investigations on parallel springing suggests that pitch frequency/bounce frequency should equal $\sqrt{\frac{ab}{K^2}}$, so that $\frac{K^2}{ab} = 0{\cdot}75$. However, this ratio must be proved in test and may therefore vary according to type of vehicle. The suspension must allow a retention of vehicle stability and precision in handling.

In explanation of the relation $K^2 = ab$, it should be noted that the problem may be divided into several portions—

(1) To find a system of two masses of equal moments to the sprung part of the car.
(2) To find an elastic system which is equivalent to the springs at the axles.
(3) To place the masses over imaginary springs of known stiffness.
(4) To find by algebraic methods the spring periods.

Where in (1) $A_1$ & $B_1$ be the two masses (see Fig. 17)
$K$ = radius of gyration of the vehicle
$G$ = centre of gravity of the vehicle

for equal moments

Equal mass $A_1 + B_1 = M$
Common c.g. $A_1 a_1 = B_1 b_1$
Equal inertia moments $A_1 a_1^2 + B_1 b_1^2 = MK^2$

and as A, B & M are common, we get $a_1 b_1 = K^2$ and this governs the position of equal masses. In (2) spring stiffness = $\mu$ & $\lambda$ at dimensions a and b (wheelbase) apart, and distances l & m from spring centre C (see Fig. 18).

The three required conditions for equality of the two spring systems are—

The sums of stiffness must be equal $\therefore \lambda + \mu = \lambda_1 + \mu_1$
Both springs must have the same line of resultant action

$$\lambda_1 l_1 - \mu m_1 = O = \lambda l - \mu m$$

Both springs must have the same angular stiffness

$$\lambda_1 l_1^2 + \mu_1 m_1^2 = \lambda l^2 + \mu m^2 = \mu_1 m_1 (l_1 + m_1).$$

We therefore get from the first two conditions

$$(\lambda + \mu)\, l_1 = \mu_1\, (l_1 + m_1)$$

and, combined with the last of the three expressions (that for angular stiffness), we get

$$l_1\, m_1 = \frac{(\lambda\, l^2 + \mu\, m^2)}{(\lambda + \mu)} = h^2$$

The stiffness of the springs is therefore

$$\mu_1 = \frac{(\lambda + \mu)\, l_1}{(l_1 + m_1)} \Big/ \lambda_1\, \frac{(\lambda + \mu)\, m_1}{(l_1 + m_1)}$$

In the third part of the problem (to place these masses over the imaginary springs) erect verticals from G and C (the centre of gravity and spring centres respectively) KG and $H_1C$ equal to k and $h_1$, bisect $KH_1$ and draw EO at 90° to $KH_1$ to point O, and with O as origin describe a semicircle through H and K to include P and Q. These points are those of simple harmonic motion from the geometry of the circle, as PC, CQ $= h^2 = l_1\, m_1$ and PG GQ $= K^2 = a_1\, b_1$.

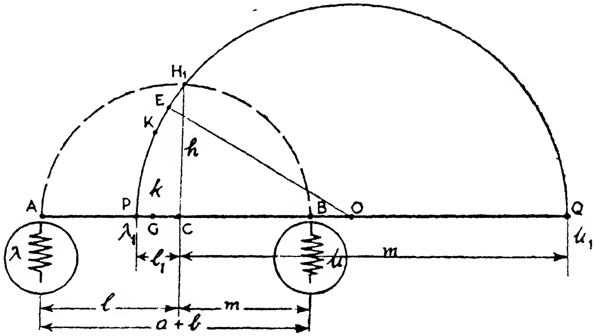



Fig. 18

We now have to find the spring system to give a harmonic point P. If $P_1G_1$ and radius of gyration GK $=$ K are known, then from a semicircle drawn through P and K from any point C we have the ordinate of the semicircle CH $=$ h; thus from the formula for the line of resultant action and for the expression for $h^2$ (second part of the problem) we have $\frac{\lambda}{\mu} = \frac{m}{l} = \frac{(m^2 - h^2)}{(h^2 - l^2)} \therefore l\, m = h^2$

Proceed to draw another semicircle on the wheelbase A B which intersects the previous circle at $H_1$. Drop the perpendicular $H_1C_1$. Point C denotes the spring centre and the ratio of spring stiffness. To define the periods in terms of the system of masses and springs

$$T_1 \doteq 2\pi\sqrt{\frac{A_1}{\lambda_1}}$$

$$T_2 = 2\pi\sqrt{\frac{B_1}{\mu_1}}$$

substituting for $A_1$ and $B_1$ we get

$$T_1 = 2\pi\sqrt{\frac{b_1 M}{m_1(\lambda + \mu)}}$$

$$T_2 = 2\pi\sqrt{\frac{a_1 M}{l_1(\lambda + \mu)}}$$

In geometrical form

$$T_1 = 2\pi\sqrt{\frac{M \times GQ}{CQ(\lambda + \mu)}}$$

$$T_2 = 2\pi\sqrt{\frac{M \times GP}{CP(\lambda + \mu)}}$$

Inseparable from suspension and its qualities is the problem of steering and the design of the front axle.

S'

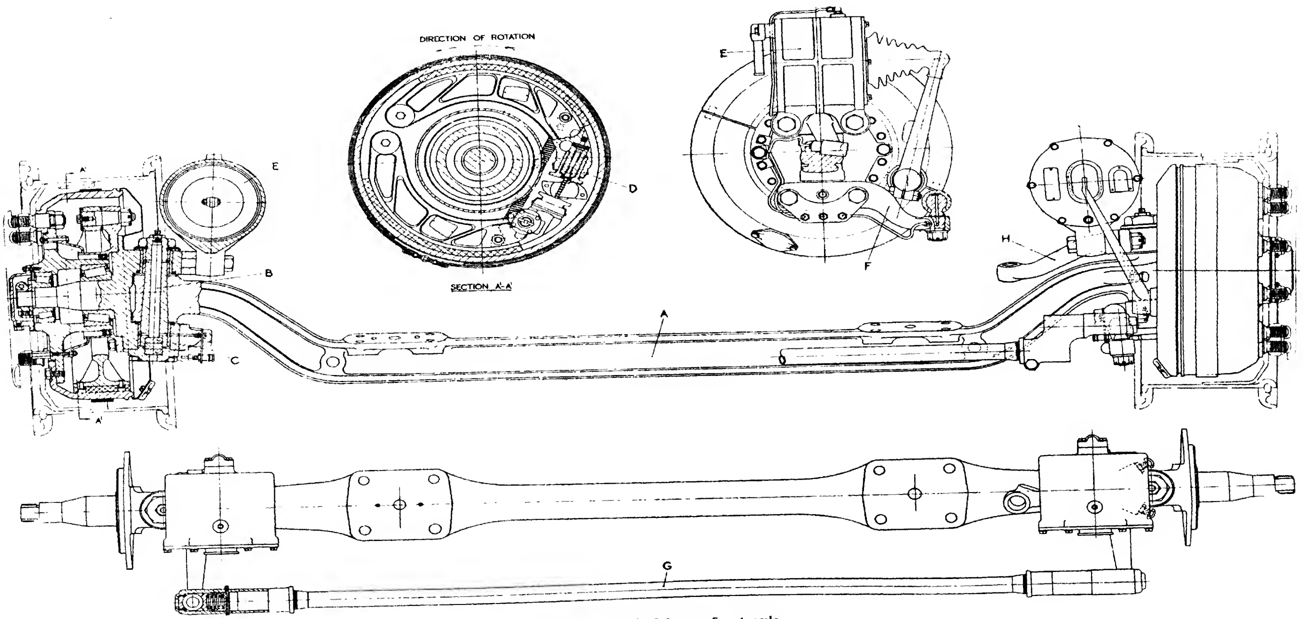



**Fig. 19—Typical heavy front axle**

A, axle bed; B, taper king pin; C, thrust bearings; D, automatic shoe adjuster; E, air brake cylinder; F, track rod arm; G, track rod; and H, steering arm

[*To face page* 37

## CHAPTER 2

# FRONT AXLES AND SUSPENSION

IT is, of course, impossible to estimate all conditions of front-axle loading, but some of the stresses are sufficiently small to become absorbed in figures of greater importance, and these will be neglected. The front axle should be considered in conjunction with the vibrations of the vehicle as a whole, together with the characteristics of the suspension system.

Of primary importance is the relation of *the proportions between unsprung and sprung mass*, bearing in mind that it is not always desirable to *reduce* the unsprung weight. A vehicle traversing rough surfaces experiences two critical speeds, one when acceleration and displacement are great, and the other when acceleration is great and amplitude small. In the case of the former condition, this occurs at low vehicle speeds, whilst the latter phenomenon is experienced at speeds five or six times that of the lower critical speed. Again, the first period is occasioned by the difference in natural periods between the front- and rear-suspension systems and the consequent chassis displacement. It takes the form of lurching at the critical speed of the rear springs, although its proportions are not serious as the displacements exist only over a short period of time. The second critical speed occurs when the front axle is vibrating with considerable amplitude and its acceleration is greater than that of the first period. Chassis displacements are smaller and rapid shocks of less amplitude occur over a longer period of time. This is the condition which imposes severe mechanical stresses upon the axle.

Reference to Fig. 20 illustrates the magnitude of shocks under varying conditions of unsprung mass, from which it will be seen that a compromise between suspen-

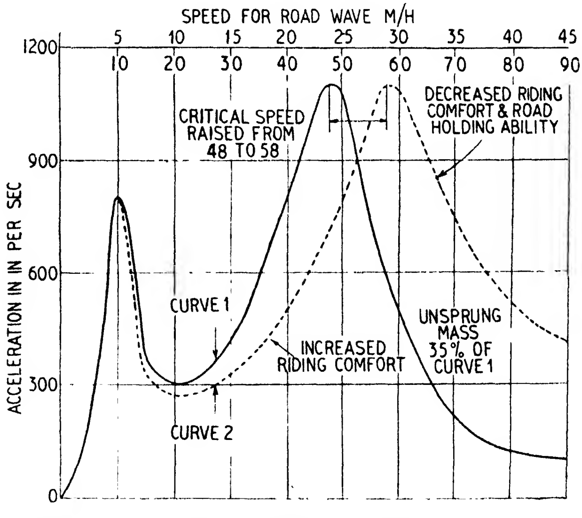



Fig. 20 -Effect of unsprung mass upon upward acceleration

sion characteristics and vehicle speed is necessary. The curves show that reduction of unsprung mass improves the riding comfort during the first portion of the curve, but after the critical speed is reached both decreased riding ability and road-holding capabilities are reduced. It should be arranged that the maximum desired vehicle speed should not coincide with the speed shown at the peak of the curve. From the diagram, the vehicle speed range to pass through the period is evident.

As an obstacle is encountered, an impact is imparted to the wheel, the maximum acceleration of the chassis being reached before the rise becomes appreciable. This velocity is a function of impulse and unsprung mass. Its magnitude is $V = \frac{I}{m}$, where I = impulse and m = mass unsprung. The acceleration in an upward direction is given by

$$a = \frac{I}{M}\left(\frac{K}{m(F + K)} + \frac{D^2}{m^2}\right)^{\frac{1}{2}}$$

where M = sprung mass supported by one wheel
F = spring factor of suspension
D = damping factor of road springs
K = spring factor of tyre

from which it will be noted, when compared with Fig. 20, that a reduction of unsprung weight increases the lift acceleration due to impact. It depends, however, upon the vehicle speed and the size or character of the obstruction.

An approximation of the impulse which also represents the *shock load applied to the axle* is given by the expression

$$I = \frac{Khl}{2V}$$

where h = height of obstruction
and l = length of obstruction.

*The shape of the axle bed* is not generally considered other than for requirements of dimensional clearances under engine sump and road clearance, but it has some small bearing upon the oscillation of the front end of the vehicle. As compared with the dropped centre axle the straight beam raises the centre of oscillation to a more disadvantageous position, whilst at the same time it decreases the angular displacement of the wheel for a similar rise of tyre from the ground. In consequence the resistance to motion is of smaller magnitude than that of the dropped axle bed. As opposed to this condition, the straight beam affords a reduction in gyroscopic effect from the wheels because of this reduced angular motion. An increase in axle track would, of course, produce similar advantages, and the conclusion therefore is that a satisfactory compromise is necessary between track and axle drop.

*Weight distribution* is of importance as its influence is felt in connection with the distressing phenomenon of wheel wobble. It is not proposed to deal with this subject at length; it will be sufficient to consider the vibration which occurs at high speed wherein the axle vibrates about its centre point, simultaneously with the transverse rotational displacement of the axle ends. It has been established that if it were possible to dispose all the mass at the axle centre, then the period of transverse vibration would be considerably reduced, and would almost certainly be entirely damped out by the application of comparatively small forces acting at the spring pad centres.

A parallel to this condition may be cited in the rear axle, wherein a considerable proportion of the mass is concentrated at the axle centre. The damping of these vibrations is not assisted by the addition of front-wheel brakes, on account of the increase in inertia moment about the centre point of the axle. The effect is to reduce the period of vibrations and to bring it closer to the speed range of the vehicle. In order to reduce the additional inertia moment to as low a degree as possible it is desirable that as much of the mass of front brakes and steering head should be arranged at the centre of the axis of wheel rotation.

It has been seen that the *spring centres* bear relation of some importance in damping transverse oscillations as they exert a balancing moment effect, whilst of course they influence the total axle-weight distribution. A narrow spring track decreases the period, and a wide track increases the frequency of axle vibration. It is therefore correct to set the springs at as wide centres as dimensionally and constructionally possible. The dimension is controlled not only by dynamical considerations, but by such practical requirements of steering lock, frame width, and overall width. The compromise is therefore one for individual consideration.

There are obviously practical limitations to the *dimension between swivel-pin centre* and the plane of the wheel, but these centres are important as they involve the vehicle performance and care must be exercised in arranging the necessary compromise. It is appreciated that the speed at which the wheels can maintain conical precession or gyroscopic action is dependent upon the cone which is formed through these centres (see Fig. 22). The velocity is higher when the centres are longer and lower when the dimension is curtailed. Long centres are therefore indicated as one of the means of placing this velocity outside the vehicle speed range.

*The diameter of the swivel-arm bearing* should be calculated from the maximum fibre stress in the section, whilst considerable care should be exercised in the choice of material, for shock loads can only be dealt with by the margin allowed in the safety factor. The main fact which must be borne in mind is that ultimate failure is not so much occasioned by the magnitude of stress as the included range of stress. Stress reversals also tend towards rupture of material.

Transverse vibration constitutes a case where reversal of stress occurs and when assessing the limiting range the natural elastic limit should be used as a basis of calculation, the term " natural " being used advisedly since it is known that material stress limits of a material can be artificially controlled, and that after a series of reversed loadings this limit settles down to a fixed value where the allowable stress is the same for both compression and tension. This value is termed the *natural* elastic limit.

Localised stresses can be avoided if due care is taken on the detail drawing to specify the correct grade of finish, whilst the change of section should be radiused as large as possible. As a guide to illustrate the importance of this, it is stated that a 1 in radius fillet gives 100 per cent of the original fatigue limit; $\frac{1}{4}$ in radius, 92 per cent; square corner, 49 per cent; 90° Vee notch, 40 per cent of the original fatigue limit.

*Track-rod dimension* can safely be arrived at on the basis of comparison, but the stresses to which it is subjected present certain complications. If, however, a reasonable factor of safety is allowed, then the following formulae will cover the design for all practical purposes, since it is almost impossible to estimate the effects of loads imposed through any of the extraordinary circumstances which are noted in attendant sections of these notes. A reversal of end loading incessantly occurs, whilst the weight of the track rod itself assumes considerable proportions so far as stress due to vibration is concerned. Assume that the mass of the rod is as an evenly-distributed load, then the approximate calculations for maximum intensity of stress involve loads from (a) axial pull and (b) axial thrust, both these conditions combining a lateral load due to track rod weight.

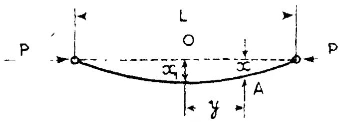



Fig. 21

Where L = length between supports
 l = lateral load carried per unit length
 P = end thrust

then see Fig. 21.

$$\text{BM at A due to lateral loading} = \frac{l}{2}\left(\frac{L^2}{4} - y^2\right)$$

and Py due to end thrust P.

The sum of these moments is equal to $-EI\frac{d^2x}{dy^2}$, where I = moment of inertia of cross section perpendicular to the plane of flexure,

$$\text{therefore } EI\frac{d^2x}{dy^2} = -\frac{l}{2}\left(\frac{L^2}{4} - y^2\right) - Py$$

and the approximate solution of this equation is written by substituting

$$\frac{lL^2}{8} \cos \frac{y}{L} \pi \text{ for } \frac{1}{2}\left(\frac{L^2}{4} - y^2\right)$$

thus $$x = \frac{lL^2 \cos \pi y/L}{8 \quad P_1 - P}$$

where $P_1$ = Eulers limiting value for the strut having both ends pivoted, that is $\frac{\pi^2 EI}{L^2}$

$$x_1 = \frac{lL^2}{8(P_1 - P)}$$

and BM at origin O

$$= \frac{lL^2}{8} \frac{P_1 - P}{P_1}$$

*Maximum intensity of bending stress is therefore*

$$f_b = \frac{BM_0 x^2}{I} \text{ or } \frac{BM_0 d}{2I}$$

where $x^2$ is half the depth of section d.

*Maximum intensity of compressive stress* (positive sign) and tensile stress (negative sign)

$$f_c = \frac{BM_0 d}{2I} \pm p$$

where p = mean intensity of stress $\frac{P}{A}$ and A = cross sectional area. These approximations assume a symmetrical section and end load considered to be the maximum possible on one wheel = $\frac{Wx}{y}$, when W = maximum load on wheel, x = distance between wheel centre and pivot axis, and y the length of knuckle arm.

The arm x—y twists about the axis xy.

The twisting moment at any point on this axis is of course FH, F being the braking force (see Fig. 22).

At other sections stress due to F is

at AA — Torsion magnitude FH and bending Fa
BB — Torsion only magnitude FH
CC — Torsion FH and bending Fc
DD — Torsion FE and bending Fd.

These stresses are in addition to those of bending due to force W.

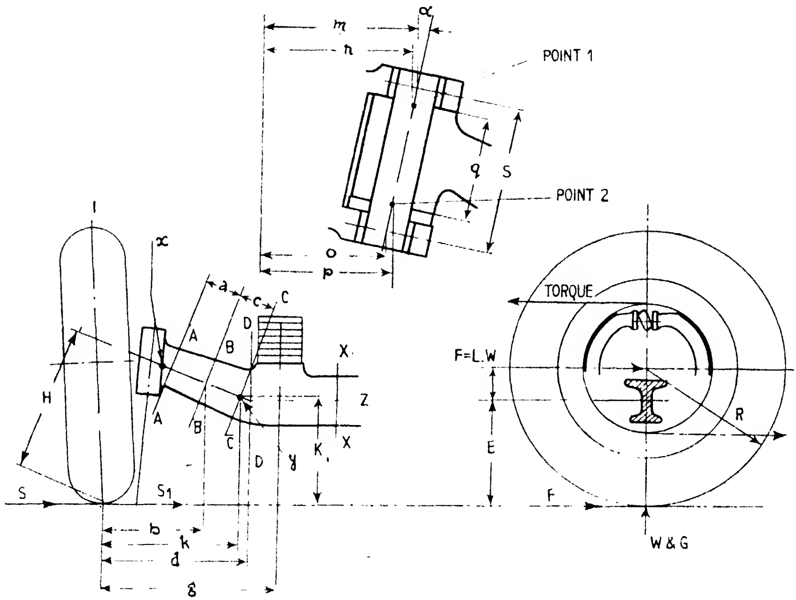



Fig. 22—Stresses due to braking

## FRONT AXLE STRESSES

### *The Axle Beam*

Referring to the part of the beam between the spring pads, say section XX (Fig. 22), point z, the whole of this section is subjected to bending moments only, and where

W = total load per wheel (lb)

g = horizontal distance from centreline spring pad to centreline of load at ground (in)

$\mu$ = coefficient of tyre and road friction

Vertical bending moment $= BM_V = Wg$

Horizontal bending moment $= BM_H = Wg\mu$

Resultant $BM_R = \sqrt{(BM_V)^2 + (BM_H)^2}$

For a beam of I section $Z = \dfrac{BD^3 - bd^3}{6D}$

and stress $= f = \dfrac{BM_R}{Z}$.

Take a point y on axle arm which is subjected to both bending and torsion where radius of wheel = R

vertical distance of y from ground = K.

Vertical $BM_V = WK$

Horizontal $BM_H = WK\mu$

Resultant $BM_R = \sqrt{(BM_V)^2 + (BM_H)^2}$

Twisting moment $T = W\mu R \left(\frac{K}{R}\right)$

Equivalent $BM_E = \frac{BM_R + \sqrt{(BM_R)^2 + T^2}}{2}$

Modulus of section for oval section $= \frac{\pi bd^2}{32}$

Modulus of section for circular section $= \frac{\pi D^3}{32}$

and stress $= \frac{BM_E}{Z}$.

From the foregoing formulae, and in accordance with the stresses to which various sections of the axle from spring pad to king-pin boss are subjected, convenient sections may be chosen, but care should be exercised in blending in of the cylindrical portion to the I section.

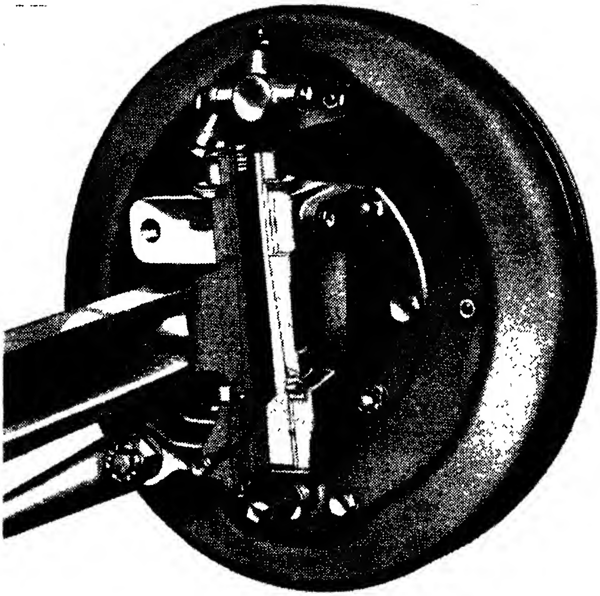


Fig. 23— Arrangement of king pin and bearings

*King Pin and Bearings*

For all practical purposes the sources of king-pin loads may be considered as emanating from

(a) total load on the axle,
(b) resistance of tyre on road when braking,
(c) braking torque,

and take the form of shear loads.

*Shear Load at Point* 1 :

(a) Due to load on axle $= \frac{Wp}{q}$ in a transverse direction.

(b) Resistance of tyre on road when braking

$$= \frac{W\mu}{2} \text{ in a longitudinal rear direction.}$$

(Note king pin in double shear.)

(c) Due to braking torque $= \frac{W\mu R}{q}$ in a longitudinal forward direction.

$$\text{Total shear load} = \sqrt{\left(\frac{Wp}{q}\right)^2 + \left(\frac{W\mu}{2} - \frac{W\mu R}{q}\right)^2}$$

$$\text{Stress} = \frac{\text{Total shear load}}{\text{Cross-sectional area}}.$$

The load at point z is calculated in a similar manner as for point 1 except that the lever arm is n. The load imposed by braking torque is in the same direction as that from resistance of tyre to road. The total shear load is therefore increased at this point.

*Load on King-pin Thrust Washer*

$$= W \sec \alpha$$

Under normal conditions static loading only is necessary as a combination of maximum braking and steering loads simultaneously is rarely encountered.

*King-pin Bearings* (Static Load):

$$\text{Load on upper bearing} = \frac{WO}{S}$$

$$\text{Load on lower bearing} = \frac{Wm}{S}$$

$$\text{Unit loading} = \frac{\text{Load on bush}}{\text{Projected area}}.$$

*Front Axle Bearing Loads* (Normal Running) (see Fig. 24):

Where A $=$ centre distance between bearings
B $=$ distance from centreline of outer bearing to line of vertical ground reaction
R $=$ running radius of tyre
$W_F$ $=$ front axle load

$$\text{Radial load on inner bearing} = \frac{W_F}{2} \times \frac{B}{A}$$

$$\text{Radial load on outer bearing} = \frac{W_F}{2} - \left(\frac{W_F}{2} \times \frac{B}{A}\right).$$

*Bearing Loads when Cornering :*

$$\text{Centrifugal force} = \frac{W_F V^2}{gr}$$

Load on outer wheel due to CF =

$$\frac{W_F V^2}{gr} \times \frac{h}{T}$$

Vertical ground reaction on outer wheel =

$$0{\cdot}5\ W_F + \left(\frac{W_F V^2}{gr} \times \frac{h}{T}\right) = G_V$$

Side reaction at outer wheel =

$$\frac{G_V}{W_F} \times \frac{W_F V^2}{gr} = S$$

*Radial load on inner bearing* due to vertical ground reaction $G_V$

$$= \frac{G_V B}{A} = G_1.$$

Radial load on inner bearing due to side reaction S

$$= \frac{\frac{G_V}{W_F}\left(\frac{W_F V^2}{gr}\right) \times R}{A} = S_1$$

*Total radial load on inner bearing*

$$= G_1 + S_1 = L_1$$

Radial load on *outer* bearing due to G

$$= G_V - G_1 = G_2$$

Radial load on *outer* bearing due to S

$$= \frac{S \times R}{A} = S_2$$

*Total radial load on outer bearing*

$$= S_2 - G_2.$$

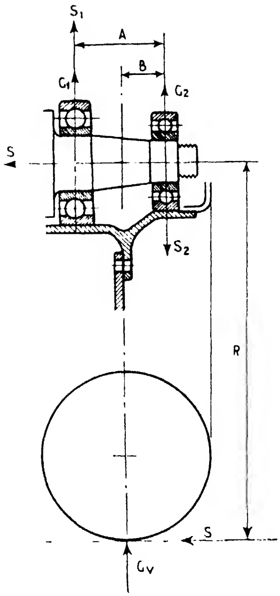



**Fig. 24—Bearing loads**

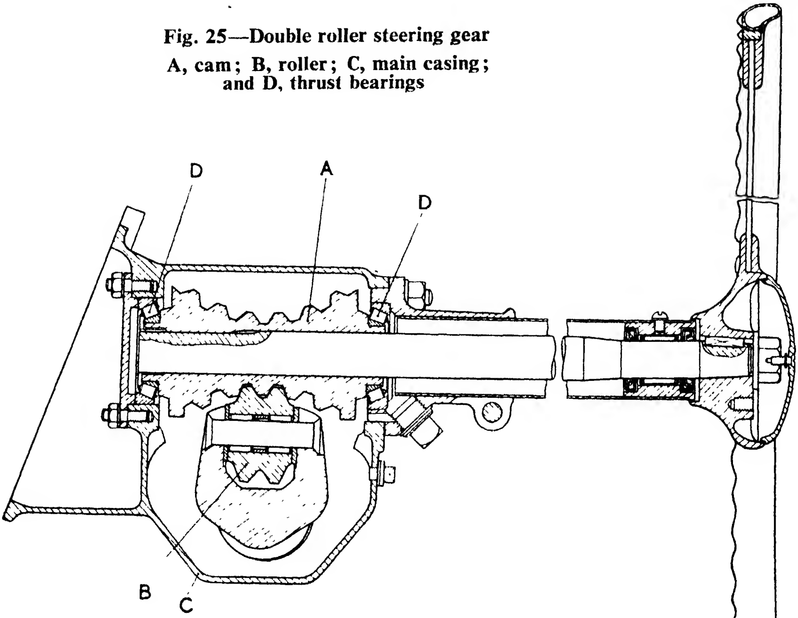
## STEERING

The effect of front-axle design upon steering qualities has been briefly set down, but braking and suspension effects present further problems, the incidence of which are almost incalculable with any degree of accuracy.

In considering suspension, the tyre must be considered as an additional damping medium through which vibrations and oscillations may be absorbed. *Ground contact area* should be taken into account when deciding upon the steering reduction ratio necessary; these areas of course differ with tyre size and pressure, low-pressure tyres having the greater pressure round the *edges* of the contact area.

This condition would be ideal if it were possible to concentrate such loads at the centre of the tread. Furthermore the effective centre of contact of a pneumatic tyre is not at the base of its vertical

**Fig. 25—Double roller steering gear**
**A, cam; B, roller; C, main casing; and D, thrust bearings**




axis when the vehicle is in motion, but at a distance forward of the intersection of this axis and the ground line. The reason for this is accounted for by the fact that the front portion of the tyre becomes compressed or deflected during motion whilst the rear half is allowed to recover, the internal resisting forces increasing the vertical loads required for compression whilst the forces of recovery or expansion in the rear half are reduced. This dimension is therefore a function of pressure, and steering will be affected further in its castor action.

If W = load on wheel (lb)
F = rolling resistance

$$\left(F = \frac{Pr}{V}\right)$$

V = velocity of vehicle (ft/sec)
P = power consumed by tyre (ft. lb/sec)
r = running radius of tyre (ft)
x = offset of vertical thrust (ft)

then $$F = \frac{Pr}{V}$$

$$F = Wgx$$

$$= \frac{Fr}{WgV}.$$

This dimension should be taken into account in any calculation concerning castor angle (see Fig. 26).

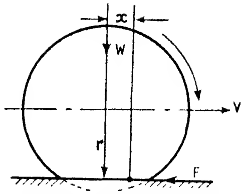



Fig. 26—Effective centre of tyre contact

Front-wheel braking necessitates compromise in the angle of inclination of the pivot pin. The addition of weight at the hub gives rise to a gyroscopic effect that is evident in extreme cases of wheel wobble. From the aspect of braking, centre-point steering is desirable as it reduces steering deflections when brakes are applied evenly. The results of uneven application will have been noted under the heading of brake performance. The track rod is subjected to very severe stresses in cases where the swivel pin is positioned at some distance from the point of ground contact with the tyre.

*The inclined pivot pin or king pin and vertical wheel* are not the soundest compromise, since the wheels are in unstable equilibrium. The path of the swivel head traverses an ellipse from lock to lock, hence the heavier steering at low speeds. At high speeds the reverse is the case, steering is rendered lighter by virtue of the additional castor effect.

Neither is *inclined wheel and vertical pivot pin* altogether satisfactory, as either a castor or anti-castor effect arises, dependent upon the radius of the path through which the wheel is constrained, since forward movement of the vehicle tends to an increase in the angle of lock. Referring to Fig. 27, the angle of the swivel head forms a cone CBO at the point of intersection with the ground, and the tendency of the wheel in motion is to describe a circle whose centre is O. The steering mechanism restrains this tendency. If $R_1$ is the radius through which the wheel is restrained, anti-castor effect will result if R is less than this radius. A compromise between the two arrangements is therefore indicated.

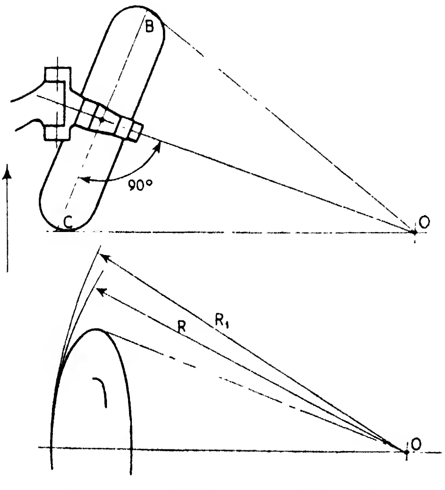



Fig. 27—Inclined wheel: vertical pivot

The inclined pivot pin gives additional castor effect, whilst it permits the swivel-head angle to be small enough to lie within the smallest turning circle of the vehicle. Positive castor effect results. The compromise of angles lightens the effort at the steering wheel and allows a certain moment at the point of contact of tyre and ground to be maintained.

It is not intended to repeat the principle of the generally-utilised *Ackermann layout of steering* geometry, since in any event it only takes care of accuracy during certain phases of steering. Study of an Ackermann layout will show that it places the greater slip angle on the inner wheel, which is the more lightly loaded, during sharp turns on a hard road. It must be appreciated that on all turns above speeds of 35/40 m/h (even turns of low curvature) are made about a centre *in front* of the wheels, so that Ackermann geometry applies only to relatively low speeds. Moreover, the actual centre of vehicle rotation differs greatly from the centre shown by Ackermann layout, which does not take into account the slip angle between the wheel planes and their paths of travel (see Fig. 29).

$\alpha$ = angle of front wheels
$\beta$ and $\beta_1$ = slip angles.

Note that $\beta_1$ controls the path of vehicle on the curve. The approximate radius of the turn $= \dfrac{1}{\alpha}$ if $\beta$ and $\beta_1$ are equal, but if $\beta$ exceeds $\beta_1$ the radius of turn increases and in consequence the angle of lock must be increased: $\beta_1$ is generally reduced by tractive effort. Thus this condition is often encountered and R (radius of turn) may be stated.

$$R = \frac{1}{\alpha + \beta_1 - \beta} \text{ approximately}$$

or if $\beta = \beta_1$

$$R = \frac{1}{2 \sin \alpha} \sqrt{\sin^2 \alpha + 4 \cos \alpha \cos \beta \cos (\alpha - \beta)}$$

The slip angles $\beta$ and $\beta_1$ produce certain side thrusts, proportional to them, but independent of speed and load on the wheel as the slip angle of a tyre for a given side thrust is independent of speed. Tyre size and any tyre pressure increase, however, increase these thrusts, which are applied at a point to the rear of the centre of ground contact (see Fig. 28) and, in consequence, a torque is produced which tends to straighten the wheel about the king pin. It is of interest to note that, although the side thrust is independent of wheel load, this torque is increased with wheel-load in-

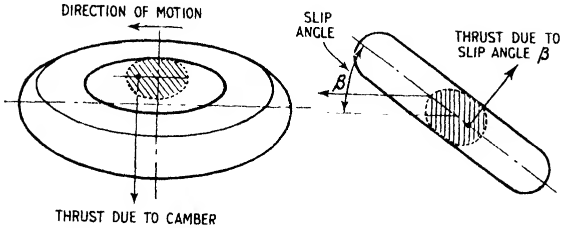



Fig. 28

**Camber thrust** **Slip angle thrust**

crease. In a parallel independent action the wheel planes tend to take up an acute angle with the plane of the road in the direction of the outside of the turn, and as the effect of changing the camber of a rolling wheel is to produce a thrust in the direction of this acute angle, the point of application is forward of the centre of ground contact (see Fig. 28).

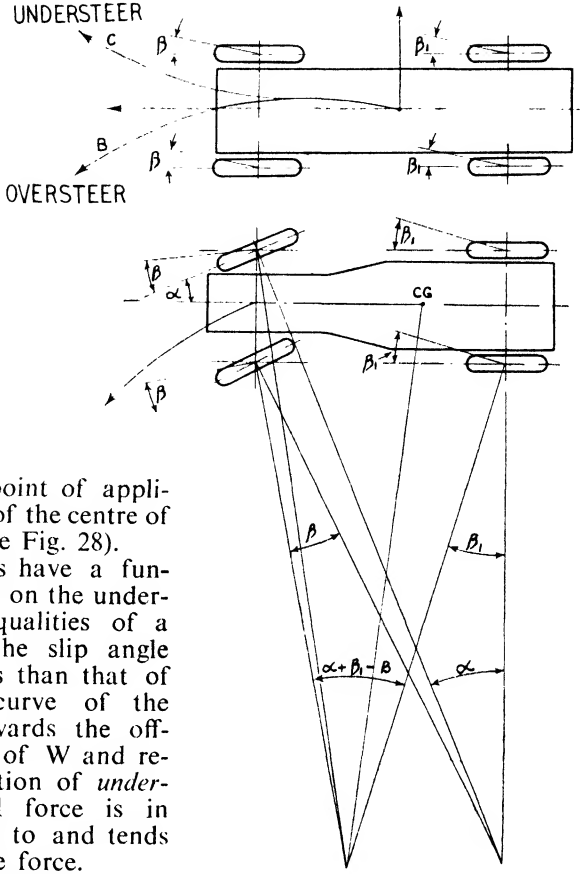


Fig. 29—Vehicle turning centre

These slip angles have a fundamental influence on the under- or over-steering qualities of a vehicle. Should the slip angle of the rear be less than that of the front, the curve of the vehicle path is towards the off-side or direction of W and represents the condition of *under-steer*. Centrifugal force is in this case opposed to and tends to balance the side force.

If the rear slip angle is greater than the front, the vehicle travels in direction of arrow B, centrifugal force from this curvature being added to the original side force. This condition is termed *oversteer*. The side force W is produced by any uneven quality in the road surface and in resistance to this load the tyres exert an equal and opposite side force which causes them to drift sideways. The phenomena resulting is evidenced by the fact that an oversteering vehicle when negotiating a bend above the certain critical speed of the vehicle for straight running will tend to run into its turn, resulting in a requirement to *reduce* the lock of the road wheel.

Understeering vehicles, on the other hand, when put into a turn, endeavour to straighten up and require constant *increase* of lock as the speed on the bend is increased.

Oversteering is obviously the more dangerous condition, and several practical remedies are effective if carried out to the correct degree. Amongst these may be noted:

(1) Examining the rear tyre pressures and increasing the pressures to reduce the slip angle.
(2) Reducing the pressure of the front tyres to increase the front slip angle.
(3) Reducing the weight on the rear axle (remove passengers to front seats or discard heavy equipment).

*Factors affecting oversteer :*

(1) Weight distribution relative to tyre pressure.
(2) Overturning couples affecting cornering power. (If excess couples at the front the tendency is to greater understeer.) This implies that a low c.g. corners better than a high c.g.
(3) Geometrical errors in steering linkage.
(4) Length of wheelbase (the longer the wheelbase the smaller the curvatures).

### *Low-speed Wobble*

This phenomenon depends upon several contributing factors—(a) castor angle, (b) frame flexibility, (c) wheel mounting, (d) front end weight, and (c) vehicle speed. The reduction of castor angle and the application of friction to the steering joints are two sources of investigation in eliminating this condition, whilst attention should be devoted to stiffer springs and tyre pressures.

### *High-speed Wobble*

The " tramp " is mainly occasioned by gyroscopic action; rocking of the axle longitudinally causes the wheels to rock about their king pins. The front-axle oscillation takes place when the natural frequency of the axle coincides with the speed of rotation of the front wheels. It can build up to alarming proportions when transverse oscillation of the axle produces angularity in the road wheels. Castor action then acts and by precessional torque or self-righting power the wheels are forced to turn in the opposite direction, which again is assisted by the axle movement. Continuation of the process occurs throughout its speed range. More rigid front-axle suspension and reduced weight of brake assemblies, together with attention to the stiffness of the frame front end, should be investigated in overcoming this difficulty.

With regard to the actual steering-box mechanism, types have developed considerably during the past twenty years. The original screw-and-nut type became generally displaced by the worm and sector, which in turn was followed by the cam and lever with fixed stud, and later with the bearing-mounted stud. This type is still popular, as also is the worm-and-roller development. An interesting

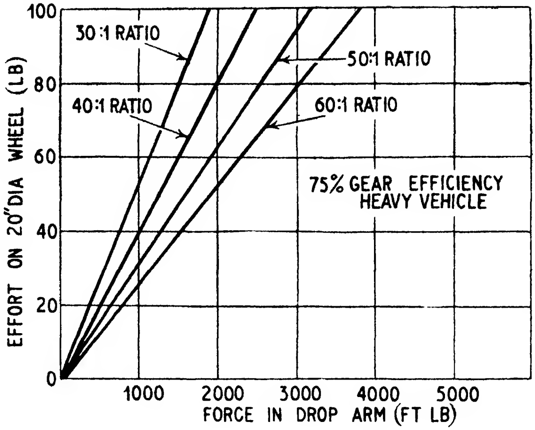



Fig. 30—Steering ratio (heavy classes)

design is that which incorporates a train of balls between the nut and the screw, thus acting as a screw and nut with a rolling instead of a sliding motion between the parts.

The drop-arm gear sector has teeth cut at an angle to the transverse axis of the drop-arm shaft, whilst to mesh with these special teeth the worm nut is placed at an inclined position in the gear. The teeth on the rack are also non-standard, being cut higher at the centre than the end teeth. This is effected in order to provide slight backlash at the ends of the travel when all backlash has been taken out at the centre of the stroke by adjustment. Adjustment is provided by a screw which moves the sector along its own axis.

In both the nut and the worm are ground helical grooves into which are fed the recirculating balls. The nut is fitted with two ball guides in order to avoid running the balls out of the end of the nut, an additional function of the guides being to deflect the balls from the groove near one end of the nut and to guide them diagonally across the outer face of the nut, returning them to the groove at a point near the nut centre. The balls are thus confined within two closed circuits, one at each end of the nut.

The method of assembly of the recirculating balls is of interest. The shaft is placed horizontally and the worm nut positioned so that the ball holes are disposed facing upwards. In the G.M.C. version of this gear there are sixty-six balls in circulation, thirty-three in each circuit; twenty-three of these are placed in the nut and ten in the guide. The two halves of the guide are held together and placed in the worm nut. The second circuit is completed in a like manner and the guides clamped up tightly.

Through these and other developments efficiencies have increased from the old 25 per cent to a figure which now approximates 70 to 90 per cent.

From experimental results which have been obtained in connection with steering-wheel torque against drag-link pressure and axle weight, it has been considered that a hand pull on the steering wheel should not exceed 30 lb for passenger cars in motion, but in achieving this figure care should be exercised that the steering ratio is not decreased

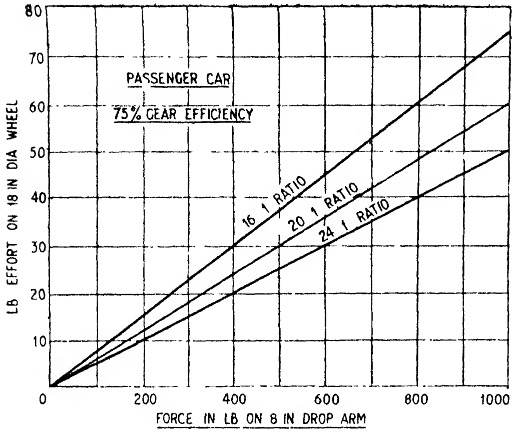



**Fig. 31—Steering ratio (light chassis)**

too much below 24 to 1, for steering must be rapid enough to enable skid recovery and sharp turns to be made. On a basis of 75 per cent gear efficiency curves have been plotted for steering-wheel effort against drop-arm force for the passenger car and heavy vehicle; the gear ratio for the latter naturally being lower, since the axle weight is greater and vehicle speed smaller than that appertaining to the passenger car, whilst manœuvrability at speed is not so necessary. Static steering torque is high on the heavy vehicle, in some cases exceeding 6,000 ft.lb with an axle load of 10 tons, as compared with 600 ft.lb for a heavy passenger car. The tests were made on dry concrete. There are, of course, occasions when this condition is encountered, and it is not considered desirable that the hand-wheel pull should exceed 75–80 lb. It should be reduced if possible. Upon such vehicles power steering is usually employed and this subject is dealt with later in this chapter (see Figs. 30 and 31).

The reversibility of the modern steering gear and its tendency to transmit road shocks to the steering wheel is a serious problem with high axle weights, and the reversed efficiency of the mechanism from steering wheel to drop arm becomes of the greatest interest (see Fig. 32).

If R = radius of steering wheel
r = radius of drop arm
A = angular velocity of the gear

then the forward velocity

$$\text{ratio} = \frac{RA}{r} = V_R$$

and for reverse motion

$$\text{the ratio} = \frac{1}{V_R}$$

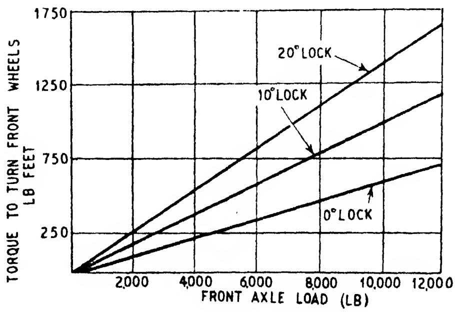



**Fig. 32—Steering torque/angle of lock**

whilst if $W$ = pull on steering wheel rim forward
$W_R$ = pull on steering wheel rim reverse
$W_D$ = drop arm weight
$E_F$ = forward efficiency
$E_R$ = reverse efficiency

pull on drop arm $W_D = WV_RE_F$

and in reverse $= W_R = \dfrac{W_D E_R}{V_R}$

Forward efficiency per cent $E_F = \dfrac{W_D}{W\,V_R} \times 100$

Reverse efficiency per cent $E_R = \dfrac{W_R}{\dfrac{W_D}{V_R}} \times 100.$

The friction attendant in the gear is not, as so often stated, unaltered when in reverse, but the relation between forward and reverse efficiency is dependent upon the forward efficiency and the ratio of the gear. For all ordinary ratios the gear reverses at a forward efficiency of less than 50 per cent. In most gears a change in friction occurs when the input torque is reduced and the gear tries to reverse. It is desirable that the effect of friction experienced is greater when the gear is in reverse.

If such a condition could be realised then we should have controlled the reverse efficiency. There is, however, a limit to the natural reverse efficiency, or in other words the degree of natural friction control, which can be expected if the gear has similar coefficients of friction on all parts for both forward and reverse motions. If $P$ = input torque lb, then the output without friction is $PV_RE_F$, whilst the loss from friction is $PV_R - PV_RE_F$ or $PV_R$ $(1 - E)$ lb. The gear would reverse under loading of value $PV_RE_F$ and without friction the output would be $PE$. If the same friction is experienced in reverse then the output is $2PE - P$ and the efficiency in reverse is $E_R = \dfrac{2E - 1}{E}$.

Suppose, therefore, a forward efficiency of 85 per cent is desired, then the lowest reverse efficiency we can obtain is $\dfrac{1\cdot70 - 1}{0\cdot85} = \dfrac{0\cdot70}{0\cdot85}$ or 82 per cent approximately and this is the limit of natural reverse efficiency.

If, therefore, the reverse efficiency is controlled a reasonably low efficiency could be maintained over the whole range, which would ensure that any small forces emanating from castor action of the front wheels would be sufficient to move the front wheels, whilst heavy forces from the wheels would be operating against a low controlled reverse efficiency. The worm and worm-wheel gear affords an example of what could be effected by the introduction of a one-way friction control operated by thrust at each end of the worm, which, when steering to the offside, produces thrust which may be forward for clockwise steering wheel rotation and rearwards for anti-clockwise wheel movement; but when the drop arm is the medium through which the loads are transmitted to the steering gear the opposite is the case and the thrust on the worm is rearwards for clockwise wheel movement.

Whilst the manual gear is compact and would appear satisfactory for the lighter passenger car, it has limitations for the heavy vehicle which can be overcome by the addition of power steering.

## POWER STEERING

In the interests of safety it is considered essential to retain complete manual control, otherwise in the event of power-gear failure serious consequences would accrue. Moreover, for numerous reasons such features as directional stability, self-centreing properties, and feel, should be retained.

One of the main problems of power steering is backlash on account of the inherent time lag which occurs in any servo. This lag produces the feel of backlash in the steering and must be avoided as much as possible since its presence introduces the possibility of wheel wobble and error in accurate control. This lag can, of course, be partially controlled. There appear to be several basic types current in America—(a) the vacuum, (b) compressed air, (c) hydraulic, and (d) electric. These mostly are in the booster group, which can be added to existing vehicles with little modification to the manual gear. Then there are the integral hydraulic types by Ross-Bendix and Saginaw-Bendix.

In none of these types do the manual gear internals carry more than the steering wheel loads, but by reducing the necessary load application on the wheel the power gear actually lessens the stresses in these parts, whilst in the case of the booster gears the shock loads are transmitted to the chassis structure direct.

In choosing the type of gear to be adopted several factors of a general nature should be considered; and with regard to the compressed-air type, if the vehicle is fitted with air brakes, the brake system may be utilised for the supply to the steering gear and this eliminates the necessity for additional extraneous parts such as pump gear, piping, etc., whilst the difficulty of avoiding loss of working fluid is non-existent.

If the brake gear is hydraulically operated, this would appear to suggest the fitting of one of the hydraulic types, either booster or integral hydraulic. It should be appreciated that a hydraulic fluid should give more positive control and probably more accuracy on account of its quality of non-compressibility. In addition, there is not the possibility of condensation in the pipes and therefore the question of freezing in low temperature is reduced ; whilst, finally, the apparatus is smaller since it operates at higher

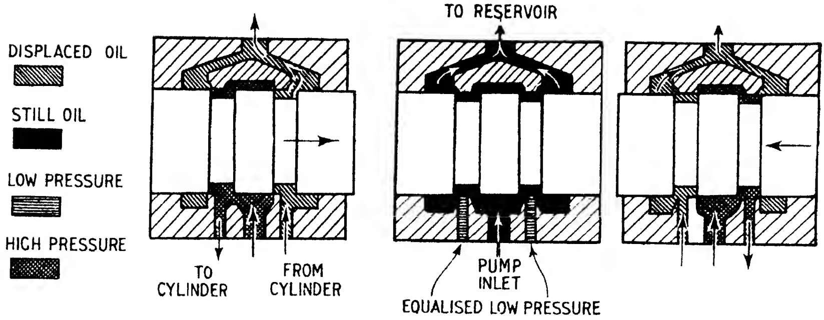



**Fig. 33—Ross-Bendix control valve**

working pressures. A short description of some of the various types is of interest.

The Ross-Bendix integral hydraulic gear conforms to legal requirements insomuch as the steering is at all times under direct manual control and, should the hydraulic assistance fail at any time, the mechanical linkage is unaffected. The gear consists essentially of a Ross cam-and-lever steering box, an engine-driven pump, a control valve, and power cylinder. An oil reservoir is also incorporated. The power cylinder is bolted to the upper part of the gearbox immediately above the steering column, whilst an extension of the lower end of the wormshaft carries a suitably-housed control valve. Two plain bearings carry the lever spindle and an upward extension carries a fixed cylindrical stud engaged in a slot which is cut in the power-operated block. This block rides in guides parallel to the axis of the worm.

The pump is driven from any convenient position on the engine and has a capacity of 3·75 gal/min at 1,000 r/min. The operating pressure is 750 lb/in$^2$, but the built-in relief valve is set to 800 lb/in$^2$ $\pm$ 75 lb/in$^2$.

When the worm is operated through the steering wheel, the lever spindle is partially rotated, bringing the hydraulic system into operation as soon as the effort at the steering wheel exceeds the pre-loading of the centreing springs of the control valve. This hydraulic control is effected through a spool-type valve which is mounted on the lower end of the wormshaft extension. The valve housing provides the oil passages through which the fluid either flows or is checked according to the position of the spool valve along the axis of the wormshaft (see Fig. 33).

Movement of the spool axially is located in fixed relationship to the worm, and movement is restrained by the fluid pressure against the plungers and by the pre-loaded centreing springs. Rotation of the worm compresses the springs between the adaptor flange and the opposite thrust washer in one direction, and the cover and its opposite thrust washer in the other. When the steering effort overcomes the centreing effect of these springs, the spool moves to control the fluid flow by cutting off the return flow on one side and to terminate the flow to the other. The fluid pressure builds up rapidly, and the fluid applies a thrust to the lever of the steering gear via the piston displacement which is, in turn, caused by the fluid flow to one end of the power cylinder when the pressure has built up. The fluid at the opposite end of the cylinder is returned to the reservoir.

Full pressure is obtained by only a very small axial movement or travel, a matter of thousandths of an inch. Shock loads from the road wheels subject the worm to thrust in the opposite direction from that applied by normal steering effort, and hydraulic power opposes instead of assists the movement of the wheels. Reference to Fig. 33 will illustrate the fluid flow through the control valve for right- and left-hand cornering.

The Bendix-Westinghouse mechanism incorporates two control valves attached to the cylinder. Between the operating linkage and the control valves a certain slackness is allowed, which takes the form of a clearance between the boss of the drop arm and its pivot; when this play is taken up by movement of the steering wheel the valve is operated and pressure is applied to the appropriate end of the power cylinder. Depending upon the direction of the steering wheel, so the rocker arm allows the correct air flow to the piston, which, when acted upon, follows up the steering-wheel movement. Upon straightening up again the valve is brought back to neutral, having first sealed its end of the cylinder from atmosphere, and admits air under pressure, after which the piston restores the play in the linkage.

The Vickers hydraulic booster is somewhat similar in principle, utilising slackness in the linkage as in the Westinghouse, but in this case the valve is integral. Power is supplied by vane pump up to 1,000 lb/in$^2$, although the only effort required from the driver is that which is required to overcome the resistance of the centreing springs and the friction in the gear, as there is no fluid reaction against the valve. This gear does not transmit road shocks to the driver, and in consequence the sense of " road feel " is lost; neither does it possess the quality of self-centreing when coming out of a turn. The gear, regardless of the manual gear characteristics, becomes virtually irreversible. The booster is attached at the rear end by bracket to the frame side member, its forward end in link with the drop arm. Flexible hose connections run from the valve in the booster body to the overload relief valve, whence the circuit is continued to the pump. The return pipes from pump to tank complete a compact layout (see Fig. 34).

Since the pre-loaded centreing springs maintain the valve in neutral position, until steering resistance is encountered, a condition is established below which the gear acts as a manual unit, when the driver has the feel of the road, and the gear is self-righting. Above this condition when the power cylinder is in action the proportion

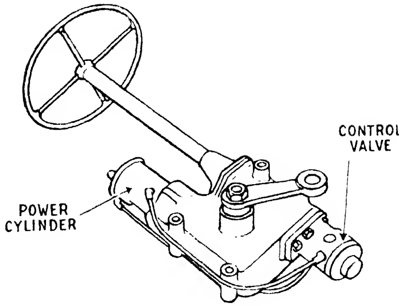



**Fig. 34 —Arrangement of steering booster**

of hand-wheel effort to drop-arm torque is maintained through fluid pressure on the reaction plungers. The centreing springs have a great influence upon directional stability and feel. There are two types of piston valve, utilising in one case two opposed springs to maintain the valve in centre, and in the other the pre-loaded springs.

The pre-loaded type are usually arranged so that as the steering-wheel movement moves the valve away from centre, it is necessary to exert a force to overcome the amount of pre-load in the spring—as one is compressed the other remains at rest. It is therefore possible to determine the required pre-load for a requisite load on the steering wheel to move the valve. With this pre-load, of course, it is possible to move the front wheels of the vehicle without moving the valve, which means that the vehicle can be maintained in its straight-ahead direction and the backlash or feel is that of the normal manual gear. The steering gear is, under these conditions, reversible, and the feel of the gear maintained, whilst the effect of castor action is felt upon the vehicle when coming out of a turn. These are desirable features.

The usual approximate ratio between steering-wheel rim movement and valve movement for the heavy vehicle is 100 to 1 for, say, a steering ratio of 30 to 1. This, of course, depends upon the helix angle of the gear and the load, and so a valve port opening of 0·010 in will require a rim movement of 1 in to close the valve, or in terms of drop-arm movement, one-tenth to three-tenths of a degree. These figures are approximately halved in the case of the passenger car. It will be appreciated, therefore, that very small front-wheel angular movement (which is usually less than that of the drop arm) is required to close or open the valve. On the basis of the 100-to-1 ratio and assuming the heavy-vehicle steering-wheel rim centreing load at its usual 5 to 9 lb, the pre-load of the centreing springs is 500 lb to 900 lb.

There are, of course, linkage losses and friction to be deducted which lighten this load. The valve, being maintained by the springs in its normal centre position, is when deflected always under the influence tending to bring it back to centre, and this tendency anticipates the function of follow-up from the power unit, by virtue of the desire of the valve to return to centre faster than the power unit will permit. This prevents the overshoot and any oscillation in the system providing the force in the spring is sufficient to move the valve at greater speed than the power unit moves under the oil flow available, and to overcome the inertia in the manual elements. Any movement of the valve away from centre unbalances the oil flow through the ports with consequent building up of back pressure on the pump. The reaction plungers, being in connection with the pump-line pressure, require additional force to move them. This force is proportional to the line pressure and it is this reaction which the driver experiences at the wheel.

*The pump size* in a hydraulic gear is of major importance. It must be capable of furnishing sufficient power to steer at an adequate steering speed when the engine is idling or at the engine speed which would correspond to a vehicle speed of 8-10 m/h. Slip in the pump must be taken into account at maximum working pressure and of fluid at a temperature of 160° F. *Steering speed* is represented as the number of seconds required to swing the wheels from lock to lock. From passenger-car experience 4 seconds is adequate, which represents 20° per second. An increase in this figure for heavy vehicles is sufficient at 6–8 seconds or 10°–15° per second (see Fig. 35).

Any effort by the driver exerted at the steering-wheel rim is additional to the steering effort provided by the power cylinder; thus if the effort at the wheel rim is 60 lb and steering wheel 22 in diameter, overall ratio 30 to 1, then with an efficiency of 75 per cent, 1,236 lb. ft is obtained at the drop arm, but due to friction and other losses the driver's effort must produce 1,650 ft. lb torque. It is almost impossible to assess the manual power output which can be maintained during a wheel swing, although it is suggested that a maximum would be 0·5 h.p. Reference to Fig. 35 shows this to be points A, B and C on the graph lines, which represents the maximum effort exerted by the driver. Now assume that the driver exerts an average pull of 30 lb on the wheel rim, which is one-half that previously mentioned, the points on the lines of Fig. 35 move to D, E and F. From this it is therefore clear that, knowing the front-wheel torque necessary, the remaining power required must be supplied by the power medium, since it indicates the limitations between hand and power control. Fig. 35 should, of course, be corrected for efficiency in any actual calculation. The mechanical efficiency of the

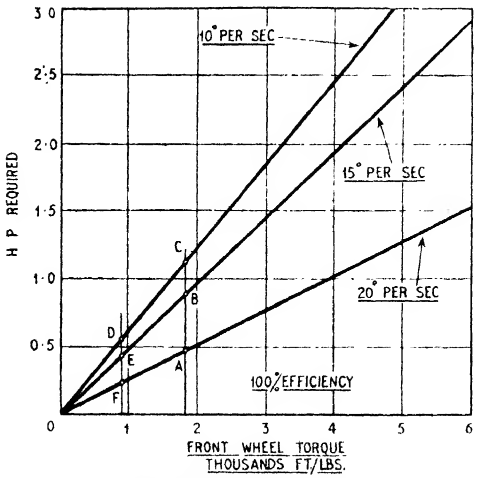



**Fig. 35—Steering speed**

power system averages 85 per cent, whilst it is constant throughout the pressure range.

The flow of oil through the control valves requires consideration as back pressure is exerted, depending amongst other things upon oil viscosity, shape and area of valve port, and any increase in back pressure may result in overheating of the oil and consequent power loss. Experience has indicated that for the heavy vehicle a velocity of oil through the pipe line should not exceed 10–15 ft/sec and 25–35 ft/sec through the valve port. It is also recommended that with a large pump the intake velocity between pump and tank should approximate 5 ft/sec.

## CHAPTER 3

# VEHICLE BRAKING AND PERFORMANCE

BEFORE dealing with the design of actual brake details, a few basic formulae are set down which may be considered as relative to overall vehicle performance; whilst they are essential for preliminary calculations, it may be found desirable to make adjustments, in view of detail considerations which follow.

## INITIAL CALCULATIONS

*Maximum deceleration for a vehicle braked on all wheels* and where the force on each wheel bears the same relation to the load supported by that wheel:

$\delta = g\mu$ where $\mu$ = coefficient of friction between tyre and road.

*For a vehicle not braked on all wheels:* Only that portion of the load taken by the braked wheels is available for road adhesion.

$$\delta = g\mu \frac{w}{W}$$

where W = total weight of vehicle
w = load borne by braked wheels.

*Weight transference* at maximum deceleration:

$$w = \frac{W\mu h}{L}$$

where w = weight transfer from rear to front axle
W = static weight on rear axle
h = height of c.g. above ground
L = wheelbase.

*Total braking force required* for given deceleration:

$$P = \frac{W\delta}{32{\cdot}2}$$

where W = load supported by braked wheels (lb)
$\delta$ = desired deceleration (ft/sec/sec)
P = braking force required at road surface.

*Total lining pressure* for given deceleration:

$$p = \frac{W\,D}{32{\cdot}2 \times \mu f \times d \times K}$$

where W = load supported by braked wheels
D = effective diameter of tyre
$\mu f$ = coefficient of friction of brake liners
d = diameter of brake drum
K = ratio of $\frac{\text{r/min of brake surface}}{\text{r/min of road wheel}}$

*Hydraulic Brakes*

Line pressure should not greatly exceed 1,000 lb/in.$^2$
*Area of cylinder required :*

$$A = \frac{RD}{4p\mu\, Kd}$$

where A = area of wheel cylinder (sq. in)
R = retarding force (lb per wheel)
D = effective diameter of tyre (in)
p = line pressure (lb/in$^2$)
$\mu$ = coefficient of friction of brake lining
d = diameter of brake drum (in)
K = constant.

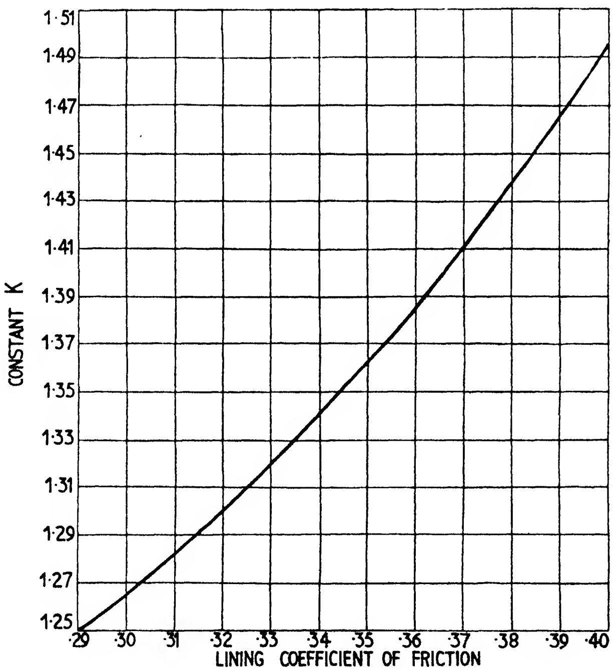



**Fig. 36—Determination of constant K**

K depends upon the servo characteristics of brake shoe layout and coefficient of friction between lining and drum.

*Maximum pedal ratio :*

$$P_R = \frac{Ap}{P}$$

where A $=$ area of master cylinder
p $=$ line pressure (lb/in$^2$)
P $=$ pedal pressure (lb)
$P_R$ $=$ pedal ratio.

*Total leverage necessary :*

$$\frac{[2P_R\,(\,(RA) + (R_1A_1)\,)\,]}{A^2}$$

where $P_R$ $=$ pedal ratio
R $=$ cam-to-lever ratio (front)
$R_1$ $=$ cam-to-lever ratio (rear)
A $=$ area of front cylinder
$A_1$ $=$ area of rear cylinder
$A_2$ $=$ area of master cylinder.

### *Dewandre Vacuum Servo*

The Dewandre system comprises a connection from the engine induction pipe (petrol) or exhauster (oil engine) to the cylinder. The depression in the induction pipe causes the piston to travel in the cylinder. Interposed on the vacuum side is a valve operated by balance lever L, the valve being closed when L is in neutral position. When the brake pedal is depressed, the balance lever rotates in the direction noted by arrow B, which opens the valve and allows suction on the piston. The piston pull being stronger than the pedal pull P, the lever rotates in the direction of arrow A, so closing the valve. The ring stop assumes its central position and equilibrium is produced, the condition being P + F = M up to the limit of vacuum. When the pedal is released, the ring stop travels to the other side, opening the air valve or release and destroys the vacuum. The servo therefore provides a multiplied proportional assistance to the limit of the depression acting upon the piston, which for calculation

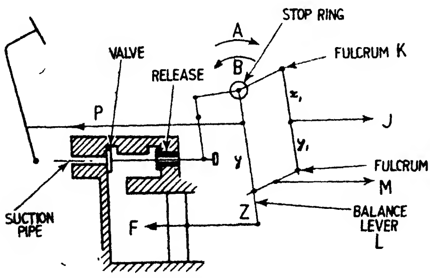



**Fig. 37—Dewandre system**

purposes may be assumed to be a minimum of one-half atmosphere.

A certain point is reached, depending upon the amount of induction depression and the pedal effort, whereupon further assistance or multiplication from the servo ceases, the effort being transmitted direct to the brakes. When the servo becomes inoperative, as example when the engine stops, the levers L & K work as one. Fig. 38 illustrates this effect and the following gives the formulae for Dewandre servo assistance—

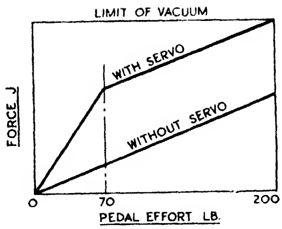



Fig. 38

*To the limit of induction depression :*

(1) $$J = P\left(1 + \frac{y}{z}\right)\left(\frac{x_1 + y_1}{x_1}\right)$$

*When the depression is zero :*

(2) $$J_1 = P\left(\frac{x}{x_1}\right)$$

*The multiplication factor is therefore :*

(3) $$\left(1 + \frac{y}{z}\right) \times \left(\frac{x_1 + y_1}{x}\right)$$

## BRAKE-POWER RATIO

It is generally realised that the ratio of braking power between front- and rear-wheel brakes should be arranged so that the front wheels should be short of the skid condition when the rear wheels are on the point of skidding. This brake-power ratio depends upon—

(1) Leverage to the various brakes;
(2) Ratio of drum to camshaft torque, which in turn depends upon shoe design, anchorage, and lining friction; and
(3) Weight distribution and weight transfer during braking.

If P = percentage by which the front wheel brakes are short of skid condition when the rears are on the point of skidding, any reduction in the ratio of $\frac{W_T}{W_R}$, $\frac{h}{m}$, and $\mu_1$ will obviously reduce P since they are all functions of weight transference during braking.

It is desirable that P should not become negative until a very low value of road-tyre coefficient of friction has been reached. If the front wheels, however, are locked and steering control minimised, the vehicle will still continue motion in an approximate straight line; whereas if the rear wheels only are locked, not only is steering control lost, but the vehicle will in all probability turn completely round unless very skilfully managed. On slippery surfaces, therefore, it would appear desirable to allow all wheels to lock simultaneously, or in other words provide for a 50–50 ratio for light pedal pressures, increasing to, say, 60–40 for heavy pedal loading. This of course can be accomplished by providing for " roll off " of the rear brakes in the linkage geometry.

In further explanation of this ratio, assume that 1,500 lb is necessary as the force at the effective tyre radius to retard the vehicle and 1,200 lb only is actually applied when the rear wheels are on the point of skidding, then the safety factor is 25 per cent or $P = 100 \dfrac{(F - P_F)}{P_F}$.

Taking all the factors into account upon which this ratio is concerned

$$P = 100\left[\frac{W_T}{W_R}\left\{\frac{\mu_1 h}{m}(R+1) + R\right\} - (R+1)\right]$$

where $W_T$ = total laden weight of vehicle (lb)
$W_F$ = static weight on front axle (lb)
$W_R$ = static weight on rear axle (lb)
$h$ = height of c.g. laden above ground (in)
$m$ = wheelbase (in)
$\mu_1$ = effective coefficient of friction tyre and ground
$P_F$ = brake power at front wheels
$P_R$ = brake power at rear wheels
$R = \dfrac{P_R}{P_F}$

When $P = 0$ all wheels skid simultaneously and the ratio is given by

$$R_L = \frac{W_R - \left(\frac{\mu_1 h}{m}\right) W_T}{W_F + \left(\frac{\mu_1 h}{m}\right) W_T}.$$

*Weight distribution during braking :*

If $W_1$ = momentary weight on front wheels
$W_2$ = momentary weight on rear wheels
then when rear wheels are just on the point of skidding:

$$W_2 = \frac{\left[W_R\left\{1 + \frac{P}{100}\right\} - \frac{\mu_1 h}{m}\cdot W_T\right]}{1 + \left(\frac{P}{100}\right)\left(1 + \frac{\mu_1 h}{m}\right)}$$

Momentary weight on front wheels:
$W_1 = W_T - W_2$.
The value of $\mu_1$ to produce limiting condition

$$= \frac{W_R - RW_F}{\frac{hW_T}{m}(R+1)}$$

*Deceleration when rear wheels are about to skid:*

$$= \frac{W_R}{W_T} \times \frac{32.2\mu_1\left(1+\frac{1}{R}\right)}{1+\left\{\frac{hW_T}{m}\left(1+\frac{1}{R}\right)\right\}} \text{ ft/sec/sec.}$$

A few observations regarding the design of brake shoe must be made as the type of shoe influences greatly the overall performance of braking. There has been much study and investigation in this direction, but an understanding of the stresses to which the many types of shoe are subjected will render the reasons for certain designs much more apparent. Dealing first with the elementary principles:

## SIMPLE TWO-SHOE BRAKE

where $T_D$ = brake drum torque (lb/in)
$T_C$ = camshaft torque (lb/in)
$F$ = total friction force (lb)
$A$ = lining area (sq. in)
$t$ = lining width (in)
$p$ = pressure at any point
$p_a$ = average pressure

$$\frac{T_D}{2} = 2PR^2t\mu \cos\phi \text{ for each shoe.}$$

$$A = (\pi_1 - 2\phi)\, Rt$$

$$p_a = \frac{2P\cos\phi}{\pi - 2\phi}$$

*For servo shoe:*

Applied couple = pressure couple – friction couple, and by substituting:

$$\frac{T_D}{2} = \frac{W_1 A}{C\frac{(\alpha - \phi + \sin 2\phi)}{4R\mu\cos\phi} - 1}.$$

*For anti-servo shoe:*

$$\frac{T_D}{2} = \frac{\mu R\cos\phi\, W_2 B}{\frac{C}{4}\left(\alpha - \phi + \sin 2\phi\right) + \mu R\cos\phi}$$

but $T_C = a\,(w_1 + w_2)$
so that we may write

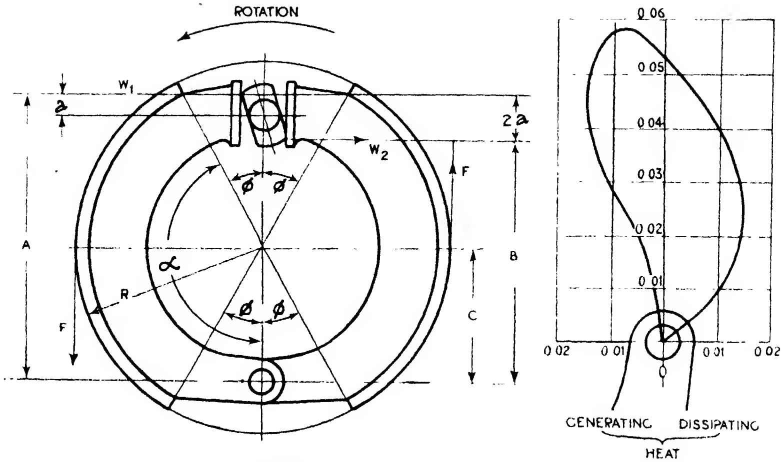



**Fig. 39—Simple two-shoe brake. Graph on right shows fulcrum-pin displacement under heat conditions**

$$T_D = \frac{2T_C}{a(x + y)}$$

where $x = \dfrac{\dfrac{C(\alpha - \phi + \sin 2\phi)}{4R\mu \cos \phi} - 1}{A}$

and $y = \dfrac{\dfrac{C}{4}(\alpha - \phi + \sin 2\phi) + \mu R \cos \phi}{\mu R \cos \phi \; B}$

[*Note:* The power of the brake varies directly with the coefficient of lining friction.]

It will be explained later why one shoe in the simple system becomes servo or self-actuating whilst the other is much less efficient.

## SELF-ENERGISED TWO-SHOE BRAKE

*For primary shoe:*

$T_{D1}$ = drum torque (lb/in)
$F_1$ = total friction force (lb)

$$T_{D1} = \frac{WA_1}{\left(\dfrac{C_1(\pi - 2\phi_1 + \sin 2\phi_1)}{4R\mu \cos \phi_1}\right) - 1}$$

$$F_1 = \frac{T_{D1}}{L.}$$

(for L see previous formula: $W_1$ is obtained from triangle of forces, Fig. 40).

$P_1$ = force due to total pressure

$$P_1 = \frac{F_1 C_1 (\pi - 2\phi_1 + \sin 2\phi_1)}{4\mu C_1 \cos \phi_1}$$

*For secondary shoe :*

$$T_{D2} = \left[ \frac{W_1 B}{\frac{C_2 (\pi - 2\phi_2 + \sin 2\phi_2)}{4R\mu \cos \phi_2} - 1} \right]$$

*Total drum torque* ($T_D$) :

$$T_D = T_{D1} + T_{D2}.$$

In comparing the two systems of brake shoes a basis of similar brake drum clearances is considered the most reliable procedure, with equal pedal efforts and travels. This of course is evident, since if cam force radius is small larger braking forces could be produced, but resulting pedal travel would become impracticable.

In order to place both shoes in contact with the drum, and since the cam for the servo shoe has to move twice as far, half the leverage between pedal and cam on the servo shoe is utilised, as against that of the conventional type shoe. Note that the brake power of the self-energised shoe varies approximately as the cube of the lining friction.

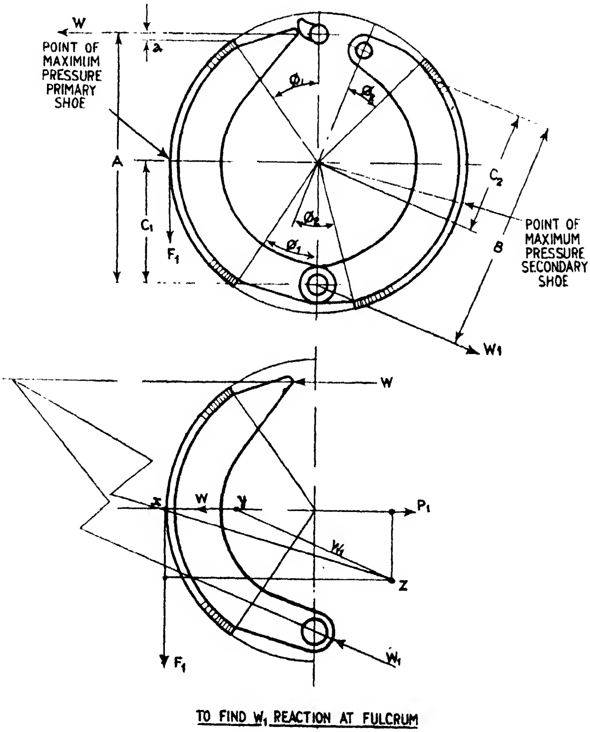



Fig. 40 - Self-energised two-shoe brake

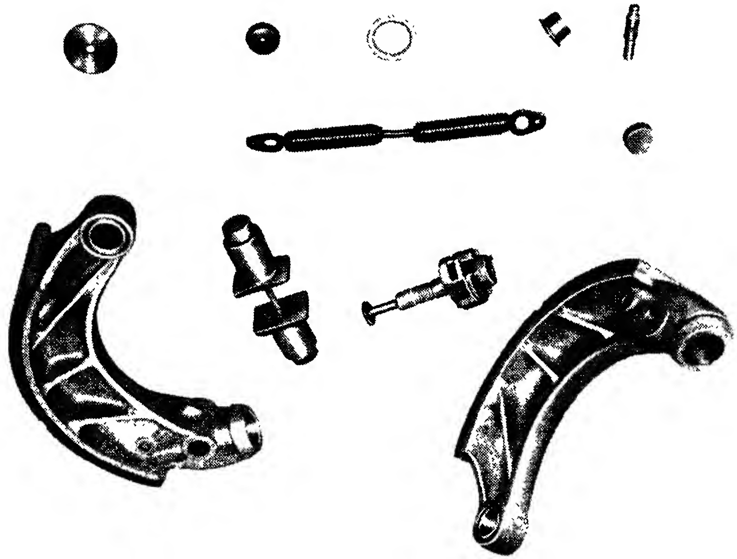


**Fig. 41 – Typical brake-shoe assembly**

For the purpose of obtaining $W_1$ reaction at fulcrum, the value of $F_1$ is required. This force acts at a distance L from vertical centreline (see Fig. 42):

$$\text{Distance } L = \frac{4R \cos \phi_1}{\sin 2\phi_1 + \pi - 2\phi_1}.$$

## BRAKE LINING PRESSURE

It is assumed in producing a pressure diagram that several theories are accepted. These, it should be noted, conform closely to results obtained in practice, and are as follows (see Figs. 42 and 43)—

(1) The lining pressures follow a sine curve as the pressure is proportional to the radial movement of the shoe (towards drum);
(2) Point of maximum pressure occurs at 90° to fulcrum pin centre;
(3) Friction varies directly as the pressure; and
(4) Wear is proportional to friction and therefore to pressure.

The resultant forces of pressure and friction act at a point located at the centre of gravity of the pressure diagram as shown at point X and distance L from drum centre.

If $\beta = 90$, that is when $\varphi_1 = \varphi$, the general formula for L is given by

$$L = \frac{2R \cos \psi (\cos \varphi - \cos \alpha)}{\alpha - \varphi + \frac{1}{2} (\sin 2\varphi - \sin 2\alpha)}$$

and when $\varphi_1 = \varphi$

$$L = \frac{4R\cos\varphi}{\sin 2\varphi + \pi - 2\varphi}$$

where $\psi$ is the difference between $\beta$ and 90° for non-symmetrical linings and

$$N = C\sin\psi$$
$$M = C\cos\psi$$

## LINING DISPOSITION

It will be appreciated that an ideal brake would utilise the whole circumferential surface of the drum and would exert throughout the maximum pressure. If the ratio of lining length to drum circumference be considered it is evident that this could form a basis of comparison between two similar braking systems relative to their respective efficiencies.

The ideal torque would be $\mu PRA$

where $\mu$ = coefficient of lining friction
$P$ = maximum pressure
$R$ = drum radius
$A$ = area of the inner working face of drum.

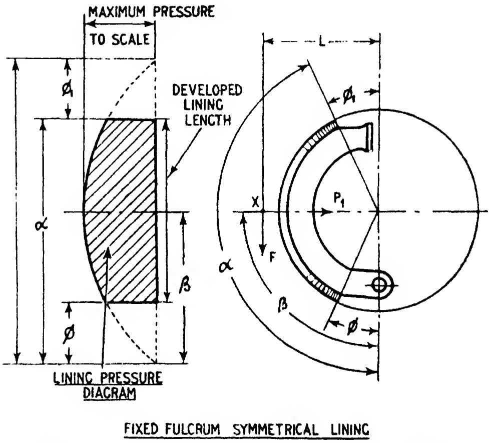



**Fig. 42—Construction of lining pressure diagram**

If K be the symbol denoting this ratio, for any brake then the torque would be

$$T = K(\mu PRA).$$

In a two-shoe brake of simple form, such as that shown in Fig. 39, torque equals

$$T = 2\mu PRA\,(\cos\theta/\pi)$$

and the value for K

$$K = \frac{(2\mu PRA\cos\theta)}{\pi\mu PRA}$$
$$= 2\cos\theta/\pi$$

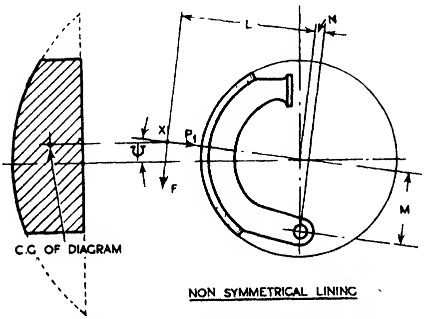



**Fig. 43—Non-symmetrical lining**

If various values of K are plotted against angle subtended by the arc of lining, it will be seen that the value of K does not increase in direct proportion to the increase in length of lining ; for instance, if the lining angle be increased from say 90° to 120°, K increases from approximately 0·45 to 0·55 but if this angle be further increased to say 150°, K increases to 0·62. The conclusion to be drawn from this supports the contention that an arc of lining subtending any angle greater than approximately 120° becomes wasteful of lining material. It should be also appreciated that chatter and jamming are liable to occur with excess lining.

It is perhaps advisable to make clear the term " self-actuation," which, far from being a condition which is peculiar to proprietary brakes, refers to almost every internal-expanding shoe type to some degree. It is self-evident from the fact that the centre of shoe anchorage is placed within the drum diameter, whilst in order to obtain a shoe brake which would not be self-actuating it would be essential that the anchorage be located at an infinite distance from the drum centre, assuming that the mechanism possessed absolute rigidity.

Considerations which are closely allied to the quality of " self-actuation " are—

(1) Uniform contact of the lining with drum face and the correct arc of shoe to provide this condition;
(2) Amount of drum distortion;
(3) Distance of the anchor pin from the drum centre; and
(4) Temperature rise and its variation in the shoe structure.

It is of course realised that, for a given mechanical advantage, the longer the shoe the greater the pedal travel required ; for instance, with the usual 120° of lining as compared with 150°, and for similar clearances, the ratio of travel remains practically constant and is approximately 2 to 1 in favour of the 120° shoe. Moreover, increase in shoe length must have some influence upon the distortion of the shoe at the pivot eye, although the condition is more often a result of overheating. Displacement of the free end is not however permanent, but in conjunction with the temperature variations at different sections of the shoe the effect upon the lining will be obvious.

The expansion and contraction of the shoe periphery during heating and cooling causes the fulcrum to distort in roughly a curve as seen in Fig. 39. It is possible that these displacements, combined with the varying temperatures along the shoe periphery, increase the probability of " chatter " by introducing one of the contributory factors of heel and toe wear of the lining.

For the amount of self-actuation possessed by an unsymmetrical shoe, the actuation constant K can be expressed for a short length of lining at each end of the shoe as

$$K = \mu \frac{R + (D \cos \beta_1)}{D \sin \beta_1} \text{ for the cam end piece, and}$$

$$K = \mu \frac{R + (Dx - \cos \beta_2)}{D \sin \beta_2} \text{ for the pivot end piece,}$$

where $F$ = factor of actuation at any point on the lining
$K$ = actuation constant
$R$ = radius of drum
$\beta_1$ = toe angle of shoe
$\beta_2$ = heel angle of shoe
$D$ = centre of anchor pin
$\mu$ = coefficient of lining friction.

For the shoe to become self-locking the "actuation factor" $A = 1$ or $\frac{1}{K}$. The sum of the two values of K therefore, over a divisor of 2, gives the locking coefficient for the whole shoe, and a $\mu$ is found which must not be exceeded.

An alternative method of calculating this coefficient, which approximately agrees with the previous method, may also be used in the interests of simplicity:

where $BF$ = braking force at radius of drum (lb)
$P$ = operating pressure (lb)
$\mu$ = lining coefficient

then $$P = BF \frac{C\left(\frac{\sin\beta}{\mu} - \cos\beta\right)}{A}.$$

Assuming $\mu$ to be as previously found in the above formula, a ratio can be extracted for P to BF and the locking point is reached when $P = BF \times O$ since $\frac{\sin\beta}{\mu} - \cos\beta = O$.

The frictional properties of the lining change at temperatures above 480–500° F. Temperature therefore becomes one of the predominating factors in brake design, since the main function is primarily to convert kinetic energy of a vehicle into heat sufficiently to cause deceleration. The wear-load factor of the lining is important also, and the calculation of this component is perhaps best considered as a measure of lost thickness for a given power absorption, through a range of suitable temperatures.

Nearly all friction materials possess an allowable unit of power absorption, the value of which is in direct proportion to the product of rubbing velocity and pressure per sq. in or intensity of unit pressure.

If $V$ = rubbing velocity
$P$ = pressure/in$^2$

$$V \times P = \frac{\text{Safe power absorption}}{\text{Permissible } \mu} = K \text{ in/lb or ft/min.}$$

Temperature developed inside the drum periphery may increase the frictional coefficient properties of certain linings of the asphalt bonded type, although it reduces it in others. The rubbing velocity also affects the power absorption value, and it is therefore desirable to reduce the drum diameter to reduce peripheral speed whilst increasing the shoe width as far as possible consistent with dimensional legal requirements, to reduce the intensity of lining pressure.

The fulcrum pin centre may be determined by application of the following formula (see Fig. 44)—

$$d = \frac{R\int_{\beta_1}^{\beta_2} \sin\beta\, d\beta}{K\int_{\beta_1}^{\beta_2} \sin^2\beta\, d\beta - \int_{\beta_1}^{\beta_2} \sin\beta\cos\beta\, d\beta}.$$

There is, however, an inherent inaccuracy in shoe operation with all types of brake utilising the simple or twin fulcrum pin and fixed camshaft. With this type the trailing shoe is only approximately half as efficient as the leading shoe, that is of course the ratio between output and input.

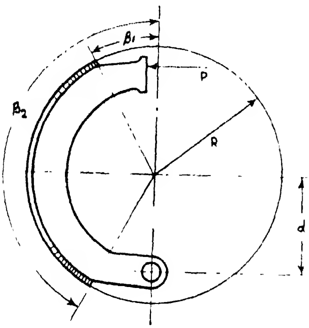


**Fig. 44—Fulcrum pin centre diagram**

The leading shoe is, as previously mentioned, helped by the rotation of the drum to become self-energising, whilst the use of a fixed camshaft imposes a limitation in the power which can be applied at the tip of that shoe. The overall efficiency of this type of brake is therefore low. This difficulty can be minimised by the adoption of a floating cam which is unrestrained as the full separating force can be applied at both shoe tips.

Even so, difficulties are not entirely eliminated, since the leading shoe still performs more work than the trailing shoe and thus the lining wear is severe and unequal. Consequent upon this is rapid increase in lining temperatures and the possibility of brake fade, which in itself may be a serious defect.

In an effort to overcome these disadvantages, the two-leading-shoe type of brake was evolved, in which both shoes are free to centralise themselves in relation to the drum periphery, whilst being held against abutments in the form of an expander and adjuster by spring pressure. Upon force being applied to the toe of the first shoe, contact is made between shoe and drum, whilst at the same time the expander applies force to the vertical arm of bell crank lever A (Fig. 45). This force is transmitted through strut B to the horizontal arm of lever C, the vertical arm of which rests against the adjuster. Reactionary force therefore holds this shoe in contact with the drum and, since the toe

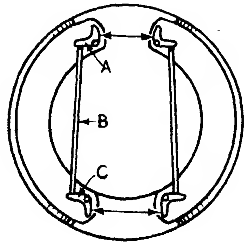


**Fig. 45—Two leading shoe brake**

is not in contact with the fixed abutment, the shoe becomes a leading shoe. In order to provide for any emergency braking in reverse gear, the links and levers are duplicated as shown in the diagram.

## BRAKE EFFICIENCY

The term efficiency is often misconstrued in connection with braking performance as representing a ratio of pedal pressure or physical effort to braking effort produced, whereas the efficiency represents a percentage governed mainly by the coefficient of adhesion between tyre and road surface. This means that to express the coefficient of adhesion as 0·7 limits the percentage efficiency to 70 per cent maximum. Under certain conditions of road surface and tyre tread it is possible occasionally to obtain 100 per cent efficient braking, but generally the maximum is 70 to 80 per cent.

The accepted meaning of 100 per cent efficient is that the brakes produce a vehicle deceleration equal to the downward acceleration due to gravity (g), that is 32·2 ft/sec/sec. The stopping-distance figures calculated on this basis bear no relation to the power required to produce such effect and, whilst the 100 per cent efficiency indicates an expression of performance in terms of maximum possible, they need to be equated against applied pedal pressures and the weight of vehicle before accurate figures can be obtained. It is of course appreciated that when the retarding forces equal the axle weights then skidding takes place, and in order to obtain an approximation of the maximum deceleration which can safely be used a chart should be plotted (Fig. 46) correlating axle weight against deceleration.

Two sets of curves A and B are shown, those marked A representing equal axle weights and coefficient of adhesion between tyre and road of unity. The curve slopes are in accordance with the transfer of weight from one axle to the other at different rates of deceleration. The curves at B are those for an adhesion coefficient of 0·7. It will be seen that for equal braking on each axle the rate of vehicle retardation is 19·5 ft/sec/sec, a figure which also defines the skid point. If, however, the braking proportions are modified to 55 per cent and 45 per cent this deceleration increases to 21 ft/sec/sec.

Referring to curves B, it will be noted that by taking a coefficient of adhesion below unity the deceleration decreases, in this case to 16·5 ft/sec/sec, whilst an increase in braking proportion on the front axle increases the safe deceleration. This curve permits the size of the brake required to decelerate a vehicle of known weight to be calculated. By inserting the value of deceleration ($\delta$) in the expression

$F = \dfrac{W\delta r}{gR}$ the drag or the braking force at the drum can be ascertained.

$F =$ drag
$W =$ vehicle weight (lb)

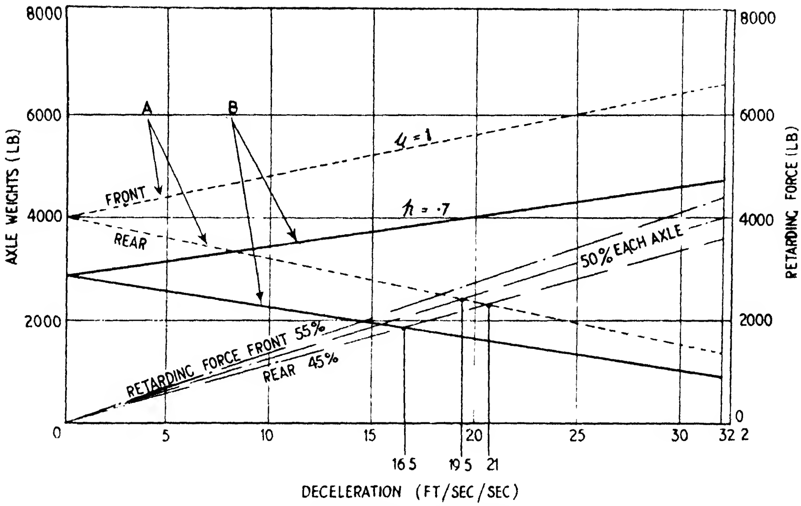



**Fig. 46—Maximum safe deceleration**

g = gravity (32·2 ft/sec/sec)
R = brake-drum radius (in)
r = effective tyre radius (in).

The linings of the shoes will only sustain a certain amount of drag if they are to have long life and it has been generally accepted that 50 lb/in² area is a safe load. The total lining area is therefore

$$\frac{W\delta r}{50gR} = \text{total lining required.}$$

The width should be selected so that it is maximum for surrounding conditions, as not only does a wide lining reduce the intensity of pressure, but it also permits increase in applied power without increasing the rate of wear.

The gradient effect upon brake efficiency and stopping distance occurs mainly on account of the loss of effective axle weight, which of course varies with the gradient and which disturbs the percentage of axle braking. There is in addition a propulsive force tending to increase the vehicle speed which must be absorbed by the brakes. These forces are set out in Table 2. In consequence the braking efficiency is considerably reduced and thus the stopping distance is increased as compared with level road braking in the direct proportion to the equivalent braking efficiency on a gradient. Comparison of such variations are shown in Table 3.

Taking all factors into account and

where W = vehicle weight
$\mu$ = coefficient of road adhesion
a = gradient angle degrees

$$E = \frac{\text{Weight} \times \text{Speed (ft/sec/sec)}}{64{\cdot}4}$$

then the stopping distance of a vehicle of known weight will be

$$S = \frac{E}{W[(\mu \cos a) - \sin a]}$$

The material from which the brake drum is manufactured is one which must have detailed consideration, as not only factors such as tensile strength, impact resistance, machinability and cost are involved, but others of equal importance to the success of the brakes assume proportions which cannot be overlooked. For instance, upon drum material, and material treatment, depends lining wear, drum wear, and also the process of drum scoring. A cursory survey is therefore necessary at this stage of a few of the more widely used materials with regard to these latter phenomena.

TABLE 2

| Gradient | Vertical force per ton of vehicle weight |
|---|---|
| 1 in 4 | 543 lb |
| 1 in 5 | 439 lb |
| 1 in 8 | 278 lb |
| 1 in 10 | 223 lb |
| 1 in 15 | 149 lb |
| 1 in 20 | 112 lb |

0·1 *per cent Carbon Steel Material:* A low-carbon steel has advantages for usage as a brake drum, since it lends itself to deep drawing and can be cold pressed. Many tests have been made in an effort to improve its wearing properties. It has been found that cold rolling to improve the surface hardness has little beneficial effect either on its resistance to wear or the friction lining wear.

TABLE 3

| Braking efficiency on level | Equivalent efficiency on gradient | | | | |
|---|---|---|---|---|---|
| | 1 in 5 | 1 in 8 | 1 in 10 | 1 in 15 | 1 in 20 |
| per cent 70 | 50 | 57 | 61 | 63 | 65 |
| 60 | 40 | 47 | 50 | 53 | 55 |
| 50 | 30 | 37 | 40 | 43 | 45 |
| 40 | 20 | 27 | 30 | 33 | 35 |
| 30 | 10 | 17 | 20 | 23 | 25 |

A similar material having a mild carbonising treatment to a surface depth of 0·030 in and Brinell 210 gives practically negligible wear and scoring under test conditions, and is many times better than the untreated steel, although further test results with increased Brinell showed little further advantage. It would appear that it is not necessarily the Brinell number which controls the wear properties but rather perhaps the micro-structure. This type of treatment also prolongs the life of the friction liners. Regarding the effect of lining area upon scoring of the drum, it has been established that increase in contact area does not effect any change upon this condition. It does naturally reduce the wear of the lining itself, and for this reason is desirable. The lining material for a low-carbon steel drum which provides the most suitable combination with a low-carbon steel drum for both lining life and drum wear should be one of the non-metallic brands.

0·3 *per cent Carbon Steel Material:* With similar brake liners this material in its hot-rolled condition acts similarly to the 0·1 per cent carbon, but whereas the 0·1 per cent carbon material shows no improvement to scoring or wear when cold rolled, the 0·3 per cent material produces a marked improvement. The experiments which have been made, cold rolled the material to a reduction in thickness of 9 per cent and evidently produced a refinement in structure.

0·5 *per cent Carbon Steel Material:* As would be expected, the increase in carbon content shows still further improvement in wearing qualities, and therefore this material, either annealed or hardened and tempered, would be suitable for a drum material.

0·3 *per cent Carbon Steel (Chromium-plated):* A surface of this material plated to a depth of 0·002 per cent and buffed with an Alundum wheel will show improvement over all the other carbon steels with reference to lining life, and the surface wear of the drum is extremely small. The friction coefficient of the lining remains normal. Incidentally it should be remembered that if chromium-plating is to resist abrasion, no undercoating of nickel or copper must be used.

*Cast Iron:* The cast-iron drum is possibly the best. It has many manufacturing advantages besides technical qualifications. Test results have indicated that Brinell hardness number bears little or no relation to wear qualities; even should the material be annealed there is no appreciable difference, probably accounted for by the fact that the increase in graphite content emanating from its free carbon compensates for reduction of hardness. It is quite possible that this graphite, well known for its lubricating properties, may become effective under severe abrasion.

The final selection of material should be made from one of the following range:

(a) 0·5 per cent carbon steel;
(b) cast iron;
(c) chromium-plated steel;
(d) 0·3 per cent carbon steel, cold rolled; and
(e) carbon steel, carburised.

## BRAKE SQUEAL

This phenomena concerns the vibrations of both the shoes and the brake drums, and it becomes more audible with increased pressure.

The initial vibrations are of little importance, but they increase in frequency and amplitude so rapidly that they become incapable of suppression. Initially the movements are believed to be produced by friction tension, and therefore a lining which possesses a high coefficient of friction is more liable to produce brake squeal. A squealing brake is nearly always an efficient one and the phenomena does not occur until the lining has become bedded and the drum polished, in which condition the effectiveness of the brake is maximum. Considerable energy is developed by the heat generated, which increases the power of the brake by no small amount.

Mechanical slackness in the brake shoe assembly should be eliminated if the unpleasant characteristic is to be avoided, whilst if a liner possessing a lower coefficient of friction is substituted, the vibration may be reduced. The design of the brake drum militates against its own structural rigidity and it is therefore very necessary to stiffen the open diameter by a suitable flange so that distortion is reduced to an absolute minimum.

## CHAPTER 4

# THE FRAME

## DESIGN

THE vehicle foundation is the frame and therefore sound design is of major importance. It should be treated as a complete structure, and not merely as two side members joined by a series of cross members. It is subject to many and varied stresses, some of which can be avoided in the design stage if due consideration is given to the fundamentals.

Before discussing the frame structure generally, it is well to consider such points. One of the most important concerns localisation of stress at numerous points through inadequate knowledge of the effect of drilling holes in the members for fitting attachments. This refers to the top and bottom flanges particularly, although too much licence should not be allowed in the vertical section of the side members. In this case it is advisable to spread hole centres and to drill those of larger dimensions as near as possible to the neutral axis.

" Flitching " should not be carried out haphazardly as a sudden discontinuance in length will cause serious localisation of stress with almost certain fatigue failure resulting. No flitch should be added without reference to the stress diagram and as slow a change as possible in stress intensity arranged by grading the ends of the flitch plate. Attachment of fittings must also be carefully considered ; in consequence the effect of drilling a hole is worthy of attention. It is well established that the maximum stress in a plate where the width is large compared with the hole diameter (drilled at the centre of the plate) occurs at the transverse diametral points AA of the hole (see Fig. 47).

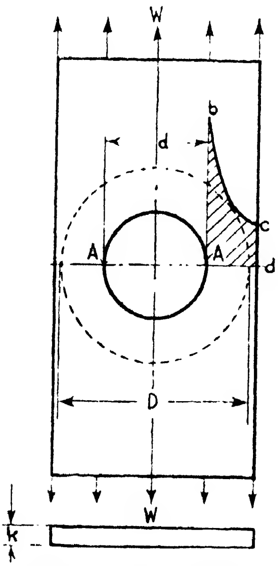



**Fig. 47—Hole drilled in plate (stress distribution)**

The distortion of normal stress distribution at those points is shown represented by the shaded area Abcd, effective on both sides of the hole. The magnitude may exceed 3W, a figure which is affected slightly by the ratio $\frac{D}{d}$, as if D is large compared with d, the outer circumferential stresses are not so greatly affected by the presence of the hole. If however $\frac{D}{d}$ approximates 4 or 3 then the hole exerts a considerable influence on the force distribution acting at circumference D and the magnitude reaches 3·85 W in the ratio $\frac{\text{W maximum.}}{\text{W}}$

In calculating the values of this expression, use is made of the theory of bending in a curved bar, wherein the force per unit length may be assumed to be $\delta$ W sin $\alpha$ at any point on the external diameter D. It is only necessary to consider one quadrant ABCD of the bar (see Fig. 48). Cross section AB is subject to the maximum stress and for equilibrium force F at AB must be

$$F = \frac{\delta DW}{2}$$

and the tensile stress equals

$$f = \frac{WD}{2T}$$

The bending moment (B.M.) at section AB is calculable as AB remains in a horizontal position due to its symmetry.

If a curved bar is bent, stability of cross section may be written as

$$\int^{\frac{\pi}{2}} \frac{da}{M_1} - r \int^{\frac{\pi}{2}} \frac{dx}{F} = 0$$

if the bar is bent by terminal couples only.

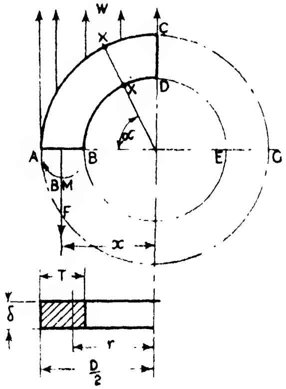


Fig. 48 —Curved bar theory

In the foregoing expression

$M_1$ = B.M. at any cross section
F = longitudinal force in cross section
r = radius of the centre of cross section from the neutral axis and is equal to

$$r = x \frac{T}{\log_n \frac{D}{d}} \text{ for rectangular section.}$$

In the expression for stability of the bar the first portion represents the relation between relative rotation between cross sections AB and its counterpart EG due to bending moment: M, the latter half of the expression, is proportional to the rotation caused by longitudinal force F. A summation of forces at a section XX are M + F

where $F = \frac{\delta DW}{2}\cos^2\alpha$ and M =

$$\text{B.M.} + \frac{\delta DW}{2}\left(1 - \cos^2\alpha\right)\frac{D}{4}\left\{\left(1 - \cos^2\alpha\right) + \frac{T}{2}\cos\alpha\right\}$$

Integrating the expression wherein the values of r, L, and M have been substituted

$$\text{B.M.} = \frac{1}{2\pi}\,\delta WD^2\left\{\left(\pi - 2\right)\frac{x}{D} + \frac{\pi r}{2D} - \left(1 - \frac{\pi}{4}\right)\frac{T}{D} - \frac{3\pi}{8} + 1\right\}$$

Stresses may be calculated from the formula for bending of curved bars, and where $M_S$ is the moment of the section about the neutral axis and S the distance of the point from that axis

$$f_1 = \frac{M_1 S}{M_S (x - S)}.$$

For the section AB and point B, $M_1 = B.M.$

$$M_S = T\delta r$$

substituting

$$f = B.M. \frac{I}{r\delta d}\left(I - 2\frac{r}{T}\right).$$

To complete the stress at the point of maximum intensity B, the combination of $f$ with $f_1$, we get W maximum $= f + f_1$.

Consider then the result of drilling holes in the flanges of a frame member which is subject to tension due to bending. As the ratio of hole diameter to flange width or $\frac{D}{d}$ approaches unity, the stress at the edge of the hole increases, and therefore the possibility of rupture is greater. As an example, stresses have been known to reach 1,500 lb/in$^2$ tension and 1,700 lb/in$^2$ compressive stress at the edge of a hole $\frac{3}{4}$ in diameter in a width of plate 6 in wide: this for an initial applied mean stress of 180 lb/in$^2$ on the cross section. This example was chosen as the width of plate obviates any boundary influence. The stress diagram for this example is shown in Fig. 49.

Rupture may occur through reduction of flange area, which might render the material initially overstressed, or partly due to stress occasioned by vibration and following fatigue. It is for this reason that a fatigue fracture may be observed commencing at the edge of the hole and spreading towards the edge of the plate or flange.

It will be noted that the stress is mainly dependent upon the area of cross section at the edge of the hole, and the practice of beading is therefore desirable where practicable.

It has been shown that care must be exercised in the drilling and selection of hole size, and the effects have been briefly noted, from which it is assumed that perhaps a large number of small holes are more advantageous than a small number of large ones for attaching frame fittings. The bolted fitting, however, will not be permanently satisfactory if the method of bolting and nutting up is not also given serious consideration.

A properly-tightened nut is one which applies to the bolt or stud a load equal to or greater than the external load to be carried in service. If this is assured and the nut tightened against rigid members then the bolt will not fail through fatigue as, regardless of external and fluctuating loads, it experiences no change in stress although the fluctuating loads may alternate from zero to maximum at high frequency. Assuming the diameter is adequate, the load is practically static.

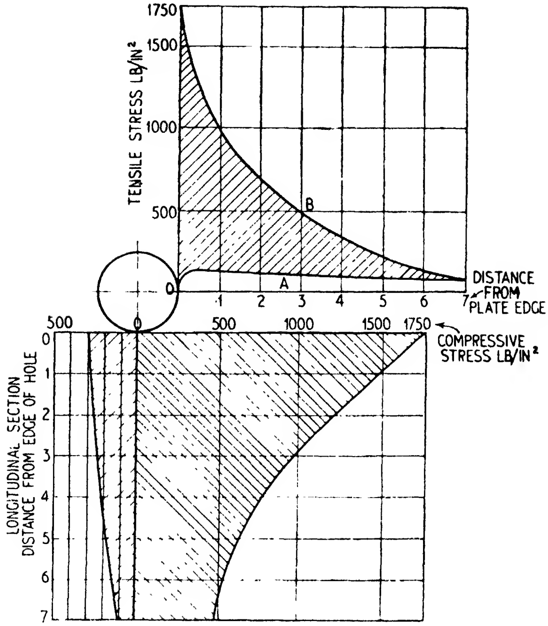



**Fig. 49—Stress distribution in drilled plate**

The fatigue strength of the material decreases as the range of dynamic stress increases and vice versa. The stress range is represented by the ratio R —

$$1 - \frac{\text{Min Stress}}{\text{Max Stress}}$$

which is the difference between the initial load and the maximum operating load. If, however, the bolt is tensioned at a stress lower than the applied load the stress range under repeated loads is increased.

Another point to bear in mind concerns the rigidity of the components to be bolted together. The design of bolted face and bosses must be such that they do not crush any more than is possible and that additional bending is not applied to the bolts through inadequate surface. A bolt-hole boss should be as large a diameter as practicable at its base. Referring to Fig. 50, it will be seen that the two parts bolted together tend to bend about O as a fulcrum; therefore O should be as far from the bolt centre as possible in order to eliminate bending stresses on the bolt. The depth of the boss should provide for as long a bolt or stud as possible.

Consider the case of a short stud attaching a thin flange. The stud when nutted is elastically elongated through its length from near the nut base to the last thread in the studded base. As an example take this dimension to be ½ in. The nut correctly tightened stresses the stud to the appropriate amount. The elongation is then, say, 0·002 in. Now assume that the flange is reduced in thickness by some means, perhaps by embedding or corrosion, by 0·001 in. The stud has

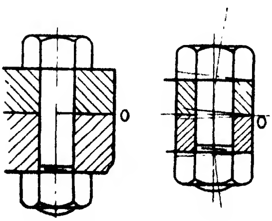



**Fig. 50—Design of bolt boss**

lost one-half its correct tension and fatigue failure will probably ensue.

Compare this with a larger stud under similar conditions. If the $\frac{1}{2}$ in now becomes 2 in the loss in tension is only one-eighth the initial tension, or in other words a loss of 0·001 in from a total elongation of 0·008 in.

In considering the design of bolt or stud it should be appreciated that most bolts fail through fatigue, and therefore any point which may be the source of raising the stress should be eliminated. In this respect the most severe stress inducers are the deep notches of the threads. It is advisable to specify either rolled or ground threads. Rolled threads are better than ground threads and these latter are in turn better than cut threads. This is so because the rolling permits smooth radii and eliminates cutting and tearing, although the fatigue durability of a cut thread can be increased by compressively stressing the material at the root by superficial rolling.

Incidentally, a rolled bolt thread should not be heat-treated after rolling, as the process of rolling often takes advantage of compressive pre-stressing, which is very desirable, and heat treatment, except at low temperature, loses this pre-stress. If it is necessary to carry out such treatment then the threads must be given a final rolling afterwards.

The diameter of the bolt shank should be as small as consistency allows with the required strength. This increases the elasticity of the bolt both in bending and elongation. Large fillets or radii should always be provided at the reduced diameter as it joins the threaded portion. Spring washers may profitably be used providing they support the desired bolt tension within their own elastic range, and do not collapse solidly against the abutments. They have the advantage of increasing the safety of the bolt by reducing its stress range and they also provide somewhat against loss of tension.

It will have been noted from the sections on steering, braking, etc., that many loads will be transmitted to the frame, all of which may be taken care of by suitable modification to the following methods of stressing. If due care, however, is not exercised in including all these loads the final frame weight will be excessive, and this fact in itself assists again in increasing stresses within the structure. In the past the frame structure intentionally possessed a certain degree of flexibility which was considered desirable to absorb the continuous shock loads tending towards ultimate fatigue failure. More recently, and with the advent of flexible unit mounting, such a feature becomes less important, whilst the reverse opinion assumes larger proportions.

Through flexible unit mounting the frame loses much of the assistance it previously received towards rigidity, although by the modern system vibration and resonance are much reduced. Front-wheel brakes affect the magnitude of frame stress and a rigid front end is absolutely necessary. Generally, rigid cross members must be located at the points of attachment of each road-spring bracket and if possible should pick up the bracket fixing bolts. It is at these points that weaving stresses are transmitted to the frame. Suitable intermediate members should be arranged as nearly equidistant as convenient.

The number of holes should in all cases be reduced to a minimum by utilising as far as possible existing bolts and holes for attaching other adjacent components. Attachment faces should be spread over as wide an area as consistent with good design in order to avoid localisation of stresses.

Before passing to the standardised method of comparative frame stressing, there are several points to note in connection with general design:

Avoid sharp upsweeps ; run in bends with the largest radii possible.

Changes in side-member profile should be gradual. When changing from deep to shallow section a good plan is to consider the change as if the section were symmetrical about its neutral axis. The side members should be reinforced by a suitable design of internal stiffener to take all buffer loads (as at front and rear axles).

Do not permit large-diameter holes to be formed near the top or bottom of the vertical surface of the channel pressing. The ideal location for such holes is on the neutral axis of the section, since along this line they have less effect upon the modulus of resistance.

All frame dimensions should bear definite relation to a datum face or line. The frame front is considered the most appropriate edge for longitudinal dimensions, which should be so arranged that they may be quickly located from that edge. Subsidiary dimensions should be positioned from the nearest controlling point. Bracket-attachment holes should be located on the frame section bearing definite location to the fit of the bracket in or on the frame; that is to say if a bracket fits inside the section then the key holes are dimensioned from the inside, and vice versa. All frame pressings are dimensioned from the inside of the section.

As a guide to the general method of frame stressing the following sequence should be followed and, whilst the calculations do not by any means exhaust all conditions of frame loading or afford a complete stress investigation, they do furnish a comparative basis with other known models (see Fig. 51).

It should be noted that for the purposes of calculation loads which tend to turn the chassis in a clockwise direction are considered to be positive; those tending to turn anti-clockwise to be negative.

(1) *Static loading for chassis only :*

(i) Set down a diagram of frame side member and springs, disposing unit weights as designed, and estimate the unsprung chassis weight: 50 per cent of this weight should be applied to one side member.

(ii) Compute spring reactions at all spring attachment points by taking appropriate moments.

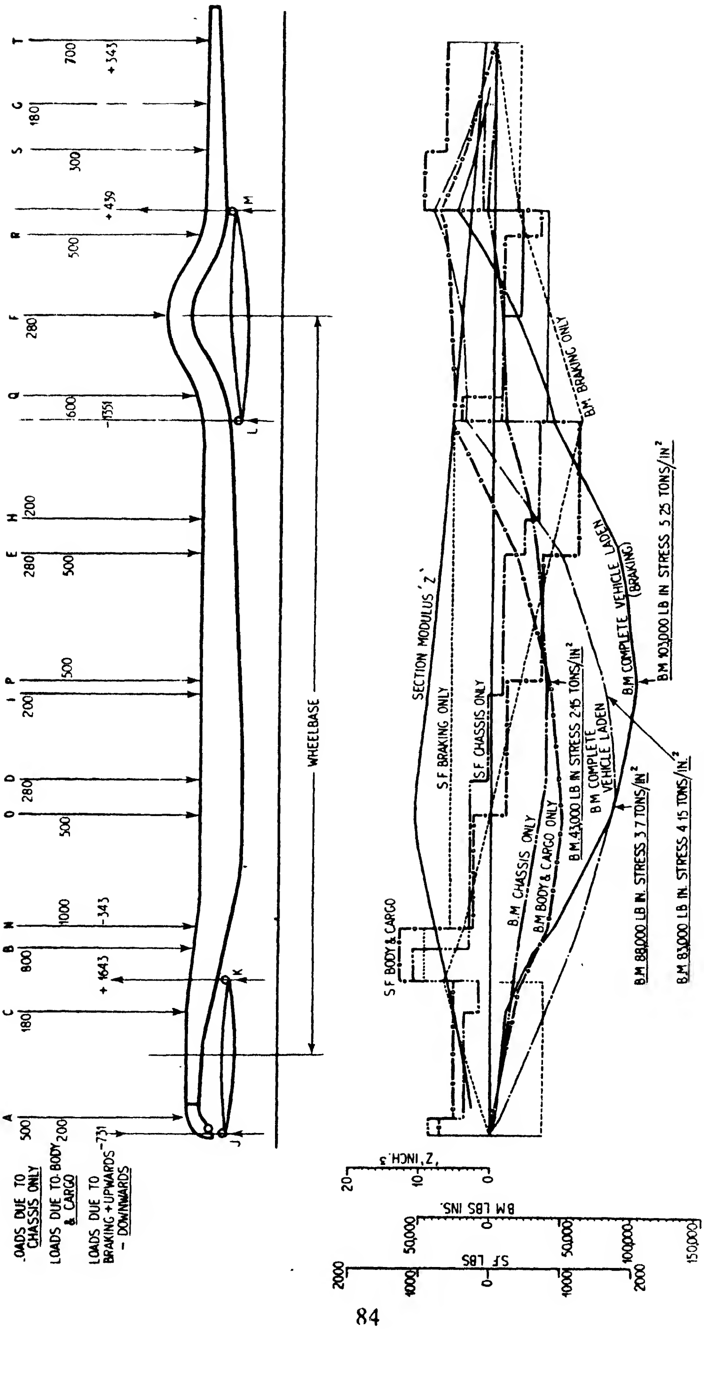



Fig. 51—Frame stress diagram

(iii) Prepare bending moments and shear forces (S.F.) for chassis loading only.

(2) *Static loading for body and passengers or body and load:*

(i) Distribute 50 per cent total body and passenger load along one side member on diagram and letter each point.

(ii) Calculate spring point reactions for laden body loads only.

(iii) Prepare B.M. and S.F. as at 1 (iii).

(3) *Braking reactions:*

(i) Define c.g. of fully-laden vehicle and body and calculate the transference of load for vehicle deceleration of 16·1 ft/sec/sec, or $\mu = 0{\cdot}5$.

If w = weight transferred from rear to front axle
W = weight support by rear axle (static conditions)
$\mu$ = coefficient of adhesion, tyre to ground
h = height of centre of gravity in ft above ground
L = wheelbase (ft)

then $w = \dfrac{W \times \mu \times h}{L}$.

This weight must, of course, be added to the static front axle load.

(ii) Apply result to front and rear axle loading.

(iii) Calculate B.M.'s and S.F. again separately and note sign.

Having observed the foregoing data, find the algebraic summation of B.M. chassis only 1 (iii) and B.M. due to body and cargo 2 (iii). This gives the complete B.M., etc., for static laden vehicles.

(4) Plot these results as diagrams, on a vertical scale.

(5) Follow this by plotting B.M. and S.F. for braking reactions 3 (iii).

(6) Combine algebraic results obtained at (4) and (5), that is static laden vehicle and braking reaction B.M.s. This now gives the bending moments for vehicle fully laden and deceleration at 16·1 ft/sec/sec.

(7) Calculate frame section moduli at several points along the side member, amongst which points should be one at each of the front axle, rear axle, and at body attachment points.

(8) From data (6) and (7) produce frame-section stresses. Plot this on B.M. diagram to scale.

The stress diagram will not only illustrate the position and magnitude of frame stress, but will emphasise any rapid change. It is this sharp change which must be avoided, and suitable adjustment to the section must be carried out by deepening or flitching. In any case the Z of the section must be increased locally at the offending point. If a moderately low stress is arranged, then this standard method of compiling approximate loading will be found adequate to meet normal circumstances.

The question of cross-member and side-rail section, however, requires further consideration with particular reference to torsional-rigidity properties. A side member may be boxed and yet afford little or no additional assistance in this respect, and the complete frame weight has thereby become increased. It is therefore necessary to consider which members are to be box section and which may conveniently be left in channel section. The principle of analysis may be based on that of two tubular cantilevers, at 90° to one another, rigidly attached at their joint and built in at their ends. See Fig. 52.

If a load P be applied at C at right angles vertically to the plane of ACB, any displacement of C due to movement of one part of the structure must be accompanied by similar movement of the other part. Bending and twisting must therefore be induced in both. If the point of B is gradually moved until the cantilevers lie parallel and B lies beside A, then the amount of twist also decreases through the process in proportion, until it becomes zero, when the angle BCA is O. On this assumption it may be stated that the torsional stiffness of each member will contribute most to the structure when the interaction of bending and twisting is a maximum, in other words when the angles between them are as nearly 90° as possible.

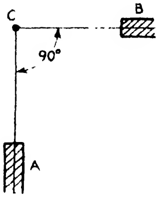


Fig. 52

The bending stiffness is, of course, represented by the product of Young's modulus E, and moment of inertia about the neutral axis I, whilst torsional stiffness for a box section is $\frac{4A^2Gt}{P}$

where A = area enclosed by the section.
G = modulus of rigidity
t = wall thickness
P = perimeter of the section.

The ratio of these two properties is therefore expressed as

$$\frac{E}{G}\frac{IP}{4A^2t} = \frac{IP}{I\cdot 6A^2t}, \text{ if Poison's ratio} = \frac{1}{4}.$$

If the section be rectangular this ratio is 1·25, for a width 2b and depth 2d, and the ratio becomes—

$$\frac{(d^3 + 3bd^2)(b + d)}{4{\cdot}8\ b^2d^2} = R.$$

The deflection of the two cantilevers produced by load P is

$$\frac{PL^3}{EI}\left(\frac{1}{6} - \frac{1}{8R + 1}\right)$$

If, however, R is assumed to be 1·25 for a doubly symmetrical section then $\delta = \frac{PL^3}{9EI}$.

The deflection is therefore reduced to two-thirds the value for that of zero torsional stiffness due to the additional torsional stiffness.

Frames for automobile chassis are usually of the deep-section channel or box section. The torsional stiffness of such a section has but secondary effect upon the frame structure stiffness as a whole, and therefore disposition of weight should be considered when boxing is carried out, and thought given as to whether the extra weight should not be applied to the cross members, leaving the main side members at channel section.

Considering the question of cross bracing: If diagonal bracing is arranged in which the subtended angle is 120°, at the centre, then the length AB = 2L $\sqrt{3}$ = 3·47L and OB = 2L. For channel section, deflection at B = $\frac{P(2L)^3}{3EI}$ = 2·67$PL^3/EI$. For box section, the corresponding result is 2·48 $PL^3/EI$. The proportional deflection therefore for a box-section frame per unit length (see Fig. 53) is

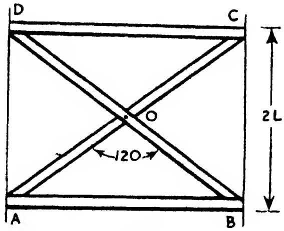


Fig. 53

$$\frac{2{\cdot}48\,\frac{PL^3}{EI}}{3{\cdot}47L} = 0{\cdot}715\,\frac{PL^2}{EI}.$$

In the case of a 60° bracing, the length AB is 1·156L and channel section shows a deflection at B of 0·517 $\frac{PL^3}{EI}$ whilst the box section gives a considerable improvement to stiffness of 0·341 $\frac{PL^3}{EI}$ and the proportional deflection per unit length 0·295 $\frac{PL^3}{EI}$. If the cruciform of diagonal bracing is replaced by a single member torsionally stiff, noted YZ and AB = 2L, the deflection at B is $\frac{PL^3}{EI}$ (see Fig. 54).

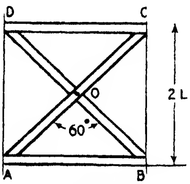


Fig. 54

Now suppose that YZ is box section equal in depth to the diagonal bracings, four times the width and of equal weight to the members DB and AC, then deflection at B is 0·558 $\frac{PL^3}{EI}$, which shows upon comparison with the previous results a very considerable increase in stiffness, weight for weight.

Assume further as another comparative step that the length of AB is 3·47L, similar to the 120° diagonal braced frame; it is again found that, all weights being equal, the deflection at B becomes $2\frac{PL^3}{EI}$, compared with 2·48 $\frac{PL^3}{EI}$ for the cruciform type (see Fig. 55).

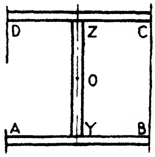


Fig. 55

The conclusion is therefore that the torsionally-stiff cross member provides greater frame stiffness than diagonal bracing, and as most of the deflection in the last example is caused by bending of YB the remaining distortion is due to twisting.

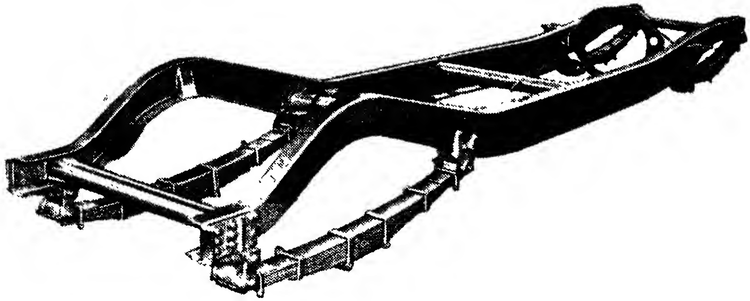


**Fig. 56—Heavy passenger vehicle frame**

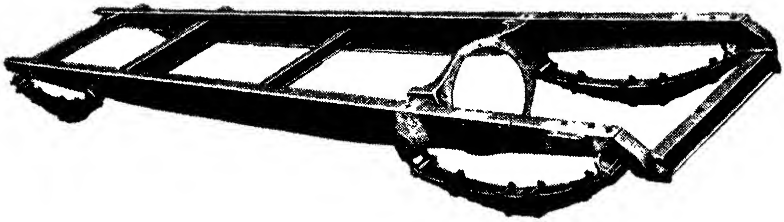


**Fig. 57—Heavy goods vehicle frame**

Still greater torsional stiffness would be obtained by employing the weight of YZ in the construction of more torsion members to reduce the uninterrupted lengths YB, YC, etc.

Alternatively the addition of a cross member through the centre of a diagonally-braced frame brings into action the bending stiffness of the side members. Although the 120° diagonals possess proportions more suitable than the 60° bracing to the automobile frame, its disadvantage lies in the fact that side members AB and DC contribute little to the stiffness, whilst the 60° angle stiffens only a short part of the frame length. A series of transverse members would therefore appear to be efficient substitutes.

# CHAPTER 5

## ROAD SPRINGS

### CALCULATIONS

MUCH has been written in previous chapters in connection with suspension, but the observations have been made mainly with reference to the suspension and its effects upon vehicle performance. The different types of springing have not been considered previously, and therefore this section will be mainly devoted to considerations of individual types of springing.

The function of the vehicle-suspension springs is mainly to increase the period in which the energy is absorbed and released, and the gradual dissipation of road shocks smooths out these objectionable surface qualities so that the reactions are more in harmony with the frequencies to which the human structure is accustomed. A low frequency of oscillation is essential to good suspension and, since it is inversely proportional to the static deflection, it therefore follows that this latter quality must be high.

The frequency formula is well known and is stated thus

$$T = 2\pi\sqrt{\frac{d}{g}}$$

or $T = 0{\cdot}319\sqrt{d}$

where $T$ = time for one complete oscillation in seconds

$d$ = static deflection (in)

and if $n$ = number of oscillations per minute

$$n = \frac{60}{T} = \frac{60}{0{\cdot}319\sqrt{d}} = \frac{188}{\sqrt{d}}.$$

Weight, of course, is often of considerable importance and thus presents a problem which is bound up closely with high static deflections. The road spring is subjected to varying range of stresses, probably reaching 120,000 lb during shock absorption to 40,000 lb upon rebound and therefore alloy steels with high fatigue values are essential.

The weight of material in any spring depends upon stress, static deflections and load in the formula $\frac{L \times \delta}{S^2} \times K$ = weight = W where L = load, $\delta$ = static deflection under load L, and S = stress existing under load L. " K " depends upon the type of spring, whether laminated, torsion spring, volute or torsion bar, and for the laminated type will fall between 30,000,000 and 35,000,000. If the leaf spring is utilised only as a means of suspension and does not have to act as a torque arm or radius rod, the torsion-spring weight is half to one-third that of the leaf spring.

It is of interest to note that in the weight formula the stress factor is squared. It will be appreciated, therefore, that any increase in allowable stress in a spring rapidly reduces the weight of the necessary

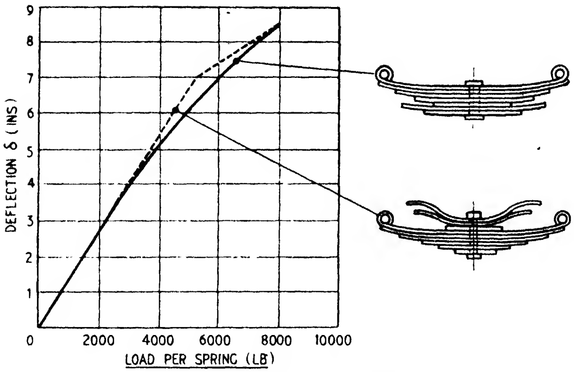


Fig. 58—Spring load/deflection

material, so that if the stress could be raised from say 40,000 to 50,000, or 25 per cent, the weight would be reduced in the ratio of 16 to 25 or roughly one-third.

We know, then, that large deflections are necessary for slow frequencies and that the load determines deflection. If it were possible to maintain the same frequency under all loads we should have the ideal spring, but in the case of the heavy commercial vehicle it is extremely difficult to obtain satisfactory frequencies for light loads; for instance, a 3-ton truck rear spring might take a load of say, 1,500 lb unladen and with overload reach as much as 6,000 lb, whilst deflections of say 5 in under overload conditions would be replaced by a deflection of perhaps 1 in only unladen; the frequencies being 85 loaded and 188 unladen.

It is for this reason that multi-stage and helper springs are often fitted to the heavier type vehicle where rear-axle loads vary considerably. The progressive or multi-stage spring has a much more gradual stiffening than the helper type, and is illustrated by a typical example (Fig. 58).

There is one fundamental principle to be considered, which the helper spring often violates, being much too stiff compared with the main spring. The increase in rate or stiffness in lb/in$^2$ must not be more rapid than the increase in load. If, for instance, a spring had a static deflection of 6 in for a load of 2,400 lb or a mean rate of 400 lb, the doubling of the load should only double the rate to a mean of 800 for the last 2,400 lb range and the travel between 2,400 and 4,800 lb would be 3 in if the same degree of comfort is to be retained at the heavy load as at the lighter weight.

There have been many spring-leaf sections evolved in an effort to reduce spring weight, a notable instance being the Toledo Woodhead grooved section and the Eaton grooved section in U.S.A., for which it was claimed that for equal load and deflection weight could be reduced by at least 7 per cent. It has been previously shown that weight does not depend upon length or width of spring leaves and therefore the following advantages for increase in both these dimensions should be given consideration :

(1) Mechanical strength of spring eyes due to the fact that the

main leaves may be heavy without the possibility of over-stressing throughout their deflection range.

(2) Smaller variation in eye centres with varying deflections.

(3) Greater resistance in the main leaf against loads from drive and torque. Interesting figures by Wöhler on some of Krupp's spring steels show that the number of vibrations before failure are dependent upon the maximum stress and the stress range, and that beyond a certain critical value if the stress range is increased, the spring life is considerably reduced. Some of these figures are reproduced :

TABLE 4

| Stress range in lb/in² | Number of reversals before failure | Condition of leaf |
|---|---|---|
| 0 to 110,000 | 39,950 | Broken |
| 0 to 88,000 | 117,000 | Broken |
| 0 to 66,000 | 468,000 | Broken |
| 0 to 55,000 | 40,600,000 | Unbroken |

It will be seen that the life was increased approximately 1,000 times to an unbroken leaf by reducing the stress range 50 per cent and further it will be noted the great life increase between the stress ranges of 66,000 and 55,000 lb/in².

The duration of spring life depends not only upon the material used in manufacture, but also upon the treatment to which it is subjected. It is contended that most springs fail through fatigue, which in turn mainly occurs when the leaves are in a state of tension—a high residual-tension stress occurring during the unloading part of the cycle which is sufficiently high to cause fatigue failure. Various methods have been suggested which have as their objective the improvement of the fatigue value of the material. One of the most promising of latest practice is that of shot-peening each leaf on its tension side and upon the two edges which increases the tensile strength of the material immediately below the surface whilst at the same time it sets up a longitudinal compressive stress in the thin skin. It is known that fatigue cracks generally emanate from the tension side of the leaf and therefore this compressive stress must be overcome by an applied stress in order that the stress on the tension side of the leaf may become a net tensile stress. The strengthening of the leaf through this layer of metal under residual compressive stress is only effective so long as this stress is not removed by any other working stresses.

It is claimed that greater endurance is obtained by peening the tension side only, rather than peening both sides. In the process the camber of the leaf is decreased, which would appear to indicate a favourable residual compressive stress. This change in camber is offset by originally setting the leaf to a slightly greater camber. The effectiveness of shot-peening is dependent upon the quantity of work done, which again depends upon the mass of shot, its velocity and the quantity used, whilst care must be exercised to ensure the leaf being completely covered and that the shot is of reasonable uniformity of size.

There should also be a limit to the amount of broken shot permissible. In practice the speed of peening varies, some conveyors moving at 16 ft/min whilst others have reached 32 or even 40 ft/min. It has been found that the lower speeds are better for increasing fatigue resistance.

There does not appear to be a great deal of data available at present which makes predetermination of shot size, velocity, etc., possible; each component has to be treated on its own particular merit. Experience is, however, rapidly being gained and an ingenious means of ascertaining the degree of peening being experienced by the work has been developed by Almen. Briefly it consists of a flat strip of steel attached to a block which in turn is clamped to the part passing under the shot stream. One side only therefore is peened (Fig. 60). Upon removal from the block, the strip is placed on a dial indicator to ascertain the change in longitudinal curvature, which (since the operation of peening is cold working on one side only) assumes an arc shape, the height of which is considered to be a measure of the degree of shot peening when correlated with fatigue resistance. Correlation can only be developed by repeated tests on actual parts until sufficient data has become available.

A time-intensity curve is plotted in a similar manner to that shown in Fig. 59. The desired intensity of peening is determined by the operator and is represented as AB. Curve (1) utilises shot of given material, size and striking velocity for various intervals of time, whilst curve (2) is obtained by varying one or more of the factors of curve (1). It will be noted that from curve (1) time of exposure is critical for the particular technique appertaining and less critical in that adopted in curve (2).

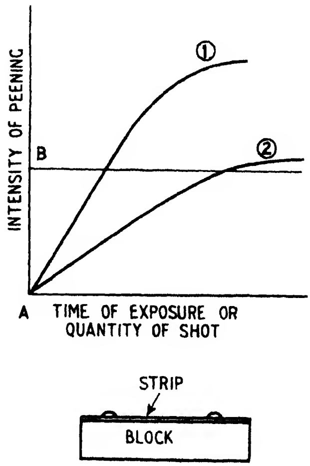


Fig. 59

Excellent results have been obtained with springs for a heavy vehicle peened with 1/32 in diameter steel shot at a nozzle pressure of 30 lb and an intensity of 1 sq. ft/min, the material being silico manganese steel, percentage of broken shot not to exceed 25 per cent although for general purposes, if the shot diameter is limited to between 0·025 in and 0·032 in in well-formed profile, good results will be obtained.

Misformed shot may possibly cause indentation, that is why broken shot is limited by specifying a percentage minimum, usually of the order of 25 per cent. The material is also most important and good shot should be to the following approximate specification:

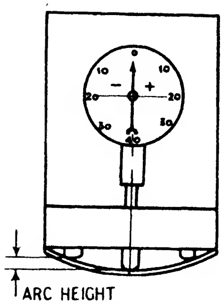


Fig. 60

| | |
|---|---|
| Combined Carbon | 3·30 |
| Graphitic Carbon | 0·15 |
| Total Carbon | 3·45 |
| Silicon | 1·80 |
| Manganese | 0·55 |
| Sulphur | 0·15 |
| Phosphorus | 0·46 |

## EFFECT OF SPRINGS ON STEERING

Steering is somewhat affected by the location and camber of the springs in relation to the chassis frame, as upon a correct combination of these two features depends the amount of body roll experienced. Rear-axle skewing, which occurs when the vehicle rolls in rounding a curve, produces a rear steering effect which is unstable on account of the reversal of transverse forces acting upon the tyres when the turn is encountered. This displacement of axle is not improved by fitting a highly positive camber spring and over-steering is a possible resulting phenomenon. A flat spring under normal load is not sufficient to prevent the axle skewing and slight oversteer when the vehicle rolls, but if a reverse-camber spring is mounted in an inclined position, that is with the rear eye lower than the front eye, the axle will be retained as square with the chassis as possible.

Tests have been carried out to determine the forces of impact between the road wheels and ground. The vehicle tested had the following dimensions and data—front-axle load 3·5 tons, rear-axle load 7 tons, wheelbase 19 ft, front and rear tyres 9·75 × 24 single and twin. A set of interchangeable single 12¾ × 24 tyres formed the basis of comparison and it was found that throughout all tests the twin tyres cushioned less than the single tyres of similar carrying capacity. The unsprung weight at the rear offside wheel was 0·8 ton and the tests showed that up to road speeds of 40 m/h the tendency was for the reaction loads to increase in proportion to speed, but thereafter the increase was slight.

The conclusion drawn from numerous tests appear to infer that—

(1) It is possible for reaction loads to reach approximately three times the static load.

(2) The average amplitude of reaction over poor road surfaces is 1·5 to 2 times the static load.

(3) Even the smoothest roads produce reactions of 1·2 to 1·3 times the static load.

(4) Providing the tyre is not overloaded the amplitude of vertical

and horizontal vibrations is greatest when the radius of a depression in the road surface is equal to that of the tyre.

(5) Unsprung weight considerably affects the vibrations and reactions if the radius of the depression exceeds that of the tyre.

(6) The reaction produced by a raised obstruction is influenced more by resilience and unsprung weight than by tyre size.

## SHOCK ABSORBERS

Even though the road springs be very carefully designed, conditions of suspension possess so many conflicting factors, due to unknown quantities, that it is often necessary to augment the performance of the spring by fitting *shock absorbers.*

*The hydraulic type* affords the necessary adjustment since by variation of its " valving " and " bleeding " it can be made available with widely differing characteristics. It is possible to adjust these characteristics to conform to predetermined resistance curves and thus, if provided with separate bleed and relief valves for compression and rebound, the shock absorber may be made either (a) single acting, (b) double acting, or (c) differential acting. The shock absorber should not be considered as an anti-roll device, for whilst its progressive action is advantageous for normal spring damping if adjusted to offer sufficient resistance to prevent roll, a harsh suspension would result and its functional utility greatly prejudiced. It must be remembered that roll is a slow-speed movement, as compared with the frequency of spring oscillation.

The single-acting hydraulic shock absorber is arranged to afford no resistance against compression, but to provide such damping effect as desired on rebound. The double-acting type provides hydraulic damping on both compression and rebound of similar magnitudes in both directions. The differential shock absorber is similar to the double-acting type, insomuch as its bleeding and valving permit resistance in both directions, but in this case of unequal amounts.

With the trend of suspension towards the higher deflection, both on front and rear springs, it is desirable to damp oscillation in both directions and the double-acting type is therefore the most suitable as it can be designed to possess such characteristics as are required to allow the road springs to freely deflect at low vehicle speeds or on smooth roads and at the same time to exercise greater control at high speeds or on rough ground.

If the vehicle is prone to wheel tramp or wobble, a double-acting shock absorber is sometimes of assistance if its characteristics are arranged so that its resistance is greater at low axle velocities as compared with the valve settings and bleeding of the shock absorber for a similar vehicle which does not possess such phenomena. Possibly a slightly larger type of differential hydraulic damper would be chosen as its resistance could be higher on rebound than compression and in consequence the possible harshness of ride which would occur through the excessive amount of damping required in the double-acting type would be avoided.

The shock absorber can, however, form a useful aid towards preventing body roll and anti-roll torsion bars may be coupled direct to the shock-absorber shaft by means of an extended boss on the lever arm splined to take the end of the torsion bar, or alternatively the shaft may be extended to carry a splined coupling to which the torsion bar could be attached, the shock-absorber arm being on the outside of the frame.

However, the general arrangement has so many possible solutions that the correct interpretation may be left to the discretion of the designer in choosing the most appropriate layout. This problem of stabilising a heavy vehicle, particularly a double-deck passenger vehicle, against roll on corners without detracting from its riding capacity is one which has been given much attention.

## THE STABILUS

The stabilus method—a German conception—was attended by a certain degree of success. Double-acting shock absorbers on each side of the vehicle were cross-coupled by high-pressure pipes, the pressures sometimes reaching the order of 3,000 lb/sq. in. In the particular arrangement in mind, the shock-absorber arms could move through a range of approximately 1¾ in without affecting the opposite component. Above this movement, a spring-loaded plunger embodied in the circuit provided solid resistance, so that the opposite shock absorber completely followed the movements of the first. Reference to Fig. 61 shows a diagrammatic layout of the pipe system in which high-pressure valves are fitted in pipes A and B and low-pressure valves in pipes C and D. Valves are also present in the pistons which allow recuperation of fluid in the system from the

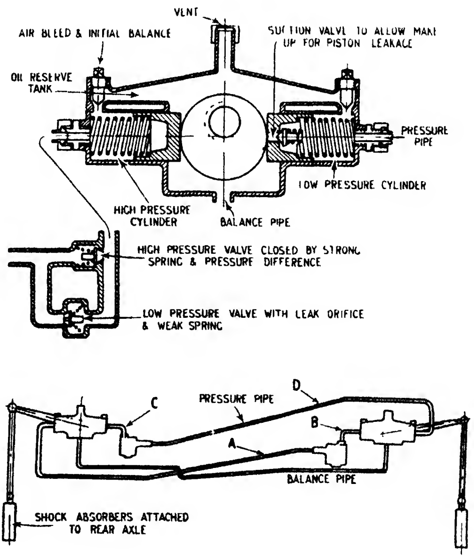


Fig. 61—The stabilus

reservoirs built into the shock absorbers. The function of the auxiliary valve is to permit levelling of the oil after replenishment, whilst the spring-loaded plungers are included to provide predetermined delay or lag between the action of the two shock absorbers, as previously mentioned.

Reverting to the actual spring design, there are several details which must not be overlooked since they play an important part in both spring life and spring performance. *Friction between the plates* has in the past been responsible for spring squeak and the digging into one another of the plate ends. There have been many designs evolved to overcome this occurrence, mainly by providing means to lubricate each plate.

Recent American practice includes an Oilite pad located under the end of each leaf, the load being transmitted from plate to plate by this pad. British makers achieved a certain degree of success with the Woodhead type of cascade lubrication, in which on certain plates reservoirs were provided each being connected to the other by grooves. The reservoirs were replenished by standard grease gun.

Other designs included roller bearings under the end of each leaf. Such a scheme as that in Fig. 62 was standardised on the French Peugeot with success, whilst a more straightforward means of lubrication is the screwed-in grease-nipple on the ends of each plate.

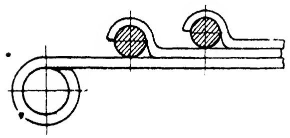


Fig. 62

*The form of spring eye* is important, particularly in a spring for a heavy vehicle, as any tendency to open out may have serious consequences. The ordinary rolled eye possesses the merit of low cost, but if the main leaf is insufficiently thick rigidity under fore and aft stresses, particularly that of braking, will be reduced and the eye will open slightly, causing additional stress to be imposed and permitting the spring pin to float. Moreover, there is always the difficulty in obtaining the correct tolerances between the eye bush and the eye, when springing open of the top leaf may be experienced even during bush assembly.

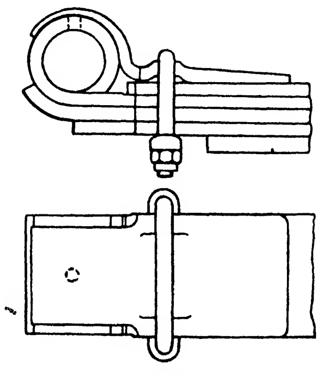


Fig. 63

The solid eye gained favour and has retained its popularity in this country mainly on account of its absolute rigidity in all directions. However, on the score of cost attempts to re-establish the rolled eye on the heavier type vehicles has resulted in a device by Samuel Fox & Co. for which it is claimed acts effectively against roll, whilst providing a substantial spring end for anchorage. This is shown in Fig. 63 which is self-explanatory.

## RUBBER APPLICATIONS

Recently much research has been made into the use of rubber as a medium providing reduction in harshness of ride, the elimination of lubricating points, reduction in static friction, the reduction of shock loading to the components so mounted, and the general cushioning of vibration effects. The application of rubber to the shackle pins of the automobile is not new, as in many private cars the Silentbloc bush has been incorporated and the Harris-type rubber bush is also widely adopted.

The problems involved are considerably augmented in the heavy vehicle. Nevertheless, it is now possible through the bonding of steel to rubber to provide some measure of cushioning to the spring shackles of the heaviest vehicle. In general, the rubber is placed in torsion and shear, the arrangement consisting of an inner shaft surrounded by an annular layer of rubber which is bonded to it and enshrouded by an outer steel housing, also bonded. The outer shell is split into two sections longitudinally, usually at 180°, the purpose of this being threefold. First it allows very high pressures to be applied to the bond between rubber and metal during the curing stage. Secondly, as the rubber shrinks after curing, the split housing allows this to take place without causing internal stress in tension, and, thirdly, it is known that to place the rubber under radial compression is advantageous, and this is possible when compressing the split shell into its housing of a somewhat smaller diameter than itself.

Applying such a principle generally to vehicle suspension, whether it be for shackles or independent suspension arms, the following basic procedure holds good for most applications in determining the size of the rubber torsion spring.

From the proposed drawing layout of the general arrangement, the length of the effective arm from the rubber spring to the steering knuckle or perhaps, shackle length if the application be a shackle bush, is determined. With this and static spring load, together with static deflection decided, the loaded moment and torsional rate of the spring is fixed by applying the pendulum formula $\delta = \left[\frac{188}{f}\right]^2$, $\delta$ being the normal length of arc through which the spring turns before it takes the static moment, whilst unit torsional rate is obtained from the division of static moment by subtended angle.

Assume the wheel to require a 4 in rise and 4 in rebound and the suspension to have a static deflection of say 6 in to 10 in, then if the static shear stress of the rubber be set at 120 lb/in² on the outside diameter of the inner shaft—a figure which takes into account the creep or set of the rubber under load (this allowable stress varies according to the Shore durometer hardness of the rubber, and will be referred to later)—the inner bush or shaft diameter may be calculated by providing sufficient area for the static stress at the bonding surface not to exceed the desired 120 lb/in$^2$.

The question of bush or shaft length is dictated by available clearances and chassis consideration, and will of course depend somewhat upon the desired degree of torsional rigidity of the shaft. The thickness of the rubber is the next consideration and the following analysis to relate thickness and spring rate are fundamental.

If in Fig. 64 $R_1$ & $R_2$ represent the inner and outer radii of the rubber,

$R$ = radius to any section
$l$ = length of bush
$S$ = unit shear stress
$T$ = torque
$G$ = shear modulus
$\delta$ = deformation degrees

$$\theta = \frac{S}{G} \text{ for small angles}$$

$$d\delta = \frac{\theta dR}{R}$$

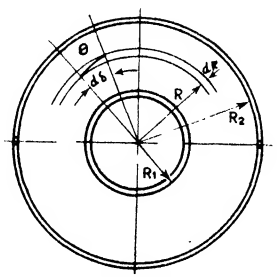


Fig. 64

Then for any given torque $T = 2\pi R^2 S l$

and as $R^2S$ is constant $S = \frac{R_1{}^2 S_1}{R^2}$

$$d\delta = \frac{SdR}{GR} \text{ or } \frac{R_1{}^2 S_1 dR}{R_2{}^2 GR}$$

$$\text{and } \delta = \frac{R_1{}^2 S_1}{G}\int_{R_1}^{R^2} \frac{dR}{R^3} = \frac{R_1{}^2 S_1}{2G}\left(\frac{1}{R_1{}^2} - \frac{1}{R_2{}^2}\right)$$

$$\text{and angular deformation } \delta = \frac{T}{4\pi l G}\left(\frac{1}{R_1{}^2} - \frac{1}{R_2{}^2}\right).$$

The value of G is generally defined by the slope of the tangent to the shear curve. Rubber in a torsion spring is considered as a series of layers of thickness dR, all in shear, the stress varying inversely as the square of the radius of the layer. In addition to this condition in mathematical form it may be represented graphically and is shown in Fig. 65 in which each successive layer is set out as possessing a thickness equal to 2½ per cent of its inside diameter. Torsional deflection of a hypothetical element which under no load is radial is shown by the dark line. Each adjoining and successive layer distorts from its radial direction by an amount equal to its angle of departure for that stress, as obtained on the shear sample and which is graphically represented by the

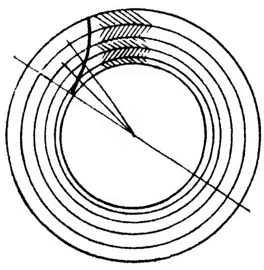


Fig. 65

stress strain curve for shear (Fig. 66). The shear sample consists of rubber cured between flat plates and stressed by parallel movement of the plates. A typical diagram is shown in Fig. 67, AB representing the layer of rubber under no load or shearing stress, whilst AC represents the amount of deformation $\frac{AC}{t}$.

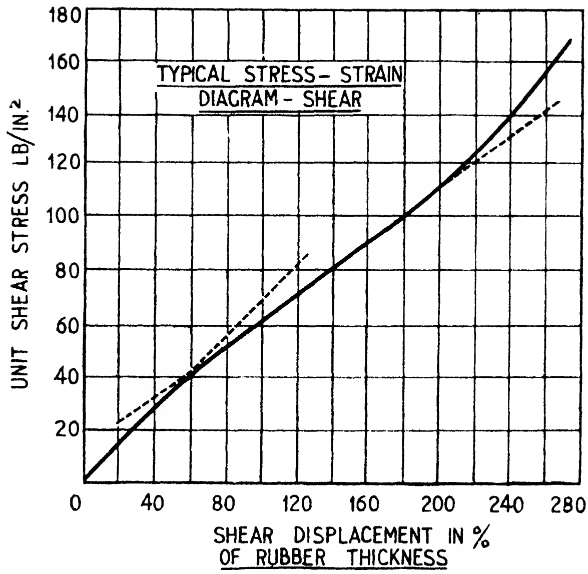



Fig. 66

One such curve is required for each rubber compound to cover the widest range of unit stress (static) or torsional rate. For the lower stresses it is only necessary to note the radius at which the required static stress appears and at that point draw the inside diameter of the rubber, or the displacement divided by the rubber thickness which is usually expressed as a percentage of shear deformation.

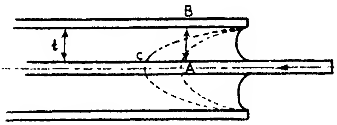



Fig. 67

This is not applicable, however, to the torsion springs: B C does not in this case represent the position of the distorted element (Fig. 68) as deformation varies with stress, which in shear is inversely proportional to the square of the radius of the rubber element from the origin.

In the shear sample, the change in length of the element BA to BC is sometimes referred to as "stretch," and the difference between the shear stress in the torsion spring and "stretch" will be appreciated by readings from a torsion spring fitted to a laden five-seater car for static and maximum bump load, wherein the unit shear stress at shaft diameter varies between these two conditions by 50 per cent, whilst outside the bond it

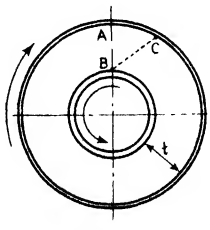



Fig. 68

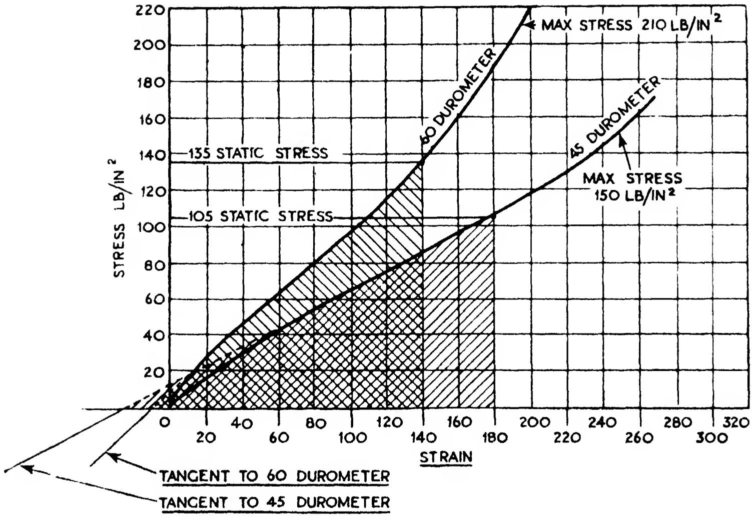



**Fig. 69—Stress/strain efficiency curves**

varied 46·6 per cent and approximated this figure at the outside diameter of the rubber. Its shear deformation at the bond varied 50 per cent, at the outside diameter of the rubber by 52 per cent, and stretch 46 per cent and 38 per cent at the respective positions.

From Fig. 68 it will be noted that the average stretch is $\frac{BC - BA}{BA}$ and in the example in question the figures for static and bump-loading conditions varied nearly 91 per cent, which clearly bear no fixed relation to the maximum stretch at the inner diameter of the rubber.

Development of the compounding of rubber and other essential ingredients required for the successful bonding to metal is such that at the present time a Shore durometer hardness of between 45 and 60 is probably the most satisfactory figure for the type of torsion spring under consideration.

If stress-strain curves are plotted, the area under the tangent serves as an index to the efficiency of the compound. Fig. 69 shows such a diagram and for a given strain indicates that 105 lb/in², is the static shear stress incurred by the 45 durometer sample. This figure for a suspension spring is stated to be quite satisfactory. By comparison with the stress strain curve for 60 durometer hardness, the area under the tangent must be the same as for the 40 stock in order that the same efficiency from unit volume may be obtained, and it will be seen that the static stress is approximately 135 lb/in², which for a wheel stroke of 8 in, that is 4 in compression and 4 in rebound, the maximum stress of 210 lb/in$^2$ is considered too high on the bond between rubber and metal.

It will also be noted that the maximum shear loads are disposed on the steeper portion of the two curves, which indicates that at high stresses the layers near the shaft, for a given angular movement, encounter less relative angular displacement than would be the case for the same bush angle at smaller loads. The effect of this is to produce relatively more deformation in the outermost layers, which are more lightly stressed, in which event the fatigue life, which depends on this deformation range as well as on unit load, is increased. However, before intelligent decisions can be made concerning each characteristic, it is necessary to develop a series of samples, all of which would be acceptable for different springs, but which differ in their characteristics regarding fatigue life. From the data obtained the designer can select the most satisfactory combination of fatigue life, creep, and hardness for this particular application.

In conclusion a few brief comments upon the process of bonding will be found to be of value in understanding some of the problems involved in making any stock selection. The process consists of vulcanising rubber in contact with a surface which has been plated with the correct type of brass, as certain constituents of compounded rubber combine chemically with a specific type of brass which is electrolytically deposited in a very thin layer. The bonded joint therefore is chemical and not thermoplastic.

Before introduction to the mould the rubber is processed. It is mixed with various ingredients. The mixer or mill consists of two rollers horizontally opposed, and which revolve at different speeds. They are, of course, adjustable. The rubber is squeezed between the rollers and forms an inherent sheet on one of them. At this stage various powders are introduced as filling agents. For strenuous duties the filler is usually one or more of the wide varieties of carbon blacks, whilst in addition certain chemicals are added so that the application of heat will permit vulcanising, necessary in order to eliminate the thermoplastic state of the rubber in its previous state.

The mixed rubber is moulded under pressure in steam-heated presses. The rubber does not melt but is forced into the moulds, which are also under the influence of heat and pressure. The bonding takes place in the mould concurrently with vulcanising at a temperature of 300/320° F and pressure in the region of 1 ton/in$^2$ for a period depending upon the size of mould of upwards of 15 minutes.

The sequence of events for the whole process is as follows:

(1) Clean the parts to be bonded.

(2) Brass plate the bonding surface.

(3) Prepare the unvulcanised rubber.

(4) Rubber and plated surface accurately positioned in the steel mould.

(5) The mould is partially closed and placed in the vulcanising press. The press closes the mould and ejects any surplus rubber.
(6) Vulcanising and bonding take place during the time the mould is under pressure.
(7) Mould opened, parts cooled, cleaned and trimmed.

In the torsion spring, the inner shaft and the outer steel shell after bonding transmit the load to the rubber area so that the rubber and the bonding supply the two resistances opposing outside forces.

A few general design formulae of a simplified nature are set down as being satisfactory for most problems concerning rubber encountered in automobile engineering.

The fundamentals to be remembered are that (a) for parallel shear units which carry a continuous load should be limited to 25 lb/in$^2$, based of course upon the weight (static) supported, (b) for a variable load 35 lb/in$^2$ and for units (c) in continuous torsion 50 lb/in$^2$ should be the maximum, but (d) for a variable torsion load the figure can be increased to 70 lb/in$^2$, based upon maximum torque.

The reason behind the higher stress figures for torsion units is that the bond area under shear is continuous and free from stress concentration. It has been said that the energy stored in a spring varies as the square of the stress, but it also varies as the reciprocal of the shear modulus, which in turn bears relation to the Shore hardness. These relations are tabulated below for the range of hardness considered to be the most economical for practical use: 40 durometer is probably the most widely used, although 30 is a soft commercial stock, with 60 durometer at the other end of the scale. It will be noted that the high modulus of the latter precludes the possibility of any higher number:

*Table* 5

| Shore Durometer Hardness | 30 | 40 | 50 | 60 |
|---|---|---|---|---|
| Shear Modulus lb/in$^2$ | 50 | 70 | 95 | 140 |

The plates to be bonded must be of sufficient section to avoid distortion when transmitting the load to the rubber, whilst in order to permit a complete cure throughout the volume of rubber the thickness should not exceed $1\frac{3}{4}$ in. Any application which requires greater deflection than can be accomplished within this limit should be arranged for duplicate or multiple units.

For the different types of rubber spring, equations numbered (1) define the shape of the rubber, whilst in explanation of the ratio m/n it should be stated that this depends upon the uniformity of stress throughout the rubber volume. If in expressions numbered (10) it is not present then m/n is unity and the stress distribution is uniform.

*Nomenclature to be used throughout all examples*

(See Fig. 70)

A = area in sq. in of rubber bond
$\delta$ = deflection (in)
G = shear modulus of rubber (lb/in$^2$)
H = height (in)
$H_1$ = height (in)
h = variable height (in)
P = load (lb)
S = shear stress (lb/in$^2$)
r = radius (in)
e = base of natural logarithms
K = radius factor
m = function of K
n = function of K
x = variable radius
R = torque arm (in)
t = thickness of rubber
$\theta$ = torsional deflection in degrees
V = volume of rubber (in$^3$)
E = efficiency = P$\delta$ or work stored (in. lb) $\times$ 2.

Note that in example 2, the shape of the rubber is such that the area at any radius is uniform, hence the unit stress is uniform throughout the volume, and also at the surfaces which are bonded. In example 3, there is an excess of rubber at the conical end; this excess provides for reduced shear stress at intermediate radii. The bond areas of both surfaces however are equal, as they should be.

Example 4 shows a design of low efficiency in the true sense, insomuch as the stress is greatest at the inner tube, diminishing towards the outer diameter to a minimum at Kr. The bond area on the inside tube is smaller than that of the outer shell. In connection with formulae for examples 3 and 4 tables on pages 104 and 105 give the values of radius factor -- 1, and the values of m and n.

Note that in example 6 the rubber follows a straight line at its outer edges and thus it is stressed lower at intermediate sections. The areas of bond surface are proportioned to give equal shear stress on both tubes, whilst in the case of example 7 the area of bond on the inside tube is the highest stressed due to the uniform height of the unit. Example 8 shows the ideal type of torsional disc spring, wherein the rubber thickness is directly proportional to the radius which gives uniform stress throughout.

In the foregoing text in connection with the hardness figures quoted it should be appreciated that they do not represent elastic characteristics, but they do afford to some extent a general guide to the rubber moduli values, or its stiffness. It should also be emphasised that hardness is determined by several factors, of which stiffness is only one. Hardness readings represent only local surface condition and may

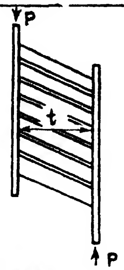
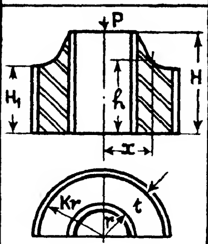
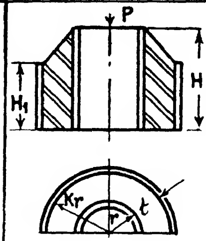
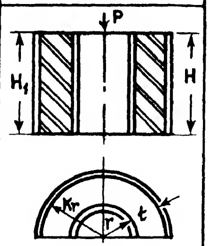
| UNITS IN SHEAR | | | |
|---|---|---|---|
| EXAMPLE 1 | EXAMPLE 2 | EXAMPLE 3 | EXAMPLE 4 |
|   |   HALF PLAN |   HALF PLAN |   HALF PLAN |
| (2) $A=\frac{P}{S}$ | (1) $xh=rH$ | $H_1h=H$ | $H_1=H$ |
| (3) $t=\frac{\delta G}{S}$ | (2) $rH=\frac{P}{2\pi S}$ | $rH=\frac{P}{2\pi S}$ | $rH=\frac{P}{2\pi S}$ |
| (4) $S=\frac{\delta G}{t}$ | (3) $m=\frac{\delta G}{rS}$ | $m=\frac{\delta G}{rS}$ | $m=\frac{\delta G}{rS}$ |
| (5) $\delta=\frac{Pt}{AG}$ | (4) $t=\frac{\delta G}{S}$ | $t=(K-1)r$ | $t=(K-1)r$ |
| (6) $P=AS$ | (5) $S=\frac{\delta G}{t}$ | $S=\frac{\delta G}{rm}$ | $S=\frac{\delta G}{rm}$ |
| (7) $V=At$ | (6) $\delta=\frac{Pt}{AG}$ | $\delta=\frac{Prm}{AG}$ | $\delta=\frac{Prm}{AG}$ |
| (10) $E=\frac{S^2V}{G}$ | (7) $A=2\pi rH$ | $A=2\pi rH$ | $A=2\pi rH$ |
| | (8) $P=AS$ | $P=AS$ | $P=AS$ |
| | (9) $V=At$ | $V=Arn$ | $V=Arn$ |
| | (10) $E=\frac{S^2V}{G}$ | $E=\frac{S^2Vm}{Gn}$ | $E=\frac{S^2Vm}{Gn}$ |
| | (11) $m=K-1$ | $m=\frac{2K\log_e K}{K+1}$ | $m=\log K$ |
| | (12) | $n=\frac{(K^3+3K^2-3K-1)}{6K}$ | $n=\frac{K^2-1}{2}$ |

**Fig. 70**

vary with surface irregularities and even with the size of the specimen. Stiffness or modulus measurements, on the other hand, represent the properties of the volume of rubber as a whole.

Hardness is perhaps the smallest concern of the designer when

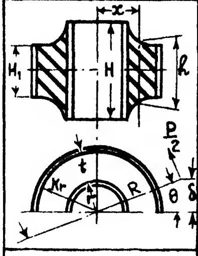
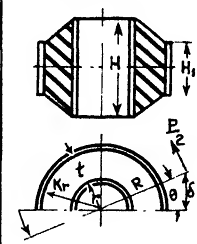
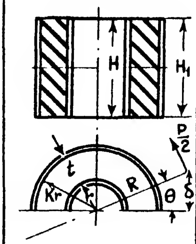
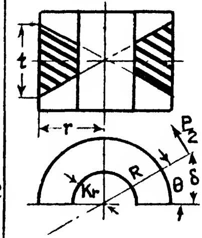
| Example 3 | | | Example 4 | | |
|---|---|---|---|---|---|
| K − 1 | m | n | K − 1 | m | n |
| 0 | — | — | 0 | — | — |
| 0·2 | 0·199 | 0·201 | 0·2 | 0·182 | 0·22 |
| 0·4 | 0·392 | 0·408 | 0·4 | 0·336 | 0·48 |
| 0·6 | 0·578 | 0·623 | 0·6 | 0·470 | 0·78 |
| 0·8 | 0·756 | 0·847 | 0·8 | 0·588 | 1·12 |
| 1·0 | 0·924 | 1·083 | 1·0 | 0·693 | 1·50 |
| 1·2 | 1·083 | 1·331 | 1·2 | 0·788 | 1·92 |
| 1·4 | 1·235 | 1·591 | 1·4 | 0·875 | 2·38 |
| 1·6 | 1·381 | 1·863 | 1·6 | 0·956 | 2·88 |
| 1·8 | 1·518 | 2·147 | 1·8 | 1·030 | 3·42 |
| 2·0 | 1·649 | 2·444 | 2·0 | 1·099 | 4·00 |

compared with other properties of rubber, such as the dynamic and static moduli of rigidity and elasticity, the damping characteristics and its resistance to fatigue. It is not desirable at this stage to review these various characteristics at length, but it is interesting to take a cursory glance at the comparative qualities of rubber and steel.

Steel relatively possesses elasticity of a very small order in direct tension, shear or compression. The engineer is aware that beyond the elastic range permanent deformation takes place, and whilst rubber in its main physical characteristics bears some similarity to metal, its possibilities regarding high deformations are extensive although the modulus figures are correspondingly smaller. For instance, if the modulus of rigidity of steel be $12 \times 10^6$ lb/sq.in. a rubber whose Durometer hardness is 50 with a static modulus of rigidity of 80 would be 150,000 times more elastic in shear than its steel counterpart.

The contrast is even more apparent when comparing the allowable stresses and strains. A rubber whose ultimate stress is 1,000 lb/sq.in. has a working stress of, say, 50 lb/sq. in., and a working strain of approximately 64 per cent. Its factor of safety is therefore 20 to 1, whilst a mild steel of 20 tons/sq. in. might have a working stress of 4 tons/sq. in. with a consequent safety factor of only 5 to 1; and a low working strain of 0·075 per cent.

Unlike metal, there is no corresponding elastic limit or final yield

| UNITS IN TORSION | | | | |
|---|---|---|---|---|
| | EXAMPLE 5 | EXAMPLE 6 | EXAMPLE 7 | EXAMPLE 8 |
| |  |  |  |  |
| 1 | $x^2h=r^2H$ | $H=K^2H_1$ | $H=H_1$ | |
| 2 | $r^2H=\frac{PR}{2\pi S}$ | $r^2H=\frac{PR}{2\pi S}$ | $r^2H=\frac{PR}{2\pi S}$ | $r^3=\frac{PR}{\pi Sm}$ |
| 3 | $m=\frac{DG}{RS}$ | $m=\frac{DG}{RS}$ | $m=\frac{DG}{RS}$ | |
| 4 | $t=(K-1)r$ | $t=(K-1)r$ | $t=(K-1)r$ | $t=\frac{DGr}{SR}$ |
| 5 | $S=\frac{DG}{Rm}$ | $S=\frac{DG}{Rm}$ | $S=\frac{DG}{Rm}$ | $S=\frac{DGr}{tR}$ |
| 6 | $D=\frac{PR^2m}{ArG}$ | $D=\frac{PR^2m}{ArG}$ | $D=\frac{PR^2m}{ArG}$ | $D=\frac{Pr^2t}{\pi r^4Gm}$ |
| 7 | $P=\frac{ArS}{R}$ | $P=\frac{ArS}{R}$ | $P=\frac{ArS}{R}$ | $P=\frac{\pi r^3Sm}{R}$ |
| 8 | $A=2\pi rH$ | $A=2\pi rH$ | $A=2\pi rH$ | $A=\pi r^2(1-K^2)$ |
| 9 | $V=Arn$ | $V=Arn$ | $V=Arn$ | $V=\pi r^2tm$ |
| 10 | $E=\frac{S^2V}{G}$ | $E=\frac{S^2Vm}{Gn}$ | $E=\frac{S^2Vm}{Gn}$ | $E=\frac{S^2V}{G}$ |
| 11 | $m=\log_e K$ | See footnote* | $m=\frac{K^2-1}{2K^2}$ | $m=\frac{2(1-K^3)}{3}$ |
| 12 | $\theta=\frac{180D}{\pi R}$ | $n=\frac{(K^3-1)(K+1)}{6K^2}$ | $n=\frac{K^2-1}{2}$ | |
| 13 | | $\theta=\frac{180D}{\pi R}$ | $\theta=\frac{180D}{\pi R}$ | $\theta=\frac{180D}{\pi R}$ |

$$^*m=\frac{(K-1)^3}{2(K^3-1)^3}\left[(K^6-1)+4K(K+1)(K^3-1)+6K^2(K+1)^2\log_e K\right]$$

**Fig. 71**

point for rubber, and rubber therefore retains most of its elasticity almost up to its point of final rupture.

In their different spheres both steel and rubber springs have their usefulness, although the properties of metal bonded rubber are such that it will absorb shocks in all directions with predetermined resistance, Example 5

| K — 1 | m |
|---|---|
| 0 | 0 |
| 0·2 | 0·182 |
| 0·4 | 0·336 |
| 0·6 | 0·470 |
| 0·8 | 0·588 |
| 1·0 | 0·693 |
| 1·2 | 0·788 |
| 1·4 | 0·875 |
| 1·6 | 0·956 |
| 1·8 | 1·030 |
| 2·0 | 1·099 |

Example 6

| K 1 | m | n |
|---|---|---|
| 0 | 0 | 0 |
| 0·2 | 0·179 | 0·185 |
| 0·4 | 0·318 | 0·356 |
| 0·6 | 0·422 | 0·524 |
| 0·8 | 0·499 | 0·696 |
| 1·0 | 0·554 | 0·875 |
| 1·2 | 0·595 | 1·061 |
| 1·4 | 0·623 | 1·261 |
| 1·6 | 0·643 | 1·472 |
| 1·8 | 0·656 | 1·691 |
| 2·0 | 0·666 | 1·926 |

Example 7

| K — 1 | m | n |
|---|---|---|
| 0 | 0 | 0 |
| 0·2 | 0·153 | 0·22 |
| 0·4 | 0·245 | 0·48 |
| 0·6 | 0·304 | 0·78 |
| 0·8 | 0·346 | 1·12 |
| 1·0 | 0·375 | 1·50 |
| 1·2 | 0·396 | 1·92 |
| 1·4 | 0·413 | 2·38 |
| 1·6 | 0·426 | 2·88 |
| 1·8 | 0·436 | 3·42 |
| 2·0 | 0·444 | 4·00 |

Example 8

| k | m |
|---|---|
| 0 | 0·667 |
| 0·1 | 0·666 |
| 0·2 | 0·661 |
| 0·3 | 0·649 |
| 0·4 | 0·624 |
| 0·5 | 0·583 |
| 0·6 | 0·523 |
| 0·7 | 0·438 |
| 0·8 | 0·325 |
| 0·9 | 0·181 |
| 1·0 | 0·000 |

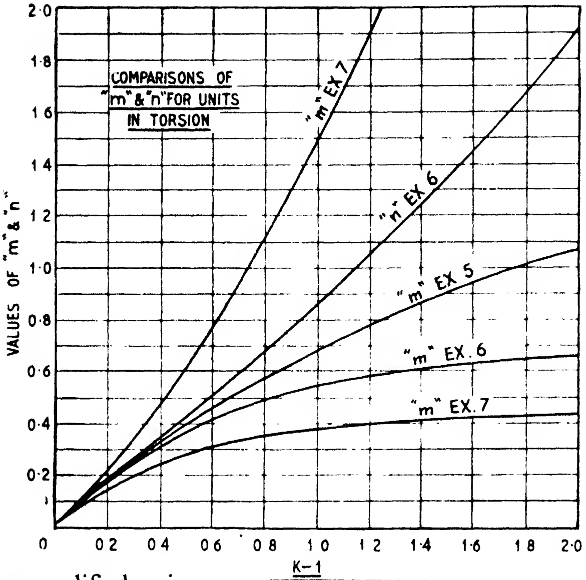



Fig. 72

but the steel spring is not so fortunately endowed. Not the least important of these comparative qualities is the non-linear characteristics of bonded rubber which can be exemplified in the rising stress strain relation in certain constructions, and the less technical qualities of sound insulation, and its non-conductivity of electricity.

Finally a word with regard to the metals to which rubber can be bonded. Among the various grades of steel, it is desirable to select those having low carbon content, since the high chromium steels present certain difficulties. Other materials include

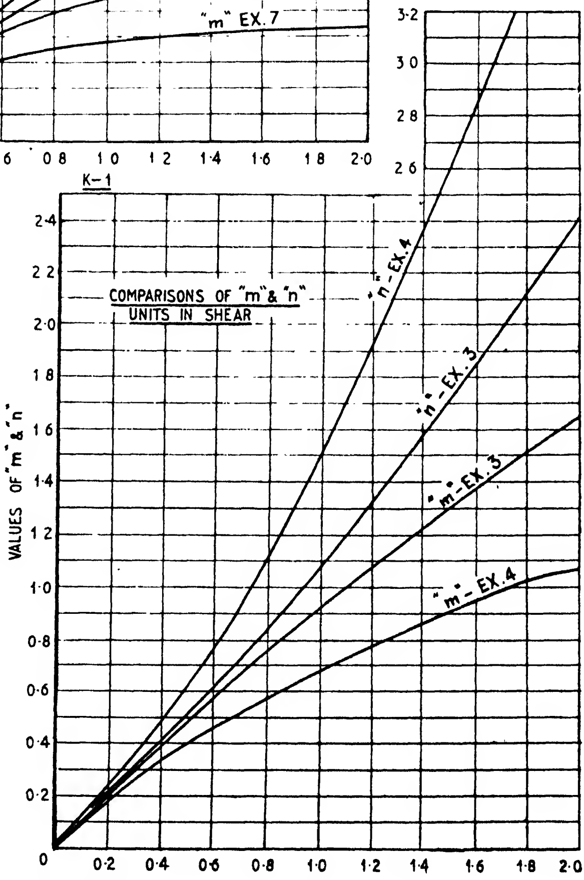



Fig. 73

cast iron, aluminium and its alloys, copper, brass, bronze, magnesium and zinc alloys.

These comparisons will serve to illustrate the utility of rubber in certain suspension applications, not only as a means of supplementing conventional layout such as the bonded rubber and steel shackle pin and leaf spring, but in the near future to form the main basis of the vehicle suspension.

Many modern designs of independent wheel suspension which employ torsion-rod springs, for example, include bonded rubber bushes in the wishbone link pivots, avoiding the necessity for lubrication; whilst, at the same time, vibrations are effectively damped at source.

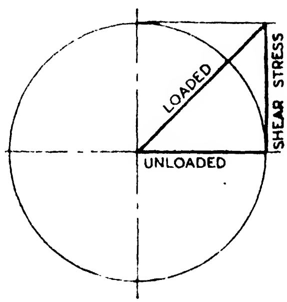


Fig. 74

## TORSION BAR SPRINGS

Rapidly gaining favour is the form of suspension which employs *torsion-bar springs*, which rod is the simple basic element of the coil spring. The calculation for bar size, angle of torsion, etc., are not in any way complicated, but the treatment of the bar is important. The stress in the rod is usually rather higher than one is accustomed to permit, but the explanation is based upon the effects of presetting beyond the elastic limit of the material; for instance, the stress which produces distribution similar to that in Fig. 75 is expressed by the formula

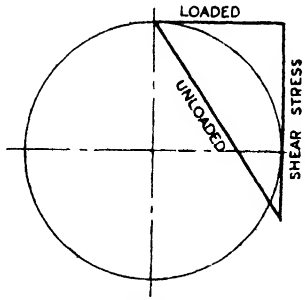


Fig. 75

$$S = \frac{16T}{\pi d^3}$$

where T = torque
and d = diameter of rod.

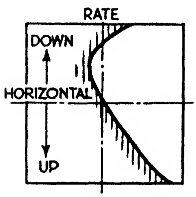


Fig. 76

Presetting, however, makes it theoretically possible for Fig. 75 to be modified in a similar manner to that shown in Fig. 74, and the stress in such a case becomes $S = \frac{12T}{\pi d^3}$. Much research has been carried out regarding the actual stress

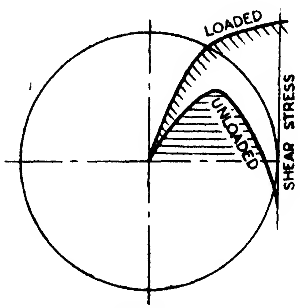


Fig. 77

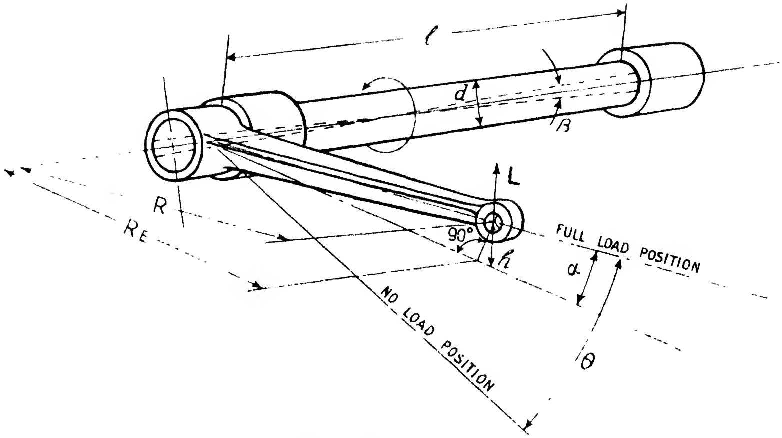



Fig. 78 -Simple torsion bar

distribution in heavily preset rods and it is believed that a distribution approximating that shown in Fig. 77 is possible.

When the stress S = actual, $S \simeq \frac{12T}{\pi d^3}$ the linear rate at the end of the lever arm has a variation similar to that found for the Horst mann suspension, and is not symmetrical on either side of the horizontal position. Its form is similar to the typical figure shown in Fig. 76, but for small arm radius the variation in rate is small.

If the following notation be used, the main calculations are as follows (see Fig. 78)—

S = surface stress in rod
d = diameter of rod
l = active length of rod
R = radius of arm
h = height of arm above horizontal
L = load on end of arm
T = torque
$\alpha$ = angle of arm above horizontal
$\beta$ = shear angle
$\theta$ = total angle traversed
$T^A$ = angular rate
$T^L$ = linear rate

Angular rate $T^A = \frac{T}{\theta}$, which is constant.

Linear rate is variable $T^L = \frac{dL}{dh}$

Torque $T = L \times R \cos \alpha$

Shear angle $\beta = \frac{\theta d}{2l}$

Surface stress in the rod, $S = \dfrac{16\,L \times R\cos\alpha}{\pi d^3}$

$T^A = \dfrac{G\pi d^4}{32l}$, where G is of magnitude $11{\cdot}5 \times 10^6$ for steel

$L = \dfrac{T}{R\cos\alpha}$

$h = R\sin\alpha$

$$T^L = \frac{1}{R\cos\alpha}\left(\frac{T^A}{R\cos\alpha} + \frac{L \times R\sin\alpha}{R\cos\alpha}\right) \text{ or } \frac{T^A + Lh}{R_E^2}$$

The design of suspension rods in torsion demands particular attention to the fixed ends. These are usually splined and from exhaustive tests it has been established that the sharp-rooted spline has a definite deleterous effect upon stress concentration and ultimate failure. The blunt serration considerably increases the fatigue strength and, if designed in conjunction with a well-radiused spline, the base may be successfully shot-peened, which not only increases the life of the splines but may correct possible material defects and those of processing, whilst preventing also the start of a fatigue fracture from " frettage " corrosion.

The relation of spline diameter to body diameter should approximate in excess of 1·25 to 1 or root of spline should be 1·15 times the body diameter. The length of spline should be as short as is consistent with requirements, since it will be appreciated that in practice splines cannot be cut so that there is full initial contact on all teeth for complete length as there is almost certain to be some slight misalignment. Moreover, the spline contacts at one end only and will cause loading beyond the elastic limit, so that yield takes place until sufficient surface contact has been made. When this happens high loading through work hardening makes possible the carrying of higher loads by an equivalent lower hardness material. Parallelism of splines to rod axis may be toleranced to 0·002 per in of wire length, when the splines are inspected by laying a small wire between two splines. A well-proportioned rod end may approximate that to proportions shown in Fig. 79. Shot-peening is acknowledged as an essential operation if long life is to be expected from highly-stressed components which are subject to fatigue.

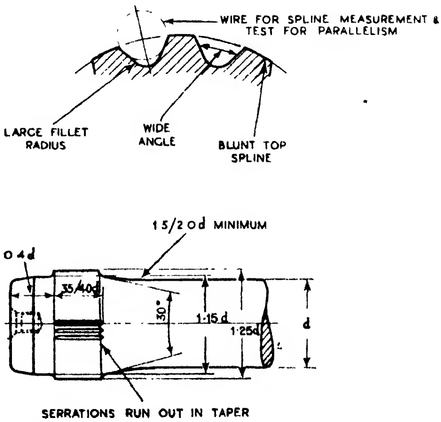



**Fig. 79—Proportions for torsion-bar end**

Another operation, that of pre-setting, is of primary importance as applied to parts the life of which depends more upon the maximum stress than the stress range. The work stored in a bar is, of course, proportional to the stress squared, but by pre-setting or " scragging " the bar will carry more than two and a quarter times the energy than that absorbed by the non-preset one. The bar is pre-set with a much higher torque than that which it will experience in service, and a permanent distortion takes place through the rearrangement of the material structure, so that not only is the surface loaded to its yield point, but also the other material to about half its outer diameter, when full torque is applied. Moreover, it is a cold working operation, and the elastic limit of the bar is higher after such treatment. The value of the elastic limit varies at different depths below the outside diameter, at which position when the bar is free a negative torsional stress is trapped, the magnitude of which is considerably greater than the load stress.

Pre-setting is essentially the last processing operation on the bar, as it is then in a highly impressionable condition. The method of carrying out pre-setting is to apply a torque to the rod which will produce a maximum shear strain angle of approximately 1·3° and a permanent set of 0·4°. Usual rod proportions have ratio of length to diameter 75 to 80 to 1 and therefore the virtual wind-up angle is in the order of 30°.

Pre-setting practice varies in detail, individual manufacturers modifying in some cases the number of torque applications, whilst others vary the magnitude of the applied torque. One British manufacturer applies repeated twists to the rod, each slightly in excess of the torsion angle to be experienced on the vehicle. As the bar settles it is constantly adjusted, and again twisted through the same angle until settling ceases after possibly 15 cycles have been required.

As a matter of interest the processing of the bars follows these lines—

(1) Upset spline ends.
(2) Normalise.
(3) Cold straighten.
(4) Rough turn to approximately 0·050 in larger diameter than finished bar.
(5) Finish grind.
(6) Cut splines.
(7) Heat treat, quench and temper.
(8) Final straighten at a temperature not less than 600° F.
(9) Shot-peen the bar.
(10) Shot-peen the splines.

Note: The body requires larger shot than the splines and recommended diameters are for body 0·060 in, and for splines 0·018 to 0·024 in diameter.

(11) Pre-set.
(12) Paint.

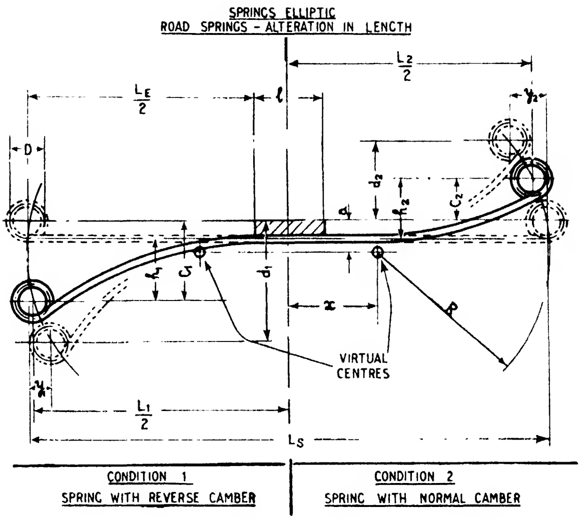



Fig. 80

## VERTICAL CENTRE OF OSCILLATION

Accurate brake and steering-linkage geometry is essential, and one of the main causes of brake ineffectiveness is the direct result of faulty positioning of the vertical centre of oscillation of the road springs, which in turn controls the calculated alteration in length of the spring. The following data covers both conditions of normal camber and reverse camber for the orthodox laminated leaf spring—

*Condition* 1 (*Reverse camber*) (see Fig. 80):

$L_1$ = spring eye centres (laden)
$h_1$ = height centre line of eye to centre line of top leaf (laden)
$C_1$ = vertical movement of centre line of eye from laden condition to flat-spring condition.

*For any required position :*

$d_1$ = vertical movement of centre line of eye below flat-spring condition.
$y_1$ = corresponding horizontal movement of centre line of eye.

$$C_1 = h_1 + \frac{D}{2}$$

$$L_S = \frac{L_1 + C_1 (2\tfrac{3}{4} C_1 - 4D)}{(L_1 - 1)}$$

$$y_1 = \frac{d_1}{L_E}\left(\frac{11d_1}{8} - 2D\right)$$

*For both conditions :*

$$L_E = L_S - 1$$

$$a = \frac{8D}{11}$$

$$R = \frac{4}{11}\left(L_E + \frac{D^2}{L_E}\right)$$

$$x = \frac{3L_E}{22} + \frac{4D^2}{11L_E} + \frac{l}{2}$$

*Condition* 2 (*Normal camber*) :

$L_2$ = spring eye centres (laden)
$h_2$ = height centre line eye to centre line of top leaf (laden)
$C_2$ = vertical movement of centre line of eye from laden condition to flat spring condition.

*For any required position :*

$d_2$ = vertical movement of centre line of eye above flat spring condition
$y_2$ = corresponding horizontal movement of centre line of eye

$$C_2 = h_2 - \frac{D}{2}$$

$$y_2 = \frac{d_2}{L_E}\left(\frac{11d_2}{8} + 2D\right)$$

$$L_S = \frac{L_2 + C_2\,(2\tfrac{3}{4}C_2 + 4D)}{(L_2 - l)}.$$

# CHAPTER 6

# THE TRANSMISSION

## PROPELLER SHAFTS

VARIATION in angular velocity between two universally jointed shafts, that is providing the joints are of the Hooke type and not constant-ratio joints, depends upon angularity between driver and driven members. Each of the joints has two rocking axes, each of which must maintain an angle of 90° one to the other at all times. Incorrect angular relation between the driving and driven forks is the origin of many transmission stresses and vibrations. It is therefore of primary importance that each fork or pin at one end should be mounted in the same plane as its corresponding member at the other end; even then, if there is any angularity in the shaft, uniform revolution will still not be possible, as the fluctuation in speed varies proportionally with the shaft angle.

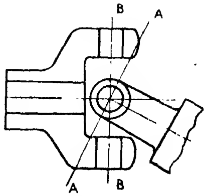


Fig. 81

When a joint of the type in Fig. 81 is in motion the pins describe what is termed two "great circles" having common diameters, since they represent the boundaries of the sections formed by the two paths. As the distance between A and B is constant it may be represented by a quadrant of the great circle. The two arcs set out in Fig. 82 are described whilst pin AB travels between the two points of intersection, and the point of maximum deviation from the vertical occurs when coincidence with these intersections is attained by either pin A or B. When A is coincident with these intersection points, angular velocity of the driven shaft is less than that of the driving shaft, and the opposite condition occurs when B is in coincidence. As the joint rotates there are four points at which the speeds of each shaft are equal.

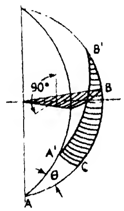


Fig. 82

In Fig. 82, point A has moved to A′ and B to B′, which point is found by describing an arc equal in length to AB. From B′ point C is found at the point of intersection of a further great-circle quadrant B′C being equal to B′A′. This geometrical representation may be interpreted thus:

$\theta$ = the angle between shafts CAA′, the angular motion of driving shaft is equalled by AA′, whilst AC = angular motion of the driven shaft, and finally B′A′C = 90°, as does B′CA′.

Solving the spherical triangle:

$$\text{Cos } \theta = \tan a \cot b, \text{ where } AA' = b \text{ and } AC = a$$

by reciprocals $\frac{\tan a}{\tan b} = \cos\theta$

and for small angles $\frac{AC}{AA'} = \cos\theta$

Solving for the ratio of angular velocity

$\tan b = \cos\theta \tan a$

and by differentiating $\sec^2 b\,db = \cos\theta \sec^2 a\,da$.

$$\therefore \frac{db}{da} = \cos\theta \frac{\sec^2 a}{\sec^2 b}.$$

As AC or b is not known, by squaring the expression for angular velocity

$\tan^2 b = \cos^2\theta \tan^2 a$

$1 + \tan^2 b = 1 + \cos^2\theta \tan^2 a$, but as

$1 + \tan^2 b = \sec^2 b$. Substituting in the differentiation of the angular velocity expression

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

which is the final ratio of angular velocity between driving and driven shafts.

If the figures are worked out for various degrees of angular motion, and shaft angles and the speed fluctuations plotted, it will be noted that the speed differences between the two shafts is considerable. If the angle is increased from 8° to 24° the 2 per cent variation becomes 18 per cent and therefore it will be readily understood that lack of uniform velocity combined with change of direction of the rotating pins is a possible source of shaft vibration.

Vibration of the shaft may be excited through either end loads, centrifugal force or torsional vibration. Should the shaft be fitted without a sliding muff considerable end thrust is experienced and, further, if the joint is of flexible type, the texture of flexible material will damp the actual shock load, but reduce the critical speed of the shaft tube. It is desirable therefore not to neglect end loading when making propeller-shaft calculations.

*The maximum bending moment on the shaft :* According to Prof. Greenhill this may be stated as

$$f = \frac{B.M.}{Z} + \frac{F}{A}$$

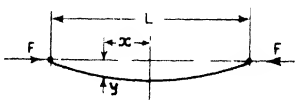


**Fig. 83—Propeller-shaft loads**

where F = end load
A = sectional area
W = weight of shaft (lb per unit length)
$\alpha$ = velocity (radians per sec)
B = flexural rigidity of the shaft or EI
y = deflection at X from the centre

and load due to centrifugal force $= \frac{W}{g}\alpha^2 y$ per unit length.

Now if the effect of gravitation be taken into account, there is one position in each revolution at which centrifugal force and weight exert their greatest effect

$$\text{and} \quad \frac{d^4y}{dx^4} + \frac{Fd^2y}{Bdx^2} - \frac{W\alpha^2}{gB}\cdot y - \frac{W}{B} = 0$$

which solves into

$$y = A_1 \sin \beta x + A_2 \cos \beta x + A_3 e^{\gamma x} + A_4 e^{-\gamma x} - \frac{g}{\alpha^2}$$

$\beta^2$ and $\gamma^2$ being the two roots of quadratic equation

$$\beta^2 = \frac{F}{2B} + \sqrt{\frac{F^2}{4B^2} + \frac{W\alpha^2}{gB}}$$

$$\gamma^2 = -\frac{F}{2B} + \sqrt{\frac{F^2}{4B^2} + \frac{W\alpha^2}{gB}}$$

y has the same value for values of x whether positive or negative. $A_1 = 0$, $A_3 = A_4$ and assuming that $y = 0$ and $x = L$, that is for no shaft deflection, and also that when $x = L$, $\frac{d^2y}{dx^2} = 0$ then

$$A_1 = 0$$
$$A_2 = \gamma^2 g \,/\, \alpha^2 (\gamma^2 + \beta^2) \cos \beta L$$
$$A_3 = A_4 = \beta^2 g \,/\, 2\,\alpha^2 (\gamma^2 + \beta^2) \cosh \gamma L.$$

$\therefore$ at any point along the shaft the bending moment is $EI\frac{d^2y}{dx^2}$ and the B.M. at the middle of the shaft is

$$EI\,(-\beta^2 A_2 + 2A_3\gamma^2)$$

*To find the lowest critical speed :*

$(\beta^2 + \gamma^2)\, A_2 \cos \beta L = 0$ and since $\beta^2 + \gamma^2$ is not zero unless $A^2 = 0$, $\cos \beta L = 0$.

Under stable conditions $\beta L = \pi, 2\pi, 3\pi$, etc.,

and the lowest critical speed $\beta = \frac{\pi}{L}$

$$\beta^2 = \frac{\pi^2}{L^2} = \sqrt{\frac{F^2}{4(B)^2} + \frac{W\alpha^2}{gB}} + \frac{F}{2B}$$

from which
$$\alpha = \sqrt{\frac{gB}{W}\left(\frac{\pi^4}{L^4} - \frac{\pi^2 F}{L^2 B}\right)}$$

The above formulae do not take into consideration any inaccuracies in weight distribution but they assume that the weight acts as a radial force in the same direction as the centrifugal force. Should the shaft not be perfectly balanced, the shaft will deflect towards the heavier side and a point will be reached when for a period there will be an excessive vibration. It will, however, disappear with any alteration in shaft speed.

At this critical speed vibration is caused by a change in the *axis* of rotation, and the shaft, instead of rotating around its geometrical centre, rotates about an axis through the centre of gravity and its geometrical centre. It therefore becomes deflected in such a manner that the geometrical centre traces a circular path around the centre of gravity of rotating mass at each revolution.

The whirling speed may be defined as that at which the stiffness is zero and its period infinite. The interval of time in which the shaft is passing through its critical value is so small that it does not permit of large shaft deflections, hence the fact that rupture does not always occur.

A brief description of what occurs in a shaft undergoing vibration may be necessary before the torsional resilience is considered: The shaft, starting from rest, increases its angular velocity up to the point where vibrations occur, the amplitude of course being small. Further increases of speed increases the vibration amplitude until eventually they reach a maximum. Still further speed increase eliminates these oscillations, until at another shaft speed the vibrations recommence and start the cycle once more. Speed may be increased until shaft distortion and fracture occur.

*Torsional resilience* must be considered, as the shaft should be designed to accommodate an angle of approximately $1\frac{1}{2}^{\circ}$ per 10 ft of shaft, which figure should be a maximum at maximum stress. Previous calculations should be subjected to a re-check to ensure this condition, and also that of adequate elastic strain energy, as the material should not be stressed torsionally above its elastic limits.

*The shear resilience of a tubular shaft* is expressed thus:

$\frac{x}{2\pi} 2\pi Lrt$, where $x = \frac{r}{R}$ fs and where fs is the intensity of shear stress at outside radius R

t = thickness of material
r = inner radius of shaft.

*The torsional resilience of a tubular shaft* is:

$$\frac{\pi L}{N} \cdot \frac{fs^{2}}{R_1^{2}} \int_{R_2}^{R_1} r^3 t = \frac{\pi L}{4N} \cdot \frac{fs^2}{R_1^2} (R_1^4 - R_2^4)$$

which equals $\frac{R_1^2 + R_2^2}{R_1^2} \times \frac{fs^2}{4N} \times$ by vol. of shaft.

where N = modulus of torsional rigidity

$R_1$ and $R_2$ = inner and outer radius of tube respectively.

*Torsional deflection* may be calculated from the usual empirical formula.

*End thrust* to which the shaft is subjected arises from many sources. It has its origin partly from a combination of resulting loads from vertical displacement of the axle and gyroscopic movement of the joints, which tend to substantially reduce the natural harmonic vibration of the shaft—when the interval of time between blows from road obstructions or inequalities is equal to the periodic time of one or more road springs. These combined loads have application at the rear end of the shaft and all act through the rear axle end

$e = \frac{AE}{L}$, which is the elastic force per unit of deflection

where $W$ = combined end loads
$A$ = cross-sectional area of shaft
$L$ = length between joints
$E$ = modulus of elasticity
$g$ = gravity
$K$ = torsional rigidity of shaft
$k$ = radius of gyration.

The frequency of torsional vibration, $n = \frac{1}{T}$, where $T$ = time of torsional vibration, and is equal to $T = 2\pi\sqrt{\frac{I}{K}}$, $I$ being the moment of inertia $\frac{Wk^2}{g}$ or $n = \frac{1}{2\pi}\sqrt{\frac{K}{I}}$ per second.

If $K$ = torsional rigidity of shaft and $N$ = modulus of transverse elasticity, $K = \frac{N\pi d^4}{3\,t}$

substituting $n = \frac{1}{2\pi}\sqrt{\frac{N\frac{\pi d^4}{32}g}{Wk^2L}}$

or $\frac{d^2}{20}\sqrt{\frac{Ng}{Wk^2L}}$ per second,

which for steel tube at $12 \times 10^6$ lb/in$^2$ approximately gives

$n = 3{,}400d^2\sqrt{\frac{1}{Wk^2L}}$ per sec.

If the shaft is built up of varying sections of different diameters the frequency of vibrations is of course different as the twist produced by unit torsion moment is the sum of twist in each system. An example of such a shaft is the swaged ends down to a small splined shaft diameter.

Frequency may be found by a modification of Morley's formula, thus: $\frac{32}{\pi n}\left(\frac{l_1}{d_1^4} + \frac{l_2}{d_2^4}\right)$ or written for steel as above with notation as Fig. 84.

$$n = 3{,}400\sqrt{\frac{1}{Wk^2\Sigma\,(l/d^4)}}.$$

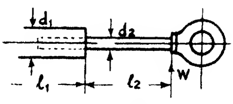


Fig. 84

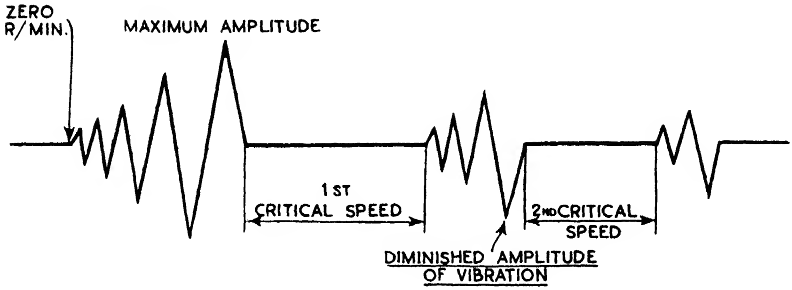



**Fig. 85—Graphical illustration of vibration phenomena**

Incidentally, when determining the material safe stress it must not be overlooked that most of the stresses are alternating and are often reversed, whilst it is more dangerous still to fail to appreciate the fact that it is the *stress range* which has an even greater bearing upon failure than does the actual magnitude of the stress. Considering stress reversals, a safe material figure is $7\frac{1}{2}$ tons compression, $7\frac{1}{2}$ tons tension and $5\frac{1}{2}$ tons in shear due to torsion. This is based on an ordinary ·30-·45 carbon steel of 15 to 22 tons yield and an approximate elongation of 10 to 18 per cent.

A graphical illustration of vibration phenomena is shown in Fig. 85. The shaft starting from rest gradually increases its angular velocity to a point where vibration occurs. The amplitude is at first small, but with an increase in speed the vibrations also build up until they eventually reach the maximum amplitude. If the shaft is on a balance machine the indicators will register the magnitude, when suddenly the indicators become at rest. It is at this speed that critical speed of the shaft has been reached. Any further increase in revolutions reproduces further vibration, but of smaller magnitude. Such phenomenon occurs with each succeeding speed increase until eventually the shaft becomes distorted and probably fails.

Stresses such as these, if applied in magnitude exceeding the elastic limit of the chosen section, produce an effect on the material structure such that rupture due to tension or compression resolves itself into ultimate failure through shear.

The actual joint arm or flange is subjected to torsional and bending stresses, introducing shear as a result of torsion. If one half the maximum low gear torque is applied to each arm, the resulting total shear stress due to torsion incorporating the polar moment of inertia will be in each arm $\frac{Sb}{2} + \sqrt{S_t^2 + \frac{Sb^2}{4}}$

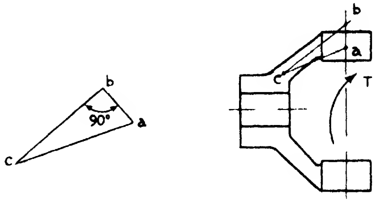



**Fig. 86**

as if cba $= 90^\circ$ then both torsional and bending moments are proportional to the length of the sides of the triangle (see Fig. 86).

## THE CLUTCH

The essential function of the clutch is to disconnect and connect the engine from the line transmission in order that the vehicle may be brought to rest without stopping the engine; or, in the case of re-engagement, it must be capable of connecting the rotating crankshaft at possibly high revolutionary speeds to the transmission, which may be stationary. The clutch provides two or more friction surfaces through which the necessary slip is taken when two differing speeds of rotation are to be synchronised.

Through the many transitional stages from cone to multiplate, thence to the single plate, the weight of the driven member, which controls the period of time necessary to stop rotation when the transmission is disengaged, has been a factor which, in conjunction with simplicity of construction, has contributed to the adoption of the single-plate type as a standard in the majority of automobiles of today. There are, of course, those vehicles which adopt the fluid-flywheel form of coupling, which was designed to damp out shocks and stresses incurred (incident with the connections of an inert shaft to one experiencing revolution), but this type of coupling demands special and separate treatment.

The plate-clutch designer has to decide upon the most suitable co-efficient for reserve factor, friction and pressure necessary, whilst in addition to consideration of pressure-plate dimensions and spring load required, other features, such as thermal characteristics, flexibility, etc., enter into the general design. The initial calculations concern the inner and outer diameter of plates in terms of engine torque, and the required spring load. The torque which the clutch should be designed to transmit should exceed that of maximum engine output, and an additional slip torque of 40 per cent has been found to cover all requirements of liner wear and minor causes which offer resistance to efficient operation. If possible it should be arranged that the facing should also have a 10 per cent additional absorption capacity when the liner is almost worn away. The first formula allows the desired coefficients and the relationship between inner and outer diameter of liners in terms of engine torque to be estimated.

If $F$ = excess torque to be absorbed
$T$ = maximum engine torque (ft. lb)
$D$ = outer diameter of friction surface (in)
$d$ = inner diameter of friction surface (in)
$\mu$ = coefficient of friction
$P$ = unit pressure at surfaces (lb/in$^2$)
$E$ = efficiency of operating mechanism
$L$ = load on springs

then

$$140\% \text{ or } F + T = 2\frac{\pi}{4}(D^2 - d^2)\,P\mu\,\frac{D + d}{4}\,\frac{1}{12}$$

or if d = cD where c = 0·75 (I.A.E. Data Sheet, 144)

$$140\% \text{ or } F + T = \frac{\pi}{96}\,P\mu\,D^3\,(1 + c - c^2 - c^3)$$

in which case

$$D = 3{\cdot}125\sqrt[3]{\frac{F + T \text{ or } 140\%T}{P\mu\,(1 + c - c^2 - c^3)}} \quad \text{or} \quad 3{\cdot}42\sqrt[3]{\frac{F+T}{P\mu}}$$

For spring load required

$$L = \frac{\pi}{4}\frac{P}{E}(D^2 - d^2) \qquad \text{or transposing} = 0{\cdot}344\,\frac{PD^2}{E}$$

It is of course possible that a clutch of the diameter indicated by the foregoing formulae cannot be accommodated, in which case a revised relationship between D and d must be chosen, although D is usually the controlling dimension. If D is definitely fixed then d should equal approximately one-third D, in which case maximum torque would equal 0·0388 $P\mu D^3$.

The energy-absorption capacity of the single-plate clutch varies, but a safe figure, provided the fabric is compressed and is of the asbestos-base type, is 25,000 to 30,000 ft. lb/in² of facing, so long as the temperature remains below 450° F. Surface velocity also should not exceed 6,000 ft/min at the effective radius, whilst the pressure upon the lining face should not be more than 12 lb/in² if long life is to be obtained.

The quality or grade of friction lining must be carefully considered in conjunction with the material it is running against. The rate of wear and the permissible unit pressure for the lining chosen must also be the subject of the most careful scrutiny; for instance, a fibre lining should not be used where temperatures are likely to exceed 220° F, nor should any liner which does not consist of an asbestos base be considered for temperatures exceeding 350° F. Such fabrics are most likely to char at higher temperatures. It is therefore desirable to specify material for the rubbing parts to possess sufficient thickness to absorb heat units generated, whilst at the same time the external surfaces must provide the maximum area for heat radiation.

Generally, within the limits of permissible surface loading, the rate of wear is directly proportional to the work done at constant temperature, but above the recommended temperature for the chosen liner the rate of wear bears no proportionate value to any further increase. Therefore the two factors which will determine the liner dimension will be the rate of dissipation of energy and the normal operating temperature.

A clutch for average duty will give satisfactory service if the design is based on a mean rate of energy dissipation of 6,000 ft. lb/in² of liner area.

The means of liner attachment to plate must be capable of carrying the total torque to be transmitted and thus the load on each rivet must be ascertained. An empirical formula which has been evolved, using a factor of safety of 3 for *each* rivet for automobile clutches is the square root of the projected area of rivet multiplied by a constant or $1{,}550\sqrt{A}$.

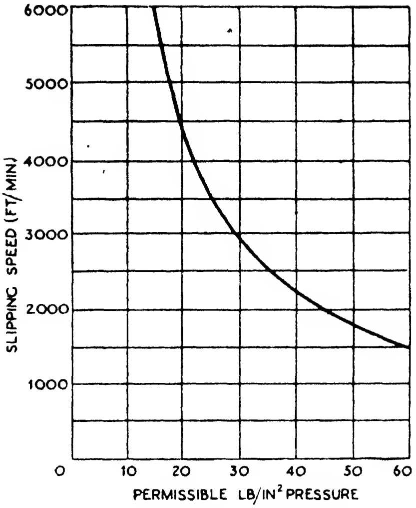



Fig. 87—Slipping speed/unit pressure curve

The slipping speed and correct unit pressure call for careful consideration, as the temperature of the rubbing parts must be kept as low as possible. For slow slipping speeds, of course, the pressure may be increased, and reference to Fig. 87 shows the approximate relationship between the two values.

To calculate the pressure in lb/in² of liner area the following expressions are sufficiently accurate:

Torque to be transmitted $= T = \dfrac{HP \times 5{,}250}{r/min}$

Tangential effort $= E = \dfrac{12T}{R}$, where R = mean radius of lining (in).

Total lining pressure $= P = \dfrac{E}{\mu \times n}$, where $\mu$ = lining friction co-efficient and n = number of linings.

Lining pressure lb/in² $= p = \dfrac{P}{A}$, where A = area of clutch lining (in).

The slipping speed is calculated from the speed at which the power is to be transmitted and the mean diameter of lining or

$$\frac{\pi D}{12} \times r/min = ft/min.$$

Fig. 87 is plotted on the basis of one lining and if two are fitted then the figure obtained for unit pressure must be halved. It is useful to compare the friction coefficient of different linings:

Ferodo RAD11 - dry 0·28

Ferodo RAD5 – lubricated 0·1
Ferodo MR – dry brakes 0·35 } may be used with materials
Ferodo MR – dry clutches 0·30 } with Brinell as low as 150.

The necessary application of axial force applied to the plate is usually by a series of coil springs of small diameter, although in some designs a large conical spring supplies the load. Alternatively several nests of two springs each, housed in a suitable cup, perform very satisfactorily. It should be borne in mind that the peripheral coil springs are subjected to centrifugal force in addition to their main function. The basic formulae for round-wire helical springs are quite straightforward and are, where

d = diameter of wire (in)
R = mean radius of coil (that is to centre of wire)
N = number of coils active
C = modulus of transverse elasticity $11 \cdot 5 \times 10^6$
K = stress intensity (10,000 to 12,000)
W = maximum safe load (lb)
F = compression or extension of spring (in)
S = maximum safe fibre stress (lb/in$^2$)
G = torsional modulus of elasticity
P = applied load (lb).

Note: S may be taken as 60,000 lb/in$^2$ for springs working under abnormal temperatures, 80,000 lb/in$^2$ for normal conditions and 100,000 lb/in$^2$ for applications not subjected to large stress variation.

G = 12,000,000 for $\frac{1}{8}$ in diameter wire and over
12,500,000 for $\frac{1}{16}$ in to $\frac{1}{8}$ in diameter wire
13,000,000 for under $\frac{1}{16}$ in diameter.

$$\text{Rate} = \frac{Cd^4}{NR^3}$$

$$\text{Maximum safe load} = W = \frac{Kd^3}{R} \text{ or } \frac{\pi S d^3}{16R}$$

$$\text{Compression of spring } F = \frac{16NPR}{Gd^4}$$

Total number of coils $= N + 1 \cdot 5$
Dead length $= d \times (N + 1)$
Weight of spring $= 0 \cdot 7d^2\, D\, (N + 1 \cdot 5)$
Weight of spring for surge frequency only $= 0 \cdot 7d^2\, DN$

$$\text{Surge frequency} = 590\sqrt{\frac{\text{Rate}}{\text{Weight}}}$$

*Table* 6

Table of Constants for Helical Springs of Round-steel Wire

| $\frac{D}{\text{S. W. G.}}$ | Kd | Cd | K |
|---|---|---|---|
| 6 | 105·5000 | 244·61560 | 14,928 |
| 7 | 82·7300 | 172·65600 | 15,184 |
| 8 | 63·2500 | 117·96480 | 15,440 |
| 9 | 46·7600 | 77·24160 | 15,696 |
| 10 | 33·5000 | 48·38400 | 15,952 |
| 11 | 25·1800 | 32·57280 | 16,144 |
| 12 | 18·3700 | 21·06000 | 16,336 |
| 13 | 12·8500 | 12·88368 | 16,528 |
| 14 | 8·5620 | 7·37280 | 16,720 |
| 15 | 6·2840 | 4·83400 | 16,848 |
| 16 | 4·4470 | 3·01824 | 16,976 |
| 17 | 3·0030 | 1·77000 | 17,104 |
| 18 | 1·9050 | 0·95560 | 17,232 |
| 19 | 1·1110 | 0·46080 | 17,360 |
| 20 | 0·7769 | 0·30196 | 17,424 |

In the case of nested springs the treatment and calculations are somewhat different. It will simplify matters at a later stage to consider the nest as possessing no clearance between coils of the inner and outer springs, and of course it is understood that in order to avoid jamming the coils must be wound right and left hand alternately (see Fig. 88).

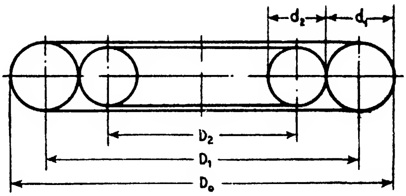



Fig. 88

The stresses are expressed in terms of functions of spring index $\frac{D}{d} = C$. For a given deflection $\delta$ the most economical nest will have equal stresses and equal solid length, in which case the indices C will be similar for all springs in the nest. If the outer diameter is represented by symbol $D_0$ then in terms of $D_0$ the wire and coil diameters will be

$$d_1 = \frac{D_0}{\frac{D}{d} + 1} \text{ and } d_2 = \frac{\frac{D}{d} - 1}{\frac{D}{d} + 1} \cdot \frac{D_0}{\frac{D}{d} + 1}$$

$$\text{and } D_1 = \frac{\frac{D}{d} \times D_0}{\frac{D}{d} + 1} \text{ and } D_2 = \frac{\frac{D}{d} - 1}{\frac{D}{d} + 1} \cdot \frac{\frac{D}{d} D_0}{\frac{D}{d} + 1}$$

where d = cross section of wire
D = mean diameter of coils.

The stiffness of the nest will be, since solid length and $\frac{D}{d}$ are constant for both springs,

$$S = \sum \frac{Gd^4}{8nD^3}$$

where G = modulus of rigidity of $11{\cdot}5 \times 10^6$ lb/in²

$$\text{or} \quad S = \frac{GD_0^2}{32L} \left[ \frac{1 - \left(\dfrac{\frac{D}{d} - 1}{\frac{D}{d} + 1}\right)^{2N}}{\left(\frac{D}{d}\right)^4} \right]$$

notation for which L = solid length of spring and N = number of springs in the nest. The stiffness or rate of the springs will of course vary as $d^2$, whilst the total load P multiplied by $\frac{d^2}{\Sigma d^2}$ will be carried by an individual spring.

The stress is therefore constant for all springs in the nest and if f = stress

$$\text{then the value will be} \quad f = \left( \frac{8PD}{\pi d^3} \cdot \frac{d^2}{\Sigma d^2} \cdot \frac{\frac{D}{d} + 0{\cdot}2}{\frac{D}{d} - 1} \right)$$

$$\text{which can be evaluated } f = \frac{8P}{\pi D_0^2} \cdot \frac{\frac{D}{d} + 0{\cdot}2}{\frac{D}{d} - 1} \cdot \frac{4\left(\frac{D}{d}\right)^2}{1 - \left[\dfrac{\frac{D}{d} - 1}{\frac{D}{d} + 1}\right]^{2N}}$$

If this expression is combined with that for stiffness

$$f = \frac{G\delta}{\pi L} \cdot \frac{\frac{D}{d} + 0{\cdot}2}{\left(\frac{D}{d}\right)^2 \left(\frac{D}{d} - 1\right)}.$$

*Table 7*

| N = 1 | | N = 2 | | N = 1 | | N = 2 | |
|---|---|---|---|---|---|---|---|
| 0·084 | 48 | 0·082 | 43 | 10·0 | 40 | 9·5 | 30 |
| 0·132 | 5 | 0·125 | 4 | 14·0 | 4 | 12·5 | 4 |
| 0·137 | 5 | 0·129 | 4 | 14·4 | 5 | 12·9 | 5 |
| 0·142 | 5 | 0.133 | 5 | 14·9 | 5 | 13·2 | 4 |
| 0·147 | 5 | 0·138 | 4 | 15·4 | 5 | 13·6 | 3 |
| 0·152 | 5 | 0·142 | 5 | 15·9 | 4 | 13·9 | 4 |
| 0·157 | 5 | 0·147 | 4 | 16·3 | 5 | 14·3 | 4 |
| 0·162 | 5 | 0·151 | 5 | 16·8 | 5 | 14·7 | 3 |
| 0·167 | 5 | 0·156 | 4 | 17·3 | 5 | 15·0 | 4 |
| 0·172 | 5 | 0·160 | 5 | 17·8 | 6 | 15·4 | 4 |
| 0·177 | 6 | 0·165 | 4 | 18·4 | 5 | 15·8 | 4 |
| 0·183 | 5 | 0·169 | 5 | 18·9 | 5 | 16·2 | 4 |
| 0·188 | 5 | 0·174 | 4 | 19·4 | 5 | 16·6 | 4 |
| 0·193 | 6 | 0·178 | 5 | 19·9 | 6 | 17·0 | 4 |
| 0·199 | 5 | 0·183 | 5 | 20·5 | 5 | 17·4 | 4 |
| 0·204 | 5 | 0·188 | 4 | 21·0 | 6 | 17·8 | 4 |
| 0·209 | 6 | 0·192 | 5 | 21·6 | 5 | 18·2 | 4 |
| 0·215 | 5 | 0·197 | 5 | 22·1 | 6 | 18·6 | 4 |
| 0·220 | 5 | 0·202 | 4 | 22·7 | 6 | 19·0 | 5 |
| 0·225 | 6 | 0·206 | 5 | 23·3 | 5 | 19·5 | 4 |
| 0·231 | 6 | 0·211 | 5 | 23·8 | 6 | 19·9 | 4 |
| 0·237 | 5 | 0·216 | 5 | 24·4 | 6 | 20·3 | 4 |
| 0·242 | 6 | 0·221 | 4 | 25·0 | 6 | 20·7 | 5 |
| 0·248 | 5 | 0·225 | 5 | 25·6 | 6 | 21·2 | 4 |
| 0·253 | 6 | 0·230 | 5 | 26·2 | 6 | 21·6 | 5 |
| 0·259 | 5 | 0·235 | 5 | 26·8 | 6 | 22·1 | 4 |
| 0·264 | 6 | 0·240 | 5 | 27·4 | 6 | 22·5 | 5 |
| 0·270 | 6 | 0·245 | 5 | 28·0 | 6 | 23·0 | 4 |
| 0·276 | 5 | 0·250 | 5 | 28·6 | 6 | 23·4 | 5 |
| 0·281 | 6 | 0·255 | 4 | 29·2 | 6 | 23·9 | 4 |
| 0·287 | 6 | 0·259 | 5 | 29·8 | 6 | 24·3 | 5 |
| 0·293 | 6 | 0·264 | 5 | 30·4 | 7 | 24·8 | 5 |
| 0·299 | 5 | 0·269 | 5 | 31·1 | 6 | 25·3 | 4 |
| 0·304 | 6 | 0·274 | 5 | 31·7 | 7 | 25·7 | 5 |
| 0·310 | 6 | 0·279 | 5 | 32·4 | 6 | 26·2 | 5 |
| 0·316 | 6 | 0·284 | 5 | 33·0 | 7 | 26·7 | 5 |
| 0·322 | 6 | 0·289 | 5 | 33·7 | 6 | 27·2 | 4 |
| 0·328 | 6 | 0·294 | 5 | 34·3 | 7 | 27·6 | 5 |
| 0·334 | 6 | 0·299 | 5 | 35·0 | 7 | 28·1 | 5 |
| 0·340 | 6 | 0·304 | 5 | 35·7 | 6 | 28·6 | 5 |
| 0·346 | 6 | 0·309 | 5 | 36·3 | 7 | 29·1 | 5 |

Three functions for spring index and number of springs can now be chosen such that $C_1$ $C_2$ which are functions of $\frac{D}{d}$ and N the number of springs in nest equal.

$$\sqrt[4]{\frac{D_0^2}{ls}} = C_1$$

$$D_0\sqrt{\frac{f}{P}} = C_2.$$

Note, $C_3$ (if three springs) would equal $\sqrt{\frac{\delta}{Cf}}$. Values of $C_1$ and $C_2$ are given in Table 7 :

$$C_1 = \sqrt[4]{\frac{D_0^2}{ls}} \qquad\qquad C_2 = D_0\sqrt{\frac{f}{P}}$$

Now consider clearance is allowed between springs and springs and housing, and let notation $\alpha$ represent the clearance factor, whilst further notation is shown in Fig. 89. Then clearance at outer diameter of outer springs $= \left[\frac{(1-\alpha)}{2\alpha}\right]d$, that between the outer of the inner spring and the inner diameter of the outer spring $=\left(\frac{1-\alpha}{2\alpha}\right)(d_1+d_2)$. Should there be a third spring in the nest the last part of the expression will be substituted by $(d_2 + d_3)$ and will become $\left(\frac{1-\alpha}{2\alpha}\right)(d_2 + d_3)$.

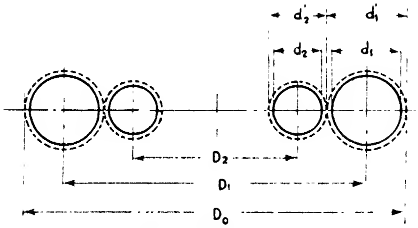


**Fig. 89—Nested springs**

In the figure the dotted lines represent nominal springs having wire diameters of $d'_1$, $d'_2$, in which the connection between the actual and nominal diameters are expressed by the equations $d_1 = \alpha d'_1$, $d_2 = \alpha d'_2$. The spring index $\frac{D}{d} = C$ is therefore modified to become $C'$ and equals $C' = \alpha C$ or $\alpha\frac{D}{d}$ and if $P'$ $S'$ $f'$ nominal spring notation replace P S and f for the actual springs having the same deflection $\alpha$

$$P' \text{ will equal } \frac{P}{\alpha^5}$$

$$S' \text{ will equal } \frac{S}{\alpha^5}$$

$$f' = f \frac{C + \frac{0.2}{\alpha}}{C + 0.2} \cdot \frac{C - 1}{C - \frac{1}{\alpha}} \div \alpha^2.$$

If these modified values are utilised in the design of nominal springs with no clearance and wire diameters at $d'_1$, $d'_2$, etc., the actual springs will possess wire diameters $\alpha d'_1$, $\alpha d'_2$, which will correspond to the actual values of P, S and f, although the actual stress will be slightly on the low side, $\alpha$ for wire diameters of $\frac{1}{8}$ in, or 10 S.W.G. and over, will be found suitable at 0·9, decreasing as the gauge decreases to 0·7 in the smaller gauges.

For circular-section springs manufacturing tolerances must be allowed in order to compensate for the variations in wire diameter and material. Should the spring rate necessitate close limits of, say, $\pm$ 4 or 5 per cent, a nominal value only for the number of coils should be specified, but if it is necessary to specify all the spring dimensions nominal values only of the spring rate must be acceptable in the order of 12 per cent tolerance up or down. The value of the spring index C or $\frac{D}{d}$ for circular sections should lie within the range of 3·5 to 8, according to the required stiffness value, and in the case of circular section wire 6·0 is a good average.

It is, however, conclusive that the best cushion design depends upon the vehicle characteristics, as some vehicles have a greater inclination to chatter under light torque than others, in which case a soft initial cushion is desirable. On the other hand, the vehicle may exhibit its worst tendency to chatter under heavy plate load and high torque, and in this event it is essential to have the deflection rate as low as possible. Since the initial spring deflection rate is stiffer, the final deflection rate will be softer for the same total cushion.

The choice of material for helical-coil springs depends to some extent upon the wire diameter. Generally, hard-drawn wire is specified for diameters up to $\frac{1}{4}$ in and a carbon or alloy steel rod for diameters over $\frac{3}{8}$ in, so that between the section of $\frac{1}{4}$ in and $\frac{3}{8}$ in diameter the choice is at the discretion of the designer. STA 2a and STA 2b, both carbon or alloy steel rod, are suitable for the larger sections, but the former withstands slightly lower stresses than the STA 2b, which is a silicro-manganese steel; STA 2c is a chrome-vanadium steel and there is little to choose between this and 2b. STA 2a is in a slightly lower category and can be used in designs wherein the stresses are not too high and the torsional elastic limit does not exceed 100,000 lb/in².

With regard to the permissible stress, it is not merely dependent upon the maximum stress at full compression, but upon the material and performance required, and should be considered as a ratio of maximum to minimum stress, or the difference between the stress at initial load and the stress at final load, which represents the "stress range". Figs. 90 and 91 show the maximum permissible

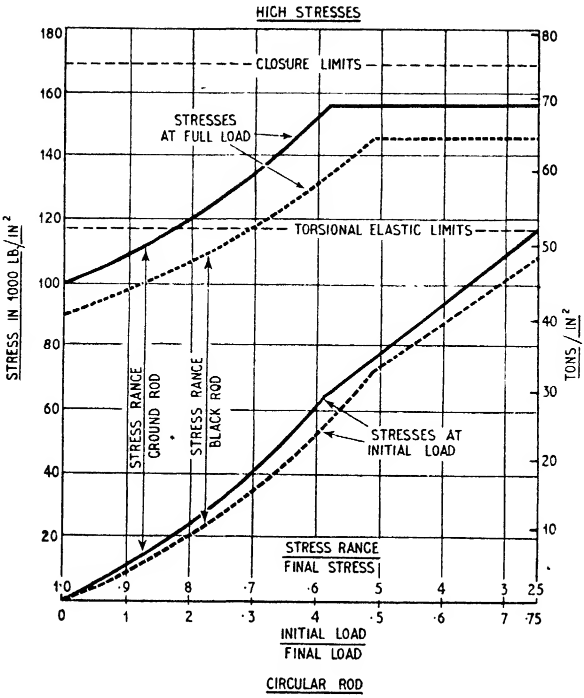
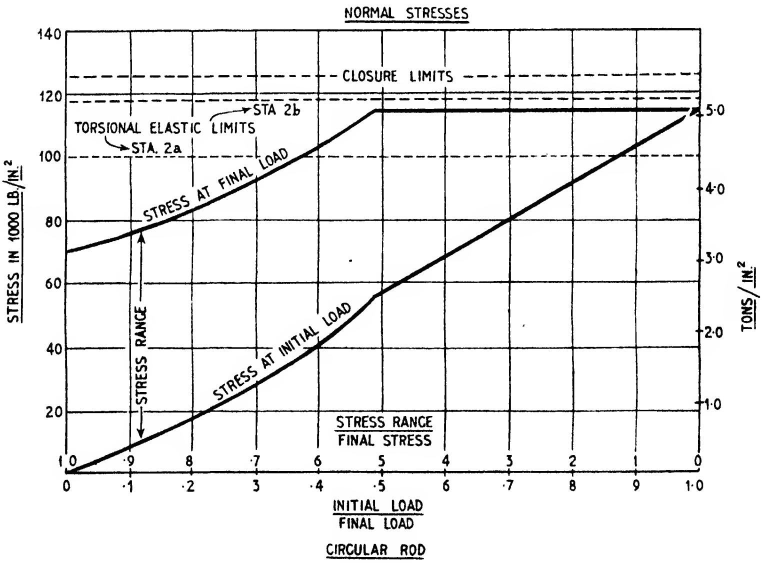



Fig. 90 — Initial/final load ratio

Fig. 91 — Initial/final load ratio



stresses against the ratio of initial to final load for high and normal stresses for alloy steel to STA 2a or STA 2b.

Where the stress range is less than 25 per cent of the maximum stress, high stresses should not be used, and for hard-drawn steel wires the maximum stress, which is related to the ultimate stress, is shown as a percentage in Fig. 92.

For carbon and alloy steels the maximum stress depends upon the torsional elastic limit of the material, which may be exceeded for springs of limited endurance if the process of "scragging" is adopted.

Incidentally it is good practice to fit an insulator plate between the spring end and the pressure plate to reduce the possible effect of plate overheating upon the temper of the spring material.

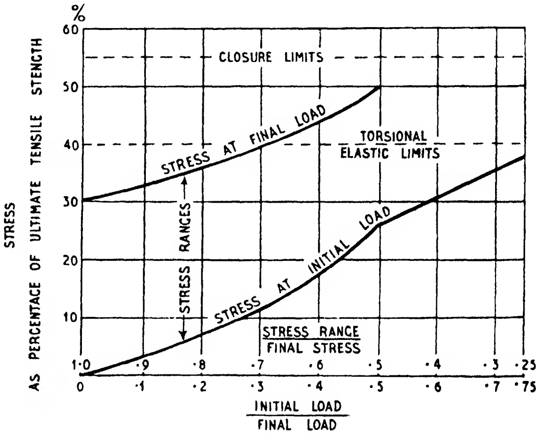



**Fig. 92—Maximum/ultimate stress ratio**

The driven plate must take up the load from the driving disc gradually and therefore the driven plate must possess a certain degree of resilience in a direction perpendicular to its axis. Many devices and forms of plate have been evolved to satisfy this condition, but not all have achieved the desired result. Any design which tends to produce a saucer-shape friction disc causes the disc to deviate from parallelism in its approach towards its mating surface, which condition not only tends to produce chatter but actually accentuates any inherently similar qualities possessed by the engine or transmission. The essentials, therefore, in the plate design must provide for progressive engagement of the surfaces and parallel approach.

It was at one time thought that a crimped or corrugated plate would ensure the necessary cushioning to permit progressive engagement, but it would appear that results indicated non-uniform deflection of the raised portions, and consequent coning of the liners. Wear under such conditions commences at the outer periphery and gradually works towards the inner diameter of the liner. It must not, of course, be inferred that all efforts have been unsuccessful; one make at least is very popular and in this case the plate is cut-away around its periphery in a series of shapes as shown in Fig. 93, in which the gaps so formed are shaded.

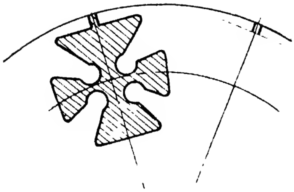


Fig. 93

All the four tongues are set to the same direction of deflection away from the plane of the main disc. Two tongues in each, cut away and diametrally opposite one another, are set to the same dimension of deflection, whilst the remaining two are set to a slightly lesser degree. The circular portion of each tongue extremity lies in the same plane as the disc face. The linings are attached on either side of the plate, on one side through the two highest tongues and on the other side at points between the cut-away gaps. Engagement therefore becomes progressive; also, since the points of attachment are uniformly spaced between the inner and outer diameters of the disc, the facings are parallel to the main face of the disc.

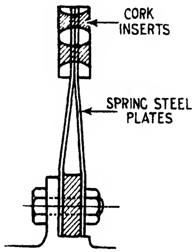



Fig. 94

Another method of providing resilient connection between the fabric and steel plate is the adoption of cork rivets. It is claimed, for instance, that a continuous face is obtained free from rivet recesses and the increased torque-transmitting capacity is consequently considerably increased. Fig. 94 illustrates this construction, the spring steel plates affording axial flexibility, due to their separation at the inner diameter by a distance disc. This spacer washer increases the disc thickness at the inside diameter by springing open the plates and it is at this point that initial engagement takes place. The disc gradually closes as pressure increases, until under full pressure the discs are parallel in thickness. The opposite procedure takes place during disengagement and absence of drag is therefore ensured.

Still one more example of cushioned engagement and parallel approach is provided by the arrangement as shown in Fig. 95. In this arrangement two plates are employed, as in the previous type. In this case the plates are parallel and are spaced at the outer diameter by several five-pronged dished-steel stars which surround each compound rivet. Each star is alternatively fitted so that the points are in contact with the opposite plate.

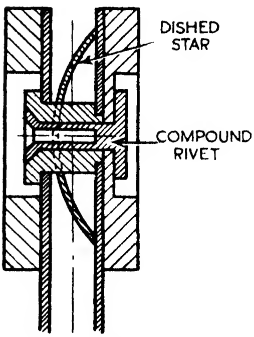



Fig. 95

Incidentally, this arrangement permits a flow of air between the plates, which it must be conceded is a decided advantage. Under engagement the spring-steel star-pieces are depressed so that the minimum width can be predetermined, and by varying the size and quantity of these discs the end load may be calculated so that it is in proportion to the engine torque. For heavier clutches these initially-loaded discs are fitted in two rows. The riveting of the fabric is interesting as it comprises a copper centre, pressed into position, whilst the head is supported on an anvil and the open end enlarged. From its construction it will be seen that it is impossible to over-rivet and consequently crush the liner.

Various engines and transmissions possess, of course, various characteristics and it is sometimes necessary to eliminate or break up torsional vibrations emanating from these sources. One such device consists of the introduction of a series of coil springs mounted round the periphery of the clutch plate. See Fig. 96, in which any angular movement of the disc relative to the clutch centre causes sliding motion and consequently has a damping effect upon any torsional vibration at the crankshaft end. The springs should be carefully calculated as it is essential that they carry the full torque of the engine, and they must be capable of long life without failure due to fatigue. They must also be designed to suit the particular car characteristics.

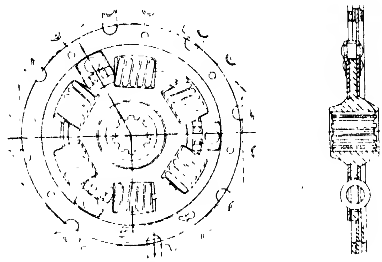


**Fig. 96—Spring characteristics of clutch damping**

Rubber, with its unique capacity to absorb vibration, has also been used as a clutch centre. In this respect, and in order to obtain the best results, the rubber must be so arranged as to be protected from excess heat, whilst it should be initially compressed so that it may withstand high stresses. However, the material should be arranged if possible to be uniformly stressed throughout its volume. There are many varying forms of these basic applications. With these resilient means of reaction it would be possible to provide that the axial thrust on the friction surfaces be applied in an ascending ratio by utilising the centrifugal force, in some way applying its magnitude through the toggle levers. It would then be possible for the clutch to remain in engagement during deceleration until the speed falls to a value equal to, or below, that at which initial engagement is effected.

Another example of the use of rubber to form a resilient clutch centre is shown in Fig. 97. The centre consists of an inner and outer cam plate, between which is interposed rubber, which has been bonded to each of these steel cams, by special bonding process. On the flanks of the points the rubber is relieved by air gaps. These gaps are for the relief of local stress concentrations. In Fig. 97 A shows the coupling at rest and B denotes the effect upon the elements

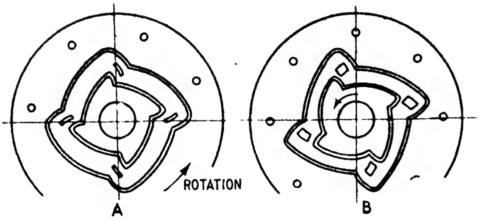



Fig. 97

of the centre when the resistance of the outer member exceeds the torque upon the inner cam.

The flanks at points of the inner cam have moved rotationally in relation to those of the outer cam and the rubber insert has been placed partly in shear and partly in compression, whilst the short portions between the radial flanks are in tension. These types of stress are reversed upon overrun and thus provide a different load characteristic.

It is possible to design the clutch centre so that the rate of resistance rise is rapid or slow, with various load characteristics, by shaping the cams to control the possible combinations of shear, compression and tension. For instance, if the flanks of the cam points are symmetrical on both sides, the characteristics will be similar for either direction of relative rotation. The rubber insert is shaped as shown in Fig. 98 so that the material is evenly stressed throughout.

In this coupling the resistance to compression as a result of high torque varies approximately as the torque and enables the rubber to withstand possibly greater shear stresses than apply in the ordinary shear-type coupling.

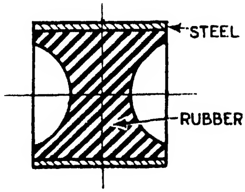



Fig. 98

One of the fundamental features of a clutch is that it should be capable of balancing engine-torque and road-traction requirements, and that it cannot be subjected to destructive slip. Such a condition is very difficult to achieve in the normal pedal-operated clutch. It is also desirable that the clutch should act as a shock absorber in the transmission, and should not become fully loaded suddenly. In contrast to the designs which have sought to achieve this objective, through the construction of the clutch-plate attachment to the friction lining, automatic devices have been produced which claim to produce varying pressure upon the friction surfaces in accordance with and proportional to the speed of revolution of the engine.

The use of centrifugal force in conjunction with weighted bell-crank levers suggests a solution, but until recently most of such designs possessed the inherent difficulty that unless the engine be running and be revved up to engagement speed, the clutch could not be engaged. Such difficulties have been overcome and one successful arrangement provides for the inclusion of centrifugal weights mounted in the flywheel which under the influence of revolution operate

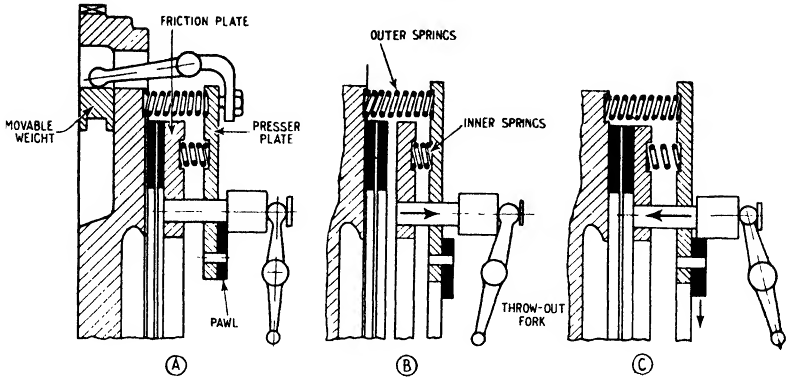



**Fig. 99—(A) When engine is running the pawls fly outwards and are kept in position by centrifugal force; (B) when engine is stationary and the clutch pedal depressed, the pawls are disengaged; and (C) when the pedal is released the clutch is engaged**

bell-crank levers arranged in such a manner that they operate the pressure plate and thus the friction plates.

The mechanism possesses most of the usual single-plate clutch components, but an additional set of coil springs is interposed between the pressure plate and the spring plate, their function being to ensure the assembly of pressure plate and spring plate moving away from the flywheel when the engine is running at idling speed, so that the clutch is automatically disengaged. Reference to Fig. 99 will also make clear the method of centrifugal-force application. Three pawls are pivoted on the spring plate and light springs tend to keep the pawls out of engagement, although not strong enough to prevent the force, even at idling speed, from moving outwards against the studs and becoming trapped by the pillar studs should the engine be switched off.

## SPECIAL CLUTCH DESIGNS

By utilising centrifugal force to increase plate pressure with increase of engine speed, it is natural to expect light pedal pressures at engine idle speeds if light-pressure plate springs are adopted. However, in the majority of such designs this advantage is practically nullified at high engine speeds on account of the combined effect of spring pressure and centrifugal force which must be overcome to provide clutch release. Most of such types of centrifugal clutches possess the inherent disadvantage that they depend upon engine speed for engagement or disengagement. The introduction of hydraulic, electric and pneumatic control have not entirely eliminated all production difficulties, although certain types of air-operated controls, using manifold vacuum as a source of power, have achieved a certain degree of success. Most of such systems fall within one of three categories:

(1) Balanced or pressure-sensitive types.
(2) Balanced cushion with variable bleed.
(3) Follow-up type valve.

Number 1 category permits the driver to change gear without the use of the clutch pedal, which is, however, retained for its use in smooth engagement during cold starting. In Number 3 type, clutch engagement is regulated only by accelerator-pedal movement, and considerable skill is required to secure smooth engagement in all gear ratios. The design is not intended to eliminate the use of the clutch pedal, but to reduce the number of operations required. Type Number 2 is the simplest form of construction and incorporates a cylinder-and-diaphragm type piston, a control valve and a governor switch, throttle switch and overruling switch.

Reverting to *the pressure-sensitive type:* In this system the speed of air bleed into the vacuum cylinder controls the rate of clutch engagement. It is independent of the manifold vacuum and therefore clutch engagement is not affected by variation in this value. The rate of air bleed is controlled by a primary regulator valve interconnected with the accelerator pedal and throttle opening. To augment this control when starting up in low gear solenoid-operated valves are incorporated. This system of control ceases to function at speeds above approximately 15 m.p.h. as there would otherwise be a tendency to freewheel above this speed. Provision to overrule the governor is necessary in order that gear changes may be made out of top gear above the governed speed, whilst to secure braking downhill when in second gear a lockout switch is brought into operation. These automatic controls form a somewhat elaborate system, which must be serviced and kept in perfect order, otherwise the system ceases to function properly.

The system of *balanced-cushion vacuum control* is much more direct. It comprises an integral power cylinder and control valve, the piston being fitted direct to the solenoid valve and regulator valve. The energised solenoid valve is opened to manifold vacuum with the accelerator pedal in the engine-idle position, below any engine speed at which the governor maintains the electrical circuit to energise the solenoid. At the same time that manifold vacuum is admitted to the power cylinder it is also admitted to the regulator, against the spring-load action upon the diaphragm which, although less than the force exerted by the full vacuum, does not allow the regulator valve to open, since the regulator diaphragm is vacuum suspended when the clutch is in the released position. Movement of the accelerator pedal breaks the energising circuit to the solenoid valve whilst, as the throttle opening increases with further pedal depression, the bleed-valve orifice also increases with consequent quick engagement for full open-throttle acceleration. This system, as in the pressure-sensitive type, incorporates a governor switch to prevent freewheeling above agreed speeds, and an overrule switch to permit gear change in excess of these speeds (see Fig. 100).

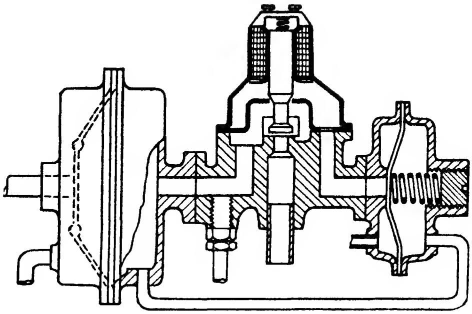


Fig. 100—Balanced cushion-type vacuum control

*The follow-up type of control* also employs a cylinder, but in this case the control valves are incorporated in the piston rod, which in turn is connected to the accelerator pedal through a toggle linkage. This linkage also forms the connection between the clutch-release shaft and piston rod. Accelerator-pedal movement closes the cylinder to atmosphere by the valve-rod movement, which opens certain ports, permitting a pressure balance on both sides of the piston. The vacuum ports are sealed off since the speed of piston-rod movement is greater than that of the valve rod, and in consequence the piston-rod ports overtake the valve plunger, the resulting follow-up action taking place throughout the stroke.

Should movement of the accelerator pedal close before full engagement the piston-rod valve will check engagement. It is for this reason that considerable skill is required to secure smooth engagement in all gear ratios, since the rate of clutch engagement varies with the gear ratio in which engagement is taking place, when once the cushion point has been reached.

## THE GEARBOX

The maximum torque which a gear will transmit is known to be proportional to the square of the tooth thickness at the base, and also to the face width and pitch diameter. It is inversely proportional to the height of the tooth. Similarly the torque capacity of a gear set varies as the cube of its linear dimensions. Of these dimensions, (a) the shaft-centre distance and (b) dimensions between bearings, are two of the most important (see Fig. 101).

In the original gearbox layout, a fairly accurate start can be made by assuming the *shaft-centre distance* conforms to $0{\cdot}5\sqrt[3]{T}$ in, or $0{\cdot}8\sqrt[3]{T}$ (the former for private cars, and the latter for trucks) and *the bearing centres* at $1{\cdot}2\sqrt[3]{T}-1{\cdot}5\sqrt[3]{T}$ in, T being maximum engine torque in lb/ft. This latter dimension naturally depends finally upon the gear-face widths and gear movements and, instead of tedious calculations in the initial stages of design, an approximation of *face width* may be taken as represented by the expression $\frac{Lp}{k}$, where L = maximum permissible load on tooth at pitch circle and p = normal diametral pitch.

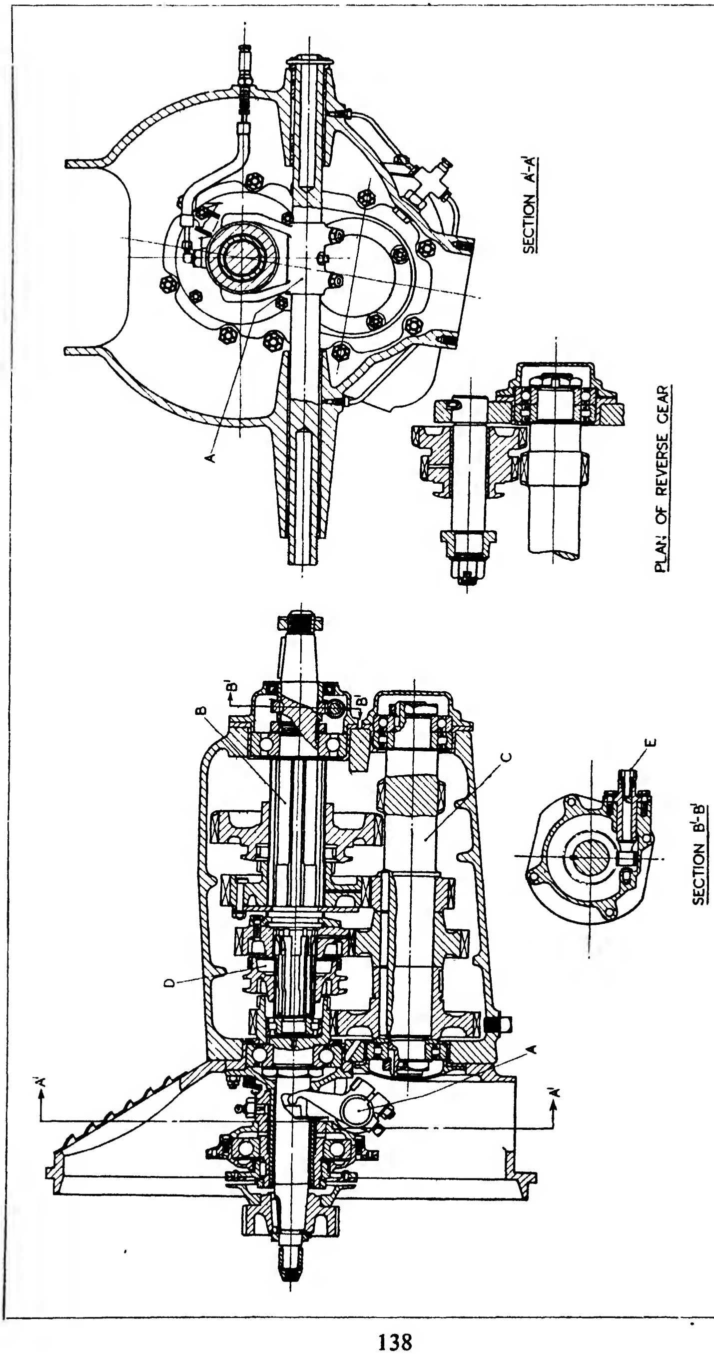



**Fig. 101—Typical arrangement of a gearbox**

A, clutch withdrawal lever; B, main shaft; C, lay shaft; D, gear engagement collar; and E, speedometer drive

K is a constant—
11,000–14,000 for the first reduction gear in a four-speed box
14,000–16,000 for third speed
16,000–21,000 for second speed
26,000–30,000 for first speed.

For a five-speed box these figures are slightly revised to—
13,000–15,000 for fourth speed
15,000–17,000 for third speed
20,000–22,000 for second speed
26,000–30,000 for first speed.

*The diametral pitch* is determined by the centre distance, the ratios required, and the helix angle of the tooth, bearing in mind that none of the main wheels should have less than, say, fifteen teeth. Under the section " Vehicle Performance " the question of suitable gear ratios is discussed, together with the selections according to geometric progression. The total number of teeth in any two pairs of mating gears is usually the same. There are three implications of this—that the helix angle is the same for all pairs of gears; that the diametral pitches are different in each pair to compensate for any difference in helix angle, or, finally, that spur gears are adopted.

In the choice of helix angle two considerations should be borne in mind, (a) that it is desirable that the circular helical advance over the face width should be at least equal to the circumferential pitch in order that tooth contact is maintained on the pitch line at some point, and (b) that the thrust load emanating from the first reduction set should not be substantially exceeded by thrust from the other gear sets, as these thrusts are in the opposite direction and therefore approximately cancel out.

So much has been written and so many excellent works are available regarding tooth data, profiles, strengths and calculations, that it is considered desirable that the designer should consult his chosen volume, as the subject cannot be treated adequately within the confines of this restricted space.

One point, however, is worthy of note, since one of the major requirements of gearbox design concerns quietness of running and maximum efficiency. These conditions can be achieved if the selected gear teeth are " corrected," that is if the addendum is increased and the dedendum decreased in the pinion and vice versa in the wheel. It is necessary to examine the tooth action to appreciate the utility of this operation. Refer to Fig. 102.

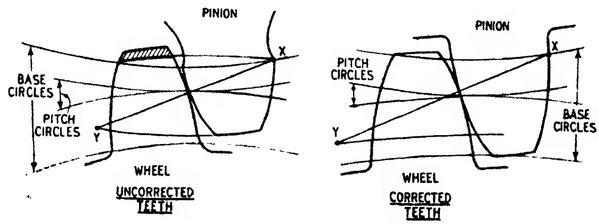



Fig. 102

The pinion tooth in Fig. 102 shows a partly-destroyed dedendum above the base-circle diameter. The extent is marked by letter X and any point below this position cannot make contact with the tooth in the wheel. The shaded portion of the wheel tooth represents the redundant material which could be removed without affecting tooth action. Correspondingly, the dedendum of the pinion tooth could be reduced, which would automatically increase its strength in addition to eliminating some of the tooth undercutting. However, continuity of action must take place and before disengagement of one tooth another must engage. The distance along the line of action XY is equal to the circular pitch on the base-circle diameter, or

$$\text{Base pitch} = \frac{\text{Pitch circle diameter} \times \pi \times \cos \text{pressure angle}}{\text{Number of teeth}}.$$

$$\text{Circular pitch} = \frac{\text{Pitch circle diameter} \times \pi}{\text{Number of teeth}}$$

For this continuity of action, the line contact XY must be greater than the base pitch and the more teeth in action at one time the less the stress in the gear teeth; this is graphically represented by the ratio of the length of line XY to the base pitch.

Fig. 102 shows the " corrected " profile, in which full advantage has been taken of the tooth cut to standard depth, so that no part of the wheel tooth works with any part of the pinion tooth other than where the profile is true involute, which now extends to the base-circle diameter. With the elimination of undercutting the strength has also been considerably increased. In order to analyse the type of contact at various positions on the tooth flank, divide the working face of one tooth into a number of equal divisions and transfer to the mating tooth the portions with which each works.

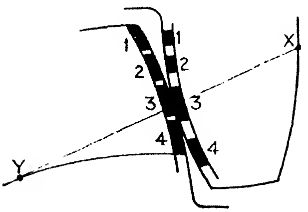



Fig. 103

Reference to Fig. 103 shows that at the pitch circle only are the divisions equal and that only rolling therefore takes place at this point. At the tip of the wheel tooth portion 1 exceeds that of point 1 on the pinion tooth, illustrating that sliding takes place at the lower part of the tooth flanks. Obviously wear is greatest at this point. Compromise will be necessary to obtain the desired amount of correction to suit individual requirements.

Correction coefficients for spur wheel and pinion and helical wheel and pinion, that is where the addendum is made equal to m (1 + K), m = module of cutter, K is correction coefficient, t = number of teeth in pinion and T = number of teeth in wheel, are given in Table 8.

Before passing from tooth considerations, and whilst appreciating that the reduction of gear noise is of primary importance, it would be well to analyse the type of noise, its category and the probable cause.

*Table* 8

| Gear | Number of Teeth | Spiral Angle | Virtual Number of Teeth | Correction Factor |
|---|---|---|---|---|
| Spur Pinion | t | 0 | t | $K = 0{\cdot}4\left(1 - \frac{t}{T}\right)$ |
| Spur Wheel | T | 0 | T | $K = -0{\cdot}4\left(1 - \frac{t}{T}\right)$ |
| Helical Pinion | t | $\alpha$ | $t\sec^3 \alpha$ | $K = 0{\cdot}4\left(1 - \frac{t}{T}\right)$ |
| Helical Wheel | T | $\alpha$ | $T\sec^3 \alpha$ | $K = -0{\cdot}4\left(1 - \frac{t}{T}\right)$ |

Gear noises generally fall within three categories:

(1) A ring or high-pitched whine.

(2) A low-frequency growl.

(3) Those emitting an irregular " hammer."

It is believed that " bounce " is primarily responsible for the majority of such noises and it is apparent under varying circumstances; by " bounce " is meant a rebound of the teeth in mesh. It is accepted that the relative velocity and angular position of two wheels in mesh will be correct when contact takes place at the pitch line, but that any profile inaccuracy produces a variation in angular position equal to the sum of such errors at the point of contact of the teeth. It follows that irregular angular velocity occurs between the driven and driving gear.

Rotational speed and tooth load are controlling factors in the magnitude of the produced effect. Immediately the condition is passed, wherein the torque transmitted to the driver produces deceleration coincident with that produced by profile inaccuracies, separation of teeth occurs, and contact is only restored through impact. The resulting rebound causes the hammering noise in category 3. This, of course, depends somewhat on the allowable backlash, and such a pair of gears can quite probably run quietly up to the speed at which separation occurs.

Even when the growling noise appears separation may be in evidence, but noise in this case may be due to local pitch errors, when the frequency of the noise would be equal to the number of revolutions of the wheel. The difference in type of noise is explained by the fact that the bounce may be damped out entirely before there is further impact. Eccentricity of bore could produce this noise by causing a constant repetition of relative pitch and profile error.

However, on the other hand, should the gear be running in constant mesh, under no load or light conditions, bouncing might occur and a prolonged rattle with no definite period being the result.

One of the main causes of " whine " is gear web or nave weakness, or a vibration of web brought into action by the general finish of the gears and rough spots on the teeth. The presence of lubricant between tooth faces, of course, damps out the natural vibration to some extent, and thus contributes to a quieter running gear.

*Internal Gears :* Fundamentally, external-gear and internal-gear systems differ in one respect only, that of base circles, in which those for the external gears lie on opposite sides of the path of tooth contact, whereas in the case of internal gears the base circles for both mating wheels lie on the same side of this contact line. It will be appreciated that the internal gears have a greater length of pressure line, and consequently the arc and duration of contact is longer, whilst the amount of overlap between meshing teeth is increased. Moreover, the internal-gear tooth possesses a difference in *tooth curvature* between contacting surfaces which results in greater bearing area. This is due to elastic distortion across the tooth face when under load.

*The sliding velocity* of the internal gear is also less than that for a similar external pair. In view of these qualities, it will be seen that a greater amount of correction is possible with the internal-gear tooth, which, whilst strengthening both teeth, permits contact to take place where the radius of tooth curvature is a minimum.

One of the points on the debit side of internal gears is that of interference which occurs at the tip of the pinion whilst passing through the internal addendum circle of the wheel. It is not advisable in ordinary application to mesh two internal gears of standard tooth form if the difference in tooth numbers is less than twelve.

In special circumstances, however, the difference may be reduced to even one tooth, but this demands a degree of correction which removes the whole of the arc of contact for some distance beyond the pitch point.

## INTERNAL TEETH

Before leaving the design of the gear teeth, a word should be said concerning the internal teeth of the gears used for dog or clutch purposes as exemplified in the constant-mesh pinion and elsewhere. It is seldom found that these teeth are generated as an internal-gear profile; more often the method employed is that of drilling and cleaning with a cutter having the same number of teeth as the clutch.

In common with other internal gears, the bore of the internal gear must be at least equal to the base-circle diameter, whilst without affecting the strength of the gear it may be opened out to the pitch-circle diameter. The strength of such an internal dog or clutch depends primarily upon the strength of the teeth in shear; thus where

F = load at pitch line (lb)
r = radius of dog (in)
W = width of the internal gear
f = safe shear stress ($lb/in^2$)

$$F = \pi r W f.$$

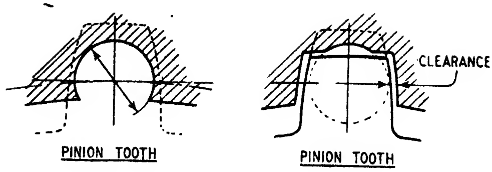



**Fig. 104—Internal dog teeth**

The diameter of drill hole is fairly standardised for various tooth forms, for instance with a 20° stub or 20° full-depth tooth, the following table may be adopted, whilst the minimum width of working tooth has been found by experience as in Table 9.

Lubrication of the gears is a most important feature, and should be studied carefully. The main characteristics of the chosen lubricant should be:

(a) Must be capable of thorough distributive flow through all ball bearings and small holes or passages, and must have no corroding effect on parts with which contact is made.
(b) Should be of such character and body as to minimise power loss through churning, and should offer minimum resistance to gear-change mechanism.
(c) Should be capable of exerting a washing action on gear teeth, etc., and possess properties for maximum heat absorption and dissipation.
(d) Must have a consistency such as to prevent leakage under normal conditions.

The effect of lubrication upon the output efficiency depends upon several factors. It would appear that excessive quantity has considerable effect, as also does that of an oil having too high a viscosity. Concerning the quantity desirable, it should be appreciated that the greater power loss occurs (due to excessive amount) in the direct-drive position. It occurs, however, in all gears and increases until the gears are completely covered. The loss is doubtless due to cavitation or oil churning, in other words reduced equivalent wheel immersion. The passages so cut by the wheel are filled either with air or lubricant at a higher temperature

*Table* 9

| 20 stub or 20 full-depth tooth | | |
|---|---|---|
| Diametral Pitch | Drill Diameter | Width of Tooth |
| 5 DP | $\frac{9}{32}$ | $\frac{9}{16}$ |
| 6 DP | $\frac{15}{64}$ | $\frac{7}{16}$ |
| 7 DP | $\frac{13}{64}$ | $\frac{3}{8}$ |
| 8 DP | $\frac{11}{64}$ | $\frac{5}{16}$ |
| 9 DP | $\frac{5}{32}$ | $\frac{9}{32}$ |
| 10 DP | $\frac{9}{64}$ | $\frac{1}{4}$ |

than the surrounding bulk. Adequate lubrication of the indirect gears is given by an oil level which immerses the teeth of one of each pair of gears, generally about one-fifth to two-fifths of the box volume.

The fall in efficiency on direct drive is considerable, between quantities of lubricant filling the box one-fifth and three-fifths full, as test figures have indicated a fall of from 97·5 to 90·5 per cent, whilst under the same conditions third gear losses are from 97.2 to 96 per cent.

The effects of viscosity are also important as, for similar input speeds, a drop in efficiency for both gears, throughout a viscosity range varying from 200 to 1,000 Redwood seconds, has been observed as 95 to 93½ per cent for direct drive and third gears falls from 96·6 to 95·5 per cent.

The relation between efficiency, input speed and viscosity is shown in Fig. 105, from which it will be seen that, for normal oil filling at two-fifths full, direct drive is the gear affected most. The low-viscosity lubricant shows decreased loss of power, most probably on account of reduced cavitation. Tooth friction varies only slightly with the viscosity of the oil and is unaffected by change in speed if torque is constant, but such friction varies with the load transmitted since it is dependent upon the contact pressure between the teeth. It will therefore be appreciated that a low-viscosity lubricant is desirable, with the filler plug arranged so that the tips of the gears only are immersed in oil. A practice adopted during initial running-in is to use temporarily an excess pressure lubricant which has the effect of tooth polishing and in consequence the possibility of film breakdown is minimised when the reversion to standard lubricant is made. However, on the larger and heavier gear boxes there are considerations other than gear teeth lubrication, as exemplified in the oil requirements of the ball bearings, the cones of a synchromesh mechanism, and the lubrication through the shaft centre of remote gears on sliding splines. In such cases it is desirable to fit a small auxiliary oil pump, preferably driven from the layshaft end, when all these and other points receive a positive and direct oil supply.

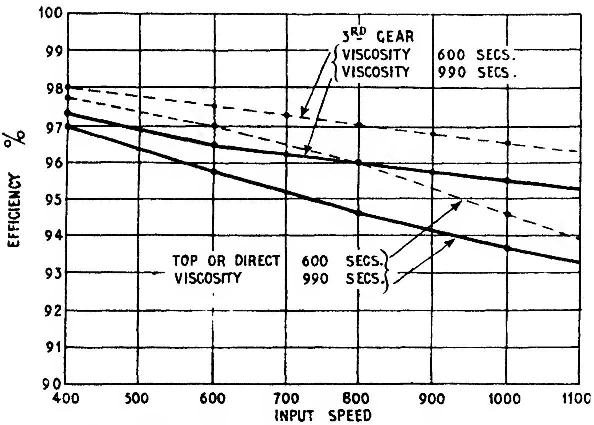



Fig. 105—Effect of lubricant on gear efficiency

## GEAR SHAFTS

*The gear shafts* are subject to stresses from combined torsion and bending moments and in extracting the shaft diameter from the formula $T_1 = R + \sqrt{R^2 + T^2}$, where $T$ = torsional moment and $R$ = bending moment, due consideration should be given to the effect of splines upon the strength of the shaft.

There is a stress concentration at the base corner of the splines and the consequent reduction in effective diameter, combined with the lowering of the fatigue-stress limit, which is due to the continuous change of section and the broken periphery of the shaft, has been the subject of much study by photo-elasticity methods. It is thereby established that the diameter upon which to base calculation is one smaller than the base of splines. Further, there is the effect of torque on the spline and its transmitted effect upon the spline base, which renders it still more desirable to assume a decreased diameter.

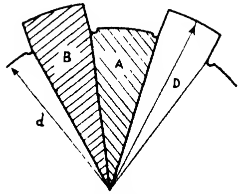


Fig. 106

The elastic limit for a splined shaft is less than that of a plain shaft of diameter equal to the base diameter of splines, whilst shear strength is reduced, depending upon the number of splines, by 5 to $7\frac{1}{2}$ per cent. When such a shaft is under torsion the sectors A and B become helices of which the lengths differ, although both are functions of the diameters A and B respectively. Measured round the helix, B is longer than A, the difference being $\pi dB - \pi dA$, and thus a shearing force is set up along the radial lines.

Continuous reversals of stress eventually cause the shaft to fracture, the fractures forming perfect sectors. Obviously the period in which such fractures occur depends upon the difference between the respective diameters D and d, and the greater the difference the earlier the fracture. It is for this reason that for an increased number of smaller splines the shaft is stronger than with a small number of large splines (see Fig. 106).

A reliable practice is to use a base diameter plus 15 per cent increase over that found from the expression $F = T_1/Z$, when it will be found unnecessary to make further additions for correction due to shaft deflection. Many differing methods of mounting the layshaft gears are available and, providing there is rigidity in the assembly, there is little more to add, except in the type which employs a rigid or fixed shaft and rotating gears.

In such a design the gear wheels are often formed in " cluster " and this arrangement is somewhat open to debate. Apart from any production difficulties the problem of gear noise again arises. Tooth inaccuracies will produce noise on one gear which must inevitably be transmitted through the layshaft train and become operative on the other gears. The effect cannot be accurately computed, but it is highly probable that any whine or ring may eventually become resonant throughout the whole system. It is possible that a break is desirable in the continuity of the gear bosses as it has been found in many cases that it splits up the medium through which sound may be transmitted.

Such calculations as are required for the determination of bearing loads and shaft deflections are straightforward and need no explanation. They are, however, set out under their appropriate heading in the section devoted to formulae.

*The gearbox casing* contributes much to the general success or otherwise of the design. Rigidity is its first essential, as it is subjected to torsion from the reaction which it transmits to its point of attachment either to engine or frame. The value of this load is equal to

$T\,(Er - 1)$ where $T$ = engine torque
$r$ = reduction ratio
$E$ = gearbox efficiency

and the highest value is therefore in reverse gear. Should the casing become distorted under load, shaft misalignment will have considerable effect upon the meshing of the gears.

It is worthy of note that the material surrounding the bearing housings is subject to tensile stress in many directions, and the available material should therefore be apportioned accordingly. Precautions must be taken to avoid resonance, and it would appear desirable to break up any flat surfaces by the introduction of suitable ribs.

## PLANETARY TRANSMISSIONS

Many of the problems which arise with the two-shaft gearbox are eliminated by the *epicyclic or planetary-type arrangement of gears.* There are two basic types of such transmissions, that consisting of spur gears entirely and that which employs internal gears. The latter type is illustrated by Fig. 107, the sun pinion A or the driving member being attached to the input shaft of the transmission. The planetary pinions C, of which there may be two, three, or more, mesh with this gear, and also with the internal gear B, and are mounted on a planet carrier, concentric with the input shaft. If ring B be rendered stationary by mechanical means and pinion A is rotating clockwise, the pinions C will roll on the internal gear in clockwise direction, whilst rotating in an anti-clockwise direction about their own axes. The planet carrier D then possesses the same directional rotation (clockwise) as the driving member, but at a reduced speed. If it is desired to increase this speed, the output shaft is locked to the driving shaft or gear A. This forms a direct drive.

*The calculation of speed ratios,* which is the ratio between the number of turns made by the arm D and the number of turns made by the sun wheel, is similar to that for ordinary gear trains. If the pinions make one complete revolution about the driving-shaft

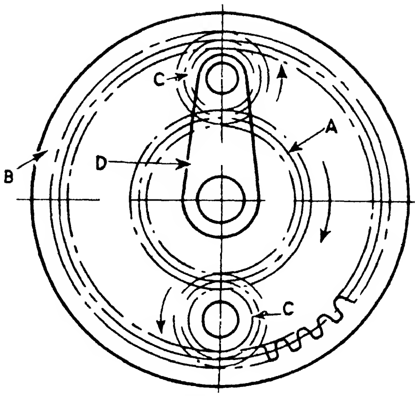


Fig. 107

centre, they will also be revolved about their own axes to the extent of $\frac{B}{C}$. By calculating the motion of the sun wheel A which is required to produce each of these planetary wheel motions and adding them together, we obtain the number of revolutions of A to provide one revolution of carrier D, which represents the gear ratio for that particular train.

The first motion of the planetary gears, that around their own axes, is expressed as $\frac{b}{c} \times \frac{c}{a} = \frac{b}{a}$. For the second motion, that of one complete revolution in the same direction as that necessary to produce the first motion of the pinions, the sum of the two is

$$\frac{b}{a} + 1 = \frac{b+a}{a}.$$

This is the reduction ratio of the gear set.

Now if it is presumed that pinion A is held against rotation and load taken off carrier D, the power being applied to ring B, the carrier will revolve in the same direction as B. If also the planetary gears with their carrier make one complete revolution, then by rolling around the sun wheel the pinions turn on their own axis $\frac{a}{c}$. To produce these two motions the angular motion of the internal gear for one revolution of the carrier D is

$$\frac{a}{b} + 1 = \frac{b+a}{b}.$$

This assembly comprises three members, the sun gear, the ring gear, and the carrier, each one of which can be held against rotation, so that six combinations are possible as power can be transmitted between the two remaining members. Two of these combinations are reducing gears, namely sun gear to planet carrier and ring gear to planet carrier, and two are step-up gears, namely planet gear to sun gear and planet carrier to ring gear, whilst the remaining two combinations are reverse gears, one step down and one step up.

*Compound Planetary-gear Set:* Fig. 108 illustrates a mainshaft or driving shaft upon which two sun wheels are attached rigidly, with two ring gears and two planet carriers to form a compound planetary

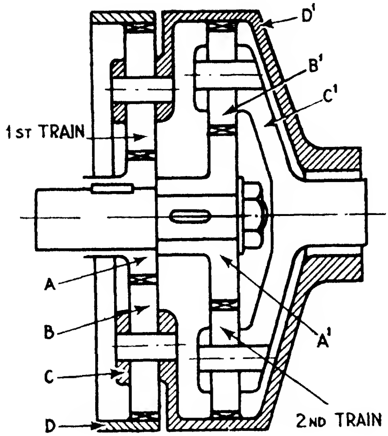



Fig. 108

set. The reduction ratio from A to C and thence to $D^1$ is $\frac{d+a}{a}$ and when the ring gear of the first train D is restrained from motion, the gear becomes a low forward gear. In the second train $A^1$, $B^1$, $D^1$, $A^1$ and $D^1$ are in motion, and consequently transmit motion to carrier $C^1$, the total motion being the sum of those which would be conveyed to it if ring gear $D^1$ were stationary and $A^1$ described $\frac{d+a}{a}$ turns, also if sun gear $A^1$ were held stationary whilst ring gear $D^1$ made one turn.

The motion imparted to $C^1$ by sun wheel $A^1$ is

$$\frac{\frac{d+a}{a}}{\frac{d^1+a^1}{a^1}} = \frac{a^1(d+a)}{a(d^1+a^1)}.$$

As the reduction from $D^1$ to $C^1$ is $\frac{d^1+a^1}{d^1}$ and $D^1$ makes one revolution the motion imparted to $C^1$ by ring gear $D^1$ is

$$\frac{1}{\frac{d^1+a^1}{d^1}} = \frac{d^1}{(d^1+a^1)}.$$

Total motion equals the sum of these two expressions

$$= \frac{a^1(d+a)+ad^1}{a(d^1+a^1)}$$

and if A revolves $\frac{(d+a)}{a}$ times, then the overall ratio (that is, is between A and $C^1$) is $\frac{(d+a)(d^1+a^1)}{a^1(d+a)+ad^1}$

*Four-speed Compound Set :* With this combination low forward speed is obtained by holding ring gear $D_2$, in which case torque is transmitted by the last train of gears, whilst the gear ratio is $r_2 = \frac{(d_2+a_2)}{a_2}$. Release of ring gear $D_2$ and restraint of $D_1$ gives second speed, torque being obtained through the last *two* trains of gears, from which it has previously been seen the gear ratio is

$$r_1 = \frac{(d^1 + a^1)\ (d_2 + a_2)}{a_2\,(d^1 + a^1) + a^1 d_2}.$$

*For third speed* sun wheel A is held and ring gear $D_1$ released, so that all three planetary trains are in action. The ratio in this case is therefore

$$\frac{(a_2 + d_2)\ [(a_1 + d_1)\ (a + d) - dd_1]}{d_2 a_1\ (a + d)}.$$

*Fourth speed*, which is direct, is obtained by locking the assembly together, that is by release of drum brakes and bringing into operation the friction clutch (see Fig. 109).

There are several small points which require attention concerning the geometry of " tooth setting ". If several planetary pinions are used in an internal-type ring gear, the relationship between the number of teeth on the driving pinion and the planetary pinions must be definite, otherwise it will not be possible to assemble the train. According to the number of trains of gears, the number of teeth a, a — a, a + 1, must be divisible by that number. Hence if three trains are used, the divisor will be 3. Assuming that a — 1 is divisible by 3 then

$$a = 3x + 1$$
$$c = a + 2b = 3x + 2b + 1$$

$$\text{and } \frac{a}{3} = x + \frac{1}{3} \text{ pitch}$$

$$\frac{c}{3} = x + \frac{2}{3} \text{ pitch.}$$

If the axes of each of the three pinions are set at 120° one to the other, and the planetaries have an equal number of teeth, then two of their tooth centres will be opposite, and a sun-pinion tooth centre will be one-third circular pitch from the line connecting the axis of the right-hand pinion and the sun pinion. If this is so then obviously a tooth centre of the ring

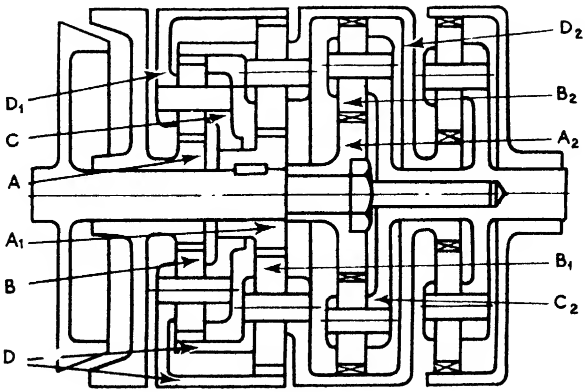


Fig. 109

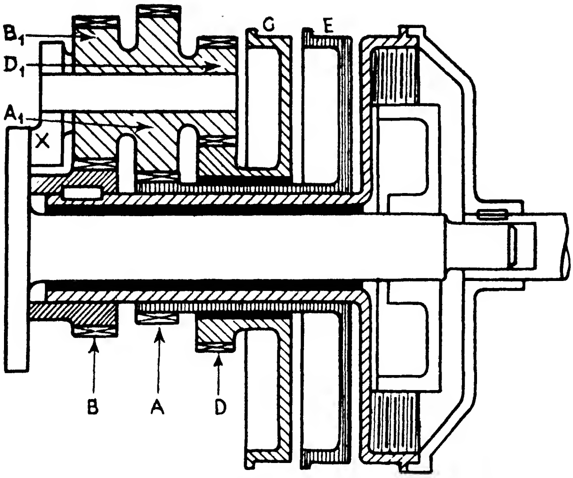



Fig. 110—All-spur type of planetary-gear set as used in the Ford Model T

gear will be one-third pitch beyond this axis line produced. If the numbers of teeth are odd, then a space and not a tooth centre is opposite another tooth centre line. Therefore the number of teeth in the planetary gears plus one should be such that they are divisible by 3, and in the case of the two planetary gear sets both driving gear and planetaries may have either an even or odd number of teeth.

If there are three trains then, if a is divisible by 3, b must also be divisible by 3 and $a - 1$, $b + 1$, $a + 1$, $b - 1$, must all be capable of division by that number. With four planetaries, both a and b may either be odd or even, but both must bear the same sign.

The Ford Model T transmission is a typical example of the all-spur type of planetary-gear set, which diagrammatically is set out in Fig. 110. In principle it comprises three sets of independent gear trains and low forward speed is obtained by holding the planet carrier against motion when power is exerted by gear A through $A_1$ and $D_1$ to gear D. The reduction is, of course, $\frac{a_1 d}{a d_1}$.

For high forward speed the drive is direct through the clutch. For reverse motion, the gear B is held against rotation. If the carrier rotates anti-clockwise $B_1$ rolls on gear B, and the gear cluster $A_1$ $B_1$ $D_1$ revolves anti-clockwise about its own axis and makes $\frac{b}{b_1}$ revolutions —in which case A makes $\frac{ba_1}{b_1 a}$ revolution, but in a clockwise direction. This, when related to the motion of the carrier, gives a total motion for A as

$$\frac{a_1 b}{a b_1} - 1 \text{ clockwise.}$$

The motion of the planetary pinions cause gear D to make $\frac{bd_1}{b_1d}$ clockwise revs and, again related to the motion of the carrier, the motion of D

$$= 1 - \frac{bd_1}{b_1d} \text{ anti-clockwise revs.}$$

Should $d_1 < b_1$, D will revolve in the reverse direction and the expression has a positive value. To find the reduction ratio use the expression

$$\frac{d\ (a_1b - ab_1).}{a\ (b_1d - bd_1)}$$

It is understood that in any spur gears the sum of the tooth numbers of mating gears is the same. If this sum is noted as x then

$$x - a = a_1$$
$$x - b = b_1$$
$$x - d = d_1$$

which if substituted in the foregoing expressions, the reduction may be stated as $\frac{d\ (b - a).}{a\ (d - b)}$

This principle may be applied to the Ford arrangement. The application of power is somewhat different as it is applied through the planet carrier X and gear B becomes the driven number. The principle enumerated however, is applicable in its appropriate stages.

*In top gear* B is connected to the driven shaft through the clutch, both brake drums being free, and the gear becomes an ordinary train, since the planet pinion cluster $B_1$ $A_1$ $D_1$ does not rotate, but revolves solid with the whole gear. B is running at engine speed. Ratio 1 to 1.

*In second gear* brake drum E is fixed and the train is $B_1$, B, $A_1$, A. It is convenient to tabulate wheel speeds for various conditions of fixed members. X is fixed and $B_1$ and $A_1$ will have one positive revolution (see Table 10).

The speed ratio for second gear is therefore

$$\frac{\frac{a_1}{a} - 1}{\frac{a_1}{a}} \text{ to } 1.$$

*Reverse gear :* In this case the reverse brake drum G is fixed, and E is free, and the train is X, $B_1$, B, $D_1$, D. See Table 11.

*Table* 10

| Fixed Member X Revolutions of | | Fixed Member A Revolutions of | | |
|---|---|---|---|---|
| X | 0 | X | $\frac{a_1}{a}$ | Driver |
| B A | + 1 | B A | $1 + \frac{a_1}{a}$ | |
| B | − 1 | B | $\frac{a_1}{a} - 1$ | Follower |
| A | $-\frac{a_1}{a}$ | A | 0 | Fixed Wheel |

*Table* 11

| Fixed Member X Revolutions of | | Fixed Member D Revolutions of | | |
|---|---|---|---|---|
| X | 0 | X | $\frac{d_1}{d}$ | Driver |
| B D | + 1 | B D | $1 + \frac{d_1}{d}$ | |
| B | − 1 | B | $-1 + \frac{d_1}{d}$ | Follower |
| D | $-\frac{d_1}{d}$ | D | 0 | Fixed Wheel |

The speed ratio for reverse gear is therefore

$$\frac{-1 + \frac{d_1}{d}}{\frac{d_1}{d}} \text{ to } 1.$$

There are many proprietary makes of epicyclic gear box in which the necessity for a separate clutch is eliminated, since by virtue of their design, if the ultimate fixed member be made free and gradually brought to rest, the driven member is gradually speeded up and the clutch action through which engine and transmission is coupled is obtained in the epicyclic gearing itself.

## ACTION AND REACTION

The principle that force or action and reaction are equal and opposite applies not only to loads and forces, but to moments and torques. Therefore if the transmission-output shaft develops a torque in one direction, the power unit tends to rotate in the opposite direction, except of course if the transmission is in direct drive, as in that case both output torques are similar in magnitude and direction. The torque is then taken by the engine case. If the output torque is greater than that at input, due to insertion of a gearbox or similar mechanism, a reaction member in the transmission is essential. The gear-set casing forms a convenient means of absorbing this reaction. The conventional automobile transmission usually comprises a number of gear trains, each of which effect torque conversion in different ratios. There are, of course, many types of mechanisms which are used to make this conversion and, in turn, some of the more widely accepted will be reviewed.

First, the *gear type of reduction*, both " clash " and synchromesh, involute spur teeth or helical gears: The main function of the gears is to transmit motion from one shaft to another, with uniform velocity, with the minimum of noise, and with as little shock loading as possible. The practice of synchronising the moving parts to be engaged considerably relieves the last two points, whilst the use of constant-mesh helical gears assists the first condition. With both the clash and constant-mesh types, the methods of bearing-load computations are somewhat similar, but there are one or two conditions to be satisfied in the synchronising type which do not apply to the straightforward sliding gear.

Synchromesh designs may be separated generally into two types, (a) the inertia lock and (b) the constant load. Cone clutches are the mediums through which the synchronising is carried out in both types and the main difference is that it is not possible to " clash " the gears by too much pressure on the control mechanism with the inertia-lock designs whereas this is not so with the latter type. Very brief explanations of the two systems will make this clear.

*Inertia-lock Synchromesh* (Fig. 111): Any endwise movement of the synchronising cone clutch brings into operation the inertia lock, which prevents positive gear engagement until there is no relative slip between the two cones. When such a condition has been reached the two gears will mesh. It will be appreciated that the time taken to synchronise is inversely proportional to the load applied, since the greater the force exerted on the hand lever the greater the force between the two cones and, in consequence, the greater the inertia-lock load. Due to the difference in speeds at the moment of change between the two cones, that on the gear B and that on the drum A, both cone and drum rotate slightly until the projections F on the drum make contact with the mainshaft spline sides.

The engaging dog C is, however, exerting pressure on the drum through the medium of the chamfered edges and it is impossible to

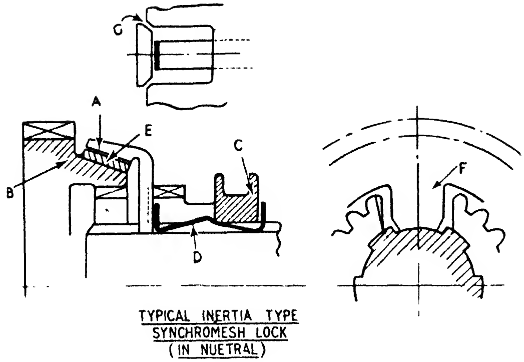



Fig. 111

move the engaging dog further during the period in which the torque on the drum is greater than that caused through the chamfered faces. This drum torque decreases, however, as the speeds approach synchronism and, when it is just less than the torque between the two chamfered . faces, the drum moves forward and permits the dog to follow through to engagement.

*Constant-load Synchromesh* (Fig. 112): In this type the gears can still be " crashed " or the gears be made to mesh before synchronisation in speed takes place by undue heavy load applied to the gear-control lever, if such a load overcomes the pressure required to overcome the springs which in turn control the ball loading, whether the mating parts are synchronised in speed or not. Reference to the rotation in Fig. 112 will show that pressure from the change-speed lever is applied to the outer ring A. This is transferred to inner member C, through the groove and the spring loaded balls, bringing the cones in contact. The speeds are thus synchronised between the shaft and engaging dog.

Additional pressure on the change-speed lever depresses the ball springs, allowing the outer member to slide and positively engage the gear. The cone angles are of considerable importance, as upon the angle depends the loads required for synchronising. The angles are usually about 6° to $7\frac{1}{2}$° as this figure also permits sufficient longitudinal movement to maintain the cone face clearance which approximates 0·005 in.

It is essential if good syn-

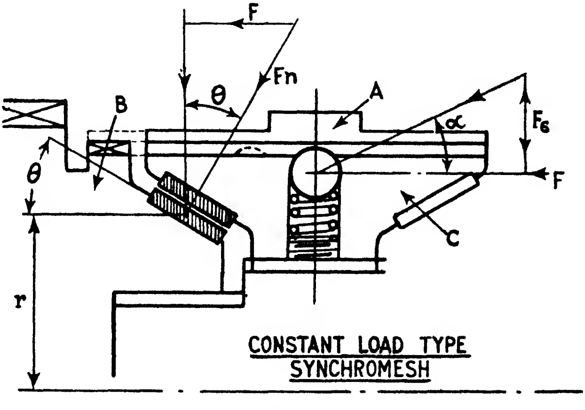



Fig. 112

chronisation is to be maintained that lubricant should be dispersed immediately the cone clutches engage ; thus, in consequence, the design of oilways requires careful consideration. The actual diameter and angle control the speed with which synchronisation takes place and the clutch must perform the function of changing the inertia of the moving parts from their running speeds to that of the new gear velocity.

The fundamental formulæ required in estimating the torque required for synchronism are of course force = mass × acceleration, or $\frac{M}{g} \times A$.

$$\text{Torque} = \frac{M}{g} \times K^2 \times \frac{2\pi \times n}{12t},$$ where K = radius of gyration (in)

M = weight (lb)
g = gravity acceleration (32·2 ft/sec/sec)
n = r/sec
t = time taken to synchronise in sec

since mass × radius of gyration² = I

$$\text{Torque} = \frac{I\pi n}{6t}$$

The maximum load applied to the cone when a change of gear from say top to third is made, occurs when the third gear road speed is such that the engine revs are maximum for that gear. Assuming the road speed to be constant, the difference in speeds of the affected parts in the gear box should be tabulated in r/sec. If this condition is maintained the speed of the mainshaft, and other parts slidably splined to it, will remain constant, but the layshaft and gears in mesh with those on the mainshaft must be speeded up together with the clutch disc and constant-mesh pinion. The layshaft and gears, clutch disc and constant-mesh pinion speeds are, of course, mainshaft speed and third-speed ratio.

The moments of inertia for all rotating parts should be ascertained and the torque required to synchronise the speeds will thus be

$$T = \frac{\pi}{6t}(In + I_1 n_1 + I_2 n_2, \text{ etc.})$$

From this torque, which is applied by the cone clutch, the normal cone load becomes $F_N = \frac{T}{r\mu}$ and the axial load $F = \frac{T \sin \theta}{r\mu}$ whilst the spring load $F_S = \frac{F \tan \alpha}{N}$ where N = number of balls.

Fig. 113 indicates the advantages of the synchromesh-type box over the conventional type with regard to gear change on gradients. The example illustrates results obtained from the test data with a standard vehicle equipped with a diesel engine and developing 110 b.h.p. at 2,000 r/min: rear-axle ratio 6·43, 9·75 diameter

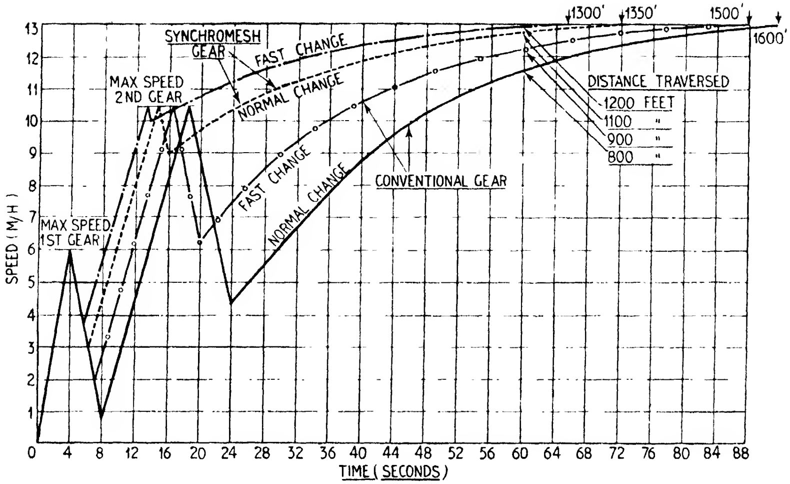



**Fig. 113—Performance advantage of synchromesh gears**

tyres, gross weight approximately 10 tons. The test gradient was 966 yd long, approximately 7 per cent gradient, and tests were taken from a standing start.

The black unbroken line represents a normal change without gears clashing. The broken line and rings concur quick change, but gears clash. The fast synchromesh change is as made with the accelerator pedal depressed throughout the gear change, an occurrence which, due to consequent heavy loading on the main clutch, is not recommended for normal use.

Comparing the graph figures, it will be seen that the vehicle with synchromesh gear was faster since in 60 seconds a distance of 1,100 to 1,200 ft had been covered, the distance for conventional-type box being 800 to 900 ft.

Gear change from first gear to second gear slows the vehicle down from 6 m/h to 1½ to 3–4 m/h, an improvement being made by the synchromesh to a reduction from 6 m/h to 3½–3 m/h; 4½ to 5½ secs are required to regain 6 m/h, as against the lower curves 8 to 10 secs. It will be noted that the change from second to third slows the vehicle from 10½ m/h to 10–9 m/h against the competitive types 6–4½ m/h.

In order to attain the speed again this box requires 22 to 31 secs, whereas on the upper curves only 3½ to 10 secs are necessary. This would appear to indicate that performance advantages to be obtained by the adoption of synchromesh gears are worthy of considerable attention.

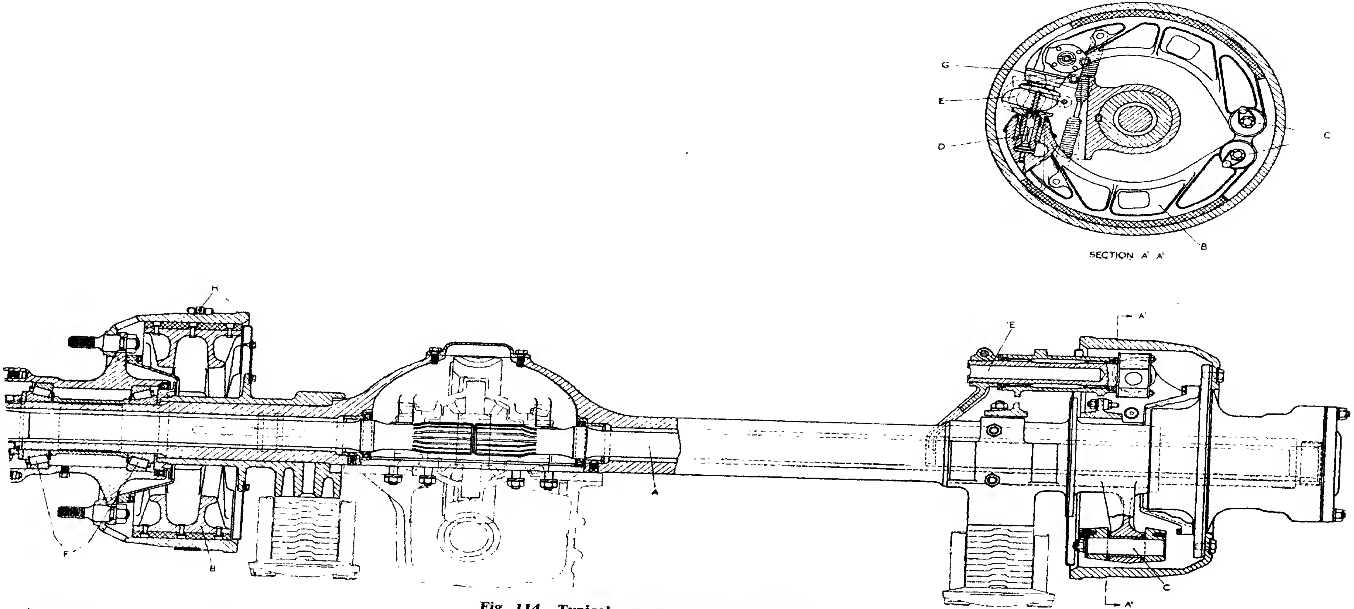



**Fig. 114—Typical passenger-vehicle rear axle**

A, splined axle shaft; B, brake shoe; C, shoe fulcrum pins; D, automatic shoe adjuster; E, shoe-operating cam; F, taper roller bearings; G, shoe pull-off springs; and H, anti-squeal band

[*To face page* 157

## CHAPTER 7

# REAR AXLES

## CASINGS

THE type of axle which incorporates the three-piece fabricated design, that is the cast centre and tubular arms, has many merits and if the design is carried through in accordance with latest practice good results should accrue. The older method of pressing the tubular arms into position required extreme accuracy in order that any key-ways, etc., should be maintained in line with each other. Moreover, the possibility of slackness developing between arms and case could never be overlooked, and the bursting stresses could never be correctly calculated.

The present method of " freezing " the tubes overcomes many of the past deficiencies and affords means whereby correct hoop stresses may be found for different ranges of shrinkage fits. Briefly, the method adopted provides for the immersion of the tube end into a liquid oxygen bath at a temperature of approximately 120° F below zero, which, for steel, gives a contraction of roughly 0·0045 in, whilst the casing is heated to a temperature of 140° F for approximately 0·0025 in expansion.

When both tube and casing are mated the interference is approximately 0·003 to 0·0045 in and the preloading is in the order of 7 tons/in$^2$. The assembly is then securely dowelled as an additional precaution against any movement. Alternative freezing agents may be used, such as methylated spirit or trichlorethyline in conjunction with solid $CO_2$ (see Fig. 115).

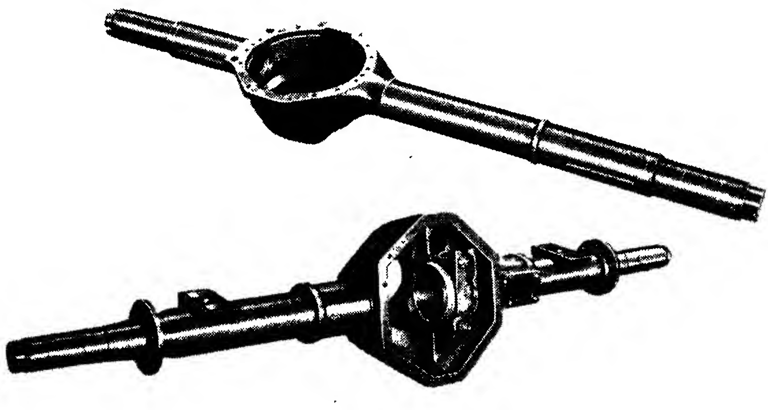


**Fig. 115—Typical rear axle casings**

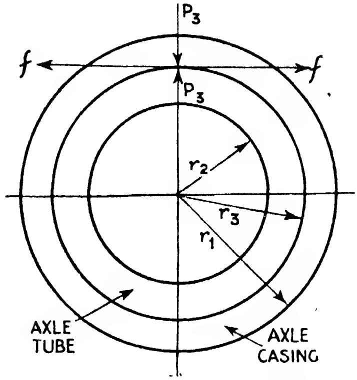



**Fig. 116—Pressures due to shrinkage**

It is interesting to analyse the hoop stresses, which may be derived from Lames basic theory, which states that:

$$p = C_1 + \frac{C_2}{r^2}$$

where $p$ = internal pressure
$r$ = internal radius
$C_1$ & $C_2$ are constants.

Where $P_3$ = initial pressure due to shrinkage at surface radius $r_3$, by derivation (see Fig. 116):

$$C_1 = \frac{P_3 r_3^2}{r_3^2 - r_2^2}$$

and $C_2 = \dfrac{-P_3 r_2^2 r_3^2}{r_3^2 - r_2^2}$

since $f$ also equals $C_1 - \left(\dfrac{C_2}{r^2}\right)$

by substitution:

$$f = \text{compressive stress in the ring or axle tube}$$
$$= P_3 \left(\frac{r_3^2}{r_3^2 - r_2^2} + \frac{r_2^2 r_3^2}{r_3^2 - r_2^2} \times \frac{1}{r^2}\right) \text{lb/in}^2$$

$$\text{and } f_1 = \text{tensile stress in the axle casing}$$
$$= -P_3 \left(\frac{r_3^2}{r_1^2 - r_3^2} + \frac{r_1^2 r_3^2}{r_1^2 - r_3^2} \times \frac{1}{r^2}\right) \text{lb/in}^2.$$

The maximum stress in the tube occurs where $r = r_2$ and again substituting we find $f_2$ = maximum stress = $\dfrac{2P_3 r_3^2}{r_3^2 - r_2^2}$

whilst maximum stress in the casing occurs at $r_3$

$$f_3 = \text{maximum stress} = -P_3 \left(\frac{r_1^2 + r_3^2}{r_1^2 - r_3^2}\right).$$

Between this range of formulae the stress can be found at any point in the section of either the arm or the casing.

$P_3$ is, of course, dependent upon the amount of interference between the two machined diameters of the tube and casing, and the relation may be expressed as:

$$\square\, d^3 = \text{interference} = \frac{4P_3 r_3^3}{E} \times \frac{r_1^2 - r_2^2}{(r_1^2 - r_3^2)(r_3^2 - r_2^2)},$$

E representing elasticity 30,000,000.

Pressure $P_3$ and stress $f$ can therefore be calculated for any desired amount of interference. A typical diagram is plotted (Fig. 117) which shows the stresses in the casing and tube for varying interferences. From these curves it will be seen that the maximum stresses occur in the tube or arm at its inner diameter and that these stresses are compressive, whilst those for the casing are tensile and are also maximum at the bore. The curves are based on a casing bore of approximately $4\frac{3}{4}$ in diameter material, $\frac{5}{8}$ in thick tube, internal bore $3\frac{1}{8}$ in diameter. Such a design also possesses the advantage that the track may be modified without the necessity for new parts. It is probably a little heavier than the one-piece casing.

The three-piece axle casing is not always manufactured in the same manner; the tubular arms may be pressed and dowelled into the cast centre. Pressing into position nearly always produces some degree of deformation in both inside and outside diameters in both parts. The extent of this condition can be calculated with accuracy providing the materials are not stressed beyond their proportional limits, whilst the assumption is made that the components are acted upon by uniformly distributed pressure acting radially both internal and external.

Where $a$ = inner radius
$b$ = outer radius
$P_1$ = inner pressure
$P_2$ = outer pressure
$R$ = radius
$E$ = modulus of elasticity
$\mu$ = Poisson's ratio
$D$ = deformation at $R$

the expression which holds good for bushes, or bearings, is (from Fig. 118)

$$D = \left(\frac{1-\mu}{E}\right)\left(\frac{a^2 P_1 - b^2 P_2}{b^2 - a^2}\right)\left(R + \frac{1+\mu}{E}\right)\left(\frac{a^2 b^2 (P_1 - P_2)}{(b^2 - a^2) R}\right).$$

Now assume two cylinders, the outer diameter of the inner cylinder

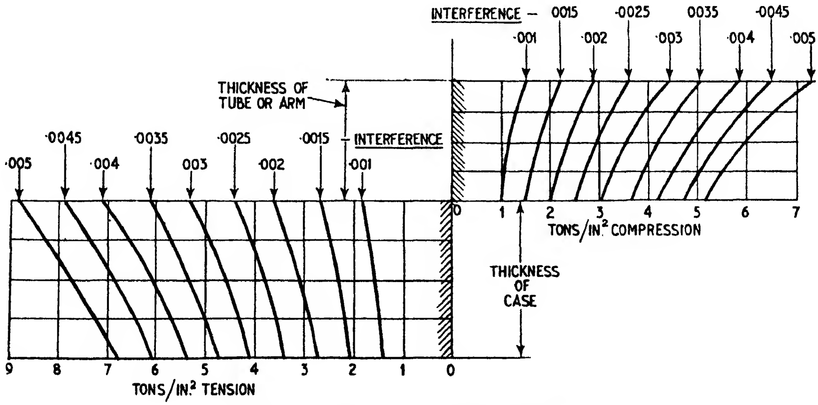



Fig. 117—Interference/stress diagram

exceeding the inner diameter of the outer cylinder by some tolerance, then with further notation:

$E_1$ = modulus of elasticity, inner cylinder
$E_2$ = modulus of elasticity, outer cylinder
$\mu_1$ = Poisson's ratio, inner cylinder
$\mu_2$ = Poisson's ratio, outer cylinder
$\delta$ = fit between inner and outer cylinder
P = pressure between cylinders
$D_1$ = increase in inner radius of outer cylinder
$D_2$ = decrease of outer radius of inner cylinder
$D_3$ = decrease of inner radius of inner cylinder
$D_4$ = increase of outer radius of outer cylinder.

Obviously the sum of the amounts of deformation of both cylinders must be equal to $\delta$.

Therefore $D_1 - D_2 = \delta$

and for the first expression for D, $D_1$ and $D_2$ are obtained.

$$D_1 = \frac{bP}{E_2}\left(\frac{b^2 + c^2}{c^2 - a^2} + \mu_2\right)$$

$$D_2 = \frac{-bP}{E_1}\left(\frac{a^2 + b^2}{b^2 - a^2} - \mu_1\right)$$

substitution in expression for $\delta$

$$\delta = \frac{bP}{E_2}\left(\frac{b^2 + c^2}{c^2 - b^2} + \mu_2\right) + \frac{bP}{E_1}\left(\frac{a^2 + b^2}{b^2 - a^2} - \mu_1\right)$$

from which P will equal:

$$P = \frac{\delta}{\frac{b}{E_2}\left(\frac{1 + (b/c)^2}{1 - (b/c)^2} + \mu_2\right) + \frac{b}{E_1}\left(\frac{1 + (a/b)^2}{1 - (a/b)^2} - \mu_1\right)}$$

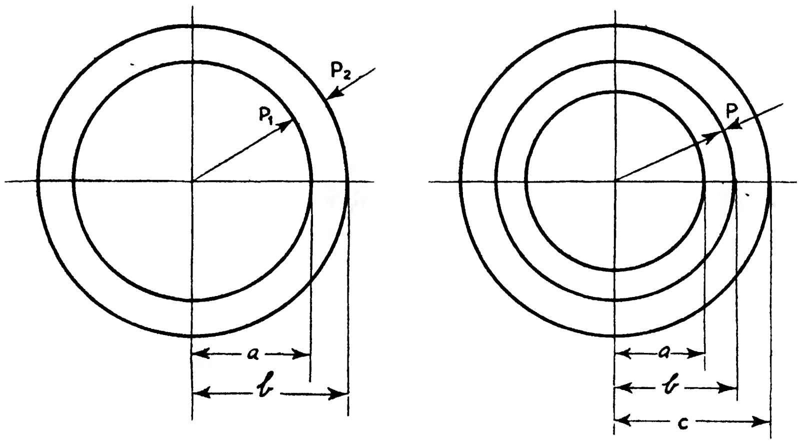



**Fig. 118—Deformation diagram**

If this value of P be used, then from expression for D, $D_3$ and $D_4$ are found to be:

$$D_3 = \frac{-2a}{[E_1(1 - (a/b^2))]} \times P$$

$$D_4 = \frac{2b\,(b/c)}{[E_2(1 - (b/c)^2)]} \times P$$

Should the inner and outer cylinders be of the same material, the expression for P, $D_3$ and $D_4$ can be simplified and, since they would be independent of elastic constants, could be combined. Therefore, combining P and $D_3$ and noting that both moduli of elasticity and both Poisson's ratios are equal:

$$D_3 = \frac{-(a/b)\,(1 - (b/c)^2)}{(1 - (a/c)^2)} \times \delta.$$

Combining P with $D_4$

$$D_4 = \frac{(b/c)\,(1 - (a/b)^2)}{(1 - (a/c)^2)} \times \delta.$$

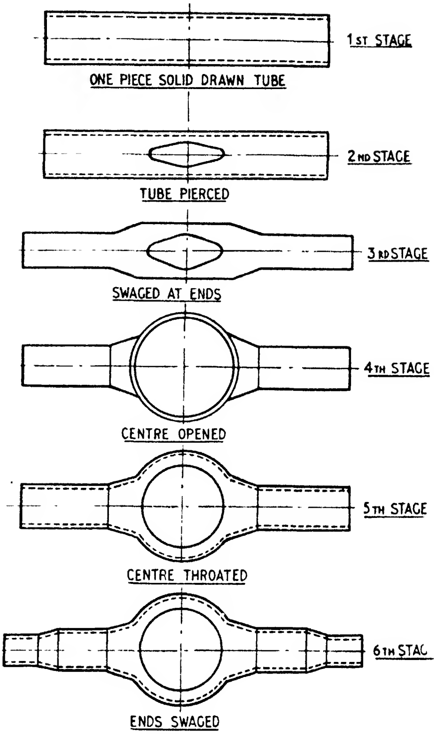



**Fig. 119—Process of axle-case formation for solid-drawn tube**

Another method of producing a lighter type of axle casing, which is now rapidly finding favour, is the one-piece forging from a single solid-drawn tube. It is claimed for this construction that the wall thickness can be controlled and therefore the maximum load-carrying capacity is attained for minimum weight. The stages through which the tube passes are shown in Fig. 119 and the finished forging is finally heat treated.

Very briefly the mechanical process is as follows : a seamless steel tube, the diameter of which is selected to suit the ultimate banjo size, is first notched and pierced to form slots disposed longitudinally and diametrically opposite. A ram is then fed up the tube at each end, forcing reducing rings over the outside diameter. The slots in the preliminary operation are used to grip and locate the tube during this process. This is a cold operation and results in a thickening of material and a reduction in diameter where the arms will eventually be formed. The tubes are then fed through an automatically-controlled furnace provided with a " walking beam " floor. After the desired diameter has been reached the slotted tube is expanded by mandrels to form the banjo centre and the edges are flanged over. The flanges are then trued and the axle subjected to the usual welding operations.

## REAR AXLE ARRANGEMENTS

There are three generally adopted hub-bearing and axle-shaft arrangements, (a) fully floating, (b) semi-floating and (c) three-quarter floating. Concerning the *fully-floating type*, the axle shaft transmits driving torque only and it is not subjected to thrust loads emanating from vehicle weight, since the weight is supported by the axle casing. Such an arrangement is shown in Fig. 120, which is a type favoured by the heavy-vehicle designer, although its production costs exceed that of the other arrangements.

Should an axle-shaft break, it may be withdrawn without dismantling the wheel or of jacking up the vehicle; moreover there is no danger of a wheel coming adrift. The two bearings employed for hub mounting share the load from each wheel, the load line usually being slightly nearer to the inner bearing. The maximum stress in the casing usually occurs at the change of section adjoining the inner bearing, its magnitude being expressed as:

$$f = \frac{\text{B.M.}}{\frac{\pi}{32}\left(D^3 - d^3\right)}$$

where B.M. = maximum bending moment (in/lb)
D = outside diameter of tube (in)
d = inside diameter of tube (in).

The bore of the bearing should therefore be such that the casing stress should not exceed this figure, which is usually arranged at approximately 10 tons/in². Axle-bearing loads emanate from two sources, (a) vehicle weight and (b) skid reaction. The maximum ground reaction is obtained when the vehicle is traversing a bend, when centrifugal force comes into effect and reduces the load on the inner wheel, increasing that on the outer wheel by a similar amount.
If W = load on rear axle
E = track of the wheels

then the reaction on the outer wheel is $R = 0{\cdot}5\,\frac{W}{2} + \left(\frac{\frac{Wv^2}{gr} \times h_1}{E}\right)$

where g = acceleration due to gravity
r = radius of bend (ft)
h = height of centre of gravity of vehicle
v = velocity of the vehicle (ft/sec).

This load is divided between the two wheel bearings in the following proportions:

Inner bearing $R \times \frac{a}{c} = R_1$

Outer bearing $R \times \frac{b}{c} = R_0$

A further load due to tractive effort at the road T is distributed:

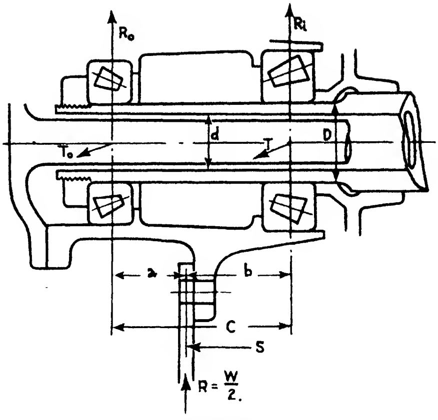



Fig. 120

Inner bearing $T \times \frac{a}{c} = T_1$

Outer bearing $T \times \frac{b}{c} = T_0$

The total radial load therefore on each bearing emanating from ground reaction is:

Inner bearing

$$\sqrt{\left(R \times \frac{a}{c}\right)^2 + \left(T \times \frac{a}{c}\right)^2}$$

Outer bearing

$$\sqrt{\left(R \times \frac{b}{c}\right)^2 + \left(T \times \frac{b}{c}\right)^2}$$

The skid reaction S is the same for each bearing, but the direction is changed and the load due to $S = S\frac{r}{c}$ is in a downward direction on the outer bearing, and upward at the inner bearing (r = running radius of the tyre). The magnitude is $\frac{RoWV^2}{Wgr}$

A check upon bearing spacing should be made with loads due to S and adjustment made to the centres if the bearing is overloaded. The axle-shaft diameter being subjected to torsion load only may be assessed from the expression

$$f = \frac{\frac{T}{2}}{\frac{\pi}{16}d^3}$$, where T = low-gear torque (that is, clutch slip torque $\times$ lowest gear ratio).

As a general rule, the *three-quarter floating axle* possesses one outer-wheel bearing and the load line is coincident with the centre line of this bearing. In consequence the axle shaft transmits torque only when the vehicle is running in a straight line. When, however, the car is rounding a bend, a bending moment is imposed on the shaft through the skid reaction.

This moment may be expressed as B.M. = 0·6Wr and the stress therefore may be found from the usual B.M. = fZ(Z in shear) (see Fig. 121).

*The axle arm* or tube is most highly stressed at the point A, from the centre line of bearing, where B.M. = WA (in/lb)

$$\text{and } f = \frac{\text{B.M.}}{\frac{\pi}{32}(D^3 - d^3)}$$

*The axle-shaft* stress at the driving end, that is the differential end,

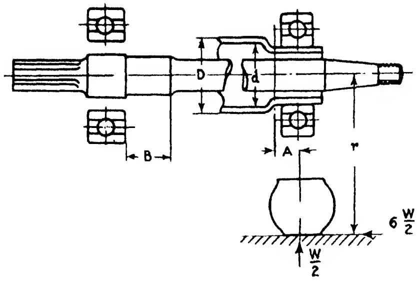


Fig. 121

may be calculated from torque considerations only and the expression is similar to that for the fully floating

$$\text{shaft } f = \frac{\frac{T}{2}}{\frac{\pi}{16}d^3}$$

and where the practice of tapering the shaft towards its inner end is carried out, for reasons of permitting a certain amount of torsional resilience which is necessary to prevent fracture when sudden torque loads are applied, the length of the reduced diameter should be at least equal to four diameters. In practice the permissible torsional deflection is in the order of $\frac{1}{2}°$ to $1°$ and if $\phi$ = angle of deflection

$$\phi = \frac{57 \cdot 3 \frac{T}{2} B}{\frac{\pi}{32} d^4 E} \quad \text{where } E = 12{,}000{,}000 \text{ lb/in}^2.$$

*Any deflection of the shaft* must be kept within the safe limits for the outer bearing for the wheel. The graphical solution is the most practical and should be used in order to obtain deflections at any point along the length of the shaft. The same deflection-diagram construction may be applied to a similar shaft of the semi-floating type.

*An axle of the semi-floating type* is generally only used on light vehicles. The axle shafts must withstand bending in addition to torsion, whilst, due to the overhang of the wheel-load line in relation to the wheel-bearing centre line, the inner bearings or differential-casing bearings are subjected to loading emanating at the wheels. If the wheel bearings are chosen to withstand straight high-speed running they will have sufficient capacity to withstand loading from cornering at lower speeds (see Fig. 122).

If $W$ = rear-axle load
$R$ = ground reaction

then $\frac{W}{2} = R$ and the loads on the bearings due to R

will be: Wheel bearing $R\left(\frac{a+b}{b}\right)$

and those due to $T = T\left(\frac{a+b}{b}\right)$

The total wheel-bearing loading can be expressed as

$$\sqrt{\left\{R\left(\frac{a+b}{b}\right)\right\}^2 + \left\{T\left(\frac{a+b}{b}\right)\right\}^2}$$

The loads on the differential bearings due to R will be:

$$R\frac{a}{b}$$

and those emanating from T $= T\dfrac{a}{b}$

To these loads, however, those from the pinion and ring gear must be added and of course the magnitude of load varies according to the type of drive, either spiral-bevel, straight-bevel, hypoid gear or worm gear.

The wheel bearings should be selected for size after the axle shaft has been stressed and a suitable diameter agreed. The procedure in setting out the graphical solution of axle-shaft strength and deflection is accurate and the stages are straightforward. An exact understanding is necessary of required conditions of the shaft to be stressed, since the maximum bending moment (which is at the centre line of wheel bearing) occurs when the vehicle is rounding a bend at tipping speed, although the B.M. which occurs when turning a bend of say 120 ft radius at 20 m/h is more valuable where shaft deflection is concerned, as supplying data for *normal* bearing performance, and since the skid reaction moment is additional to ground reaction moment on the axle shaft at the inside of the curve, the conditions are more severe for average driving.

It is therefore advised to stress for this latter condition, in which case, if

W = rear-axle weight
h = height of centre of gravity
$R_r$ = running radius of tyre
E = rear-wheel track
a = distance of wheel to bearing
r = radius of bend of road = 120 ft

then centrifugal force $= \dfrac{WV^2}{gr}$ and if V = 20 m/h (that is 29·3 ft/sec)

$= 0{\cdot}221W$

and R = (ground reaction) $= \dfrac{W}{2} - \dfrac{0{\cdot}221\,Wh}{E}$

S = (skid reaction) $= \dfrac{R}{W} \times 0{\cdot}221\ W$

(1) Bending moment therefore, which is the first value to calculate, is B.M. $= Ra + \dfrac{SR_1}{a}$

(2) Plot the B.M. diagram, and divide it into a series of figures shown shaded

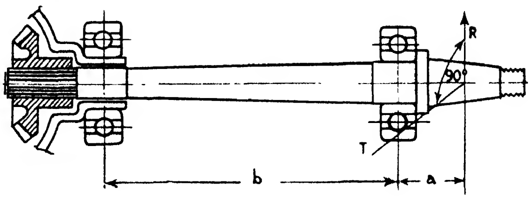



Fig. 122

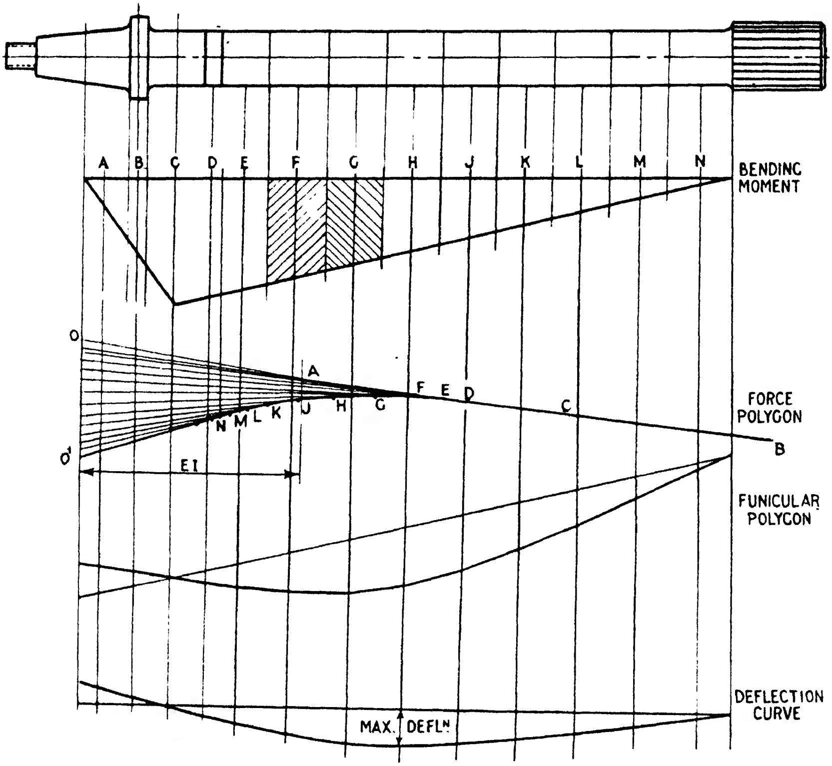



Fig. 123—Axle-shaft stresses

in Fig. 123. Take note that these divisions include any change of section in the shaft.

(3) From the point of centre of gravity of each division draw vertical lines and letter each one A, B, C, etc.

(4) Next lay out the polygon of forces. In a vertical downward direction from O, mark off segment A to scale equal to the area of division A in the bending-moment diagram. Proceed similarly with other divisions of the B.M. diagram.

(5) From the vertical scale line lay off the pole distance where
E = 30,000,000 lb for steel
and $I = \frac{\pi d^4}{64}$ where d = shaft diameter (in)
EI = flexural rigidity of division A of axle shaft.

(6) Draw OA to any convenient point A, at pole distance.
Draw from the next point down the vertical scale representing the next division B line extending until it intersects pole

distance of division B, and so on until all the divisions have been treated and the polygon of forces is complete.

(7) Now construct the funicular polygon from the diagram of forces, commencing with a line parallel to OA produced until it intersects vertical line A. Follow this procedure for each polygon of force line, in each case commencing at the previous intersection point and parallel to its corresponding division on the vertical force line. At the point of intersection of the last line through the differential support bearing, draw the closing line of the polygon to intersect the centre line of the wheel bearing.

(8) The deflection curve may now be drawn as the vertical distance of any point on the funicular polygon base line and the curve is proportional to the axle deflection. In order to obtain the *angle of deflection* at the bearing centre line, draw a tangent to the deflection curve as it passes through this position and divide it by the ratio of the two scales, that is, the vertical scale on the polygon of forces or B.M. scale, and the EI scale, or horizontal distance.

$\therefore$ if $\alpha$ = diagram angle of deflection to scale

then $\dfrac{\alpha}{\frac{EI}{BM}} = \alpha_1$, the true angle of deflection.

As previously remarked, the loads from the final drive gear must be added to the bearing loads and for different types of drive the following calculations are utilised.

*Bearing loads with spiral-bevel gears:* Rear-axle pinions generally rotate clockwise for forward vehicle motion, and they possess a left-hand spiral, that is, when viewed from the power input end. Such an arrangement is very desirable on account of the tendency in condition (1) and (4) to force the pinion out of mesh (a feature which can be overcome by the provision of an adequate thrust bearing), but in the case of conditions (2) and (3), the pinion is thrust further into mesh with the possibility of tooth wedging. As the gear teeth can be cut with either left- or right-hand spiral there are four combinations possible; the first one is the generally chosen arrangement (see Fig. 124):

(1) Left-hand spiral pinion = clockwise rotation.
(2) Right-hand spiral pinion = clockwise rotation.
(3) Left-hand spiral pinion = anti-clockwise rotation.
(4) Right-hand spiral pinion = anti-clockwise rotation.

Bearing loads will be computed for condition (1)

where $T_p$ = pinion thrust
$T_g$ = gear thrust
$F$ = tangential force
$\alpha$ = tooth pressure angle
$\beta = \dfrac{\textit{Pitch cone angle (pinion)}}{2}$
$\delta$ = spiral angle.

$$\text{Tangential force} = F = \frac{\text{Torque input (lb in)}}{\text{Mean pitch radius (in)}} = \frac{TQ}{rp_i}$$

$$rp_i = \frac{\text{Pinion pitch diameter} - \text{Tooth face} \times \sin\beta}{2}$$

$$r_g = rp_i \left(\frac{N_p}{N_g}\right) \text{ where } N = \text{number of teeth}$$

$$\beta = \tan^{-1}\left(\frac{N_p}{N_g}\right)$$

The values of $T_p$ are reversed in directions in condition (2) and (4), whilst for the four combinations the values of $T_g$ and $T_p$ are as follows:

(1) Left-hand spiral—clockwise rotation and (2) right-hand spiral—anti-clockwise rotation

$$T_p = F\left(\frac{\tan\alpha \sin\beta}{\cos\delta} + \tan\delta \cos\beta\right)$$

$$T = F\left(\frac{\tan\alpha \sin\beta}{\cos\delta} - \tan\delta \cos\beta\right)$$

(3) Right-hand spiral—clockwise rotation and (4) left-hand spiral—anti-clockwise rotation

$$T_p = F\left(\tan\delta \cos\beta - \frac{\tan\alpha \sin\beta}{\cos\delta}\right)$$

$$T_g = F\left(\tan\delta \cos\beta + \frac{\tan\alpha \sin\beta}{\cos\delta}\right)$$

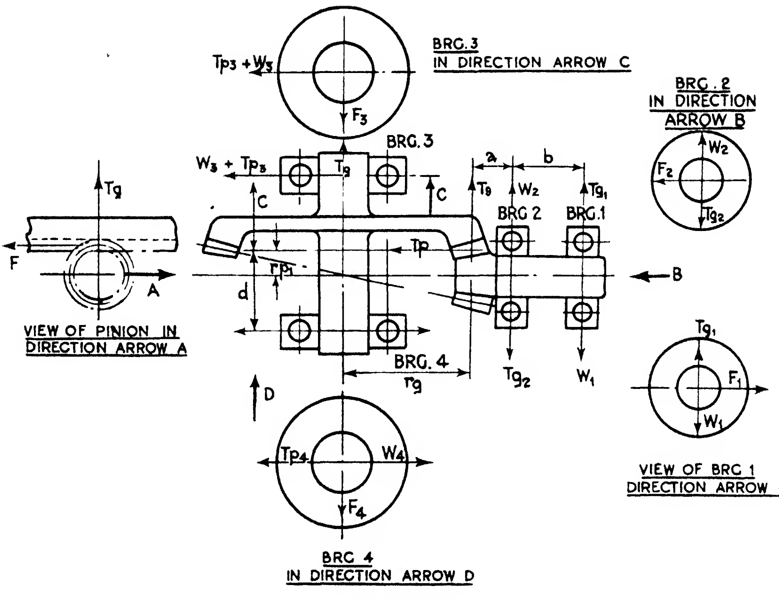



Fig. 124

## BEARING LOADS FOR SPIRAL BEVEL (Fig. 125)

| FORCE | BEARING NO. 1 | BEARING NO. 2 | BEARING NO. 3 | BEARING NO. 4 |
|---|---|---|---|---|
| F | $F \times \frac{a}{b} = F_1$ | $F \times \frac{a+b}{b} = F_2$ | $F \times \frac{d}{c+d} = F_3$ | $F \times \frac{c}{c+d} = F_4$ |
| Tg | $Tg \times \frac{a}{b} = Tg_1$ | $Tg \times \frac{a+b}{b} = Tg_2$ | $Tg \times \frac{rg}{c+d} = W_3$ | $Tg \times \frac{rg}{c+d} + W_4 = W_3$ |
| Tp | $Tp \times \frac{rp_1}{b} = W_1$ | $Tp \times \frac{rp_1}{b} = W_2 = W_1$ | $Tp \times \frac{d}{c+d} = Tp_3$ | $Tp \times \frac{c}{c+d} = Tp_4$ |
| TOTAL RADIAL LOAD | $\sqrt{F_1^2 + (Tg_1 - W_1)^2} = TR$ | $\sqrt{F_2^2 + (Tg_2 - W_2)^2}$ | $\sqrt{F_3^2 + (W_3 + Tp_3)^2} = TR_3$ | $\sqrt{F_4^2 + (W_4 - Tp_4)^2}$ |
| THRUST LOAD | Tp | — | Tg | — |
| TOTAL LOAD | $\sqrt{F_1^2 + TR_1^2}$ | $\sqrt{F_2^2 + (Tg_2 - W_2)^2}$ | $\sqrt{Tg^2 + TR_3^2}$ | $\sqrt{F_4^2 + (W_4 - Tp_4)^2}$ |

## BEARING LOADS FOR HYPOID GEARS (Fig. 126)

| FORCE | BEARING NO. 1 | BEARING NO. 2 | BEARING NO. 3 | BEARING NO. 4 |
|---|---|---|---|---|
| F | $F \times \frac{a}{b} = F_1$ | $F \times \frac{a+b}{b} = F_2$ | $F \times \frac{f}{c+f} = F_3$ | $F \times \frac{c}{c+f} = F_4$ |
| Tp | $Tp \times \frac{rpi}{b} = W_1$ | $Tp \frac{rpi}{b} = W_2 = W_1$ | $Tp \times \frac{f}{c+f} = Tp_3$ | $Tp \times \frac{c}{c+f} = Tp_4$ |
| Tg | $Tg \times \frac{a}{b} = Tg_1$ | $Tg \times \frac{a+b}{b} = Tg_2$ | $Tg \times \frac{d}{c+f} = Vert_3$ | $Tg \times \frac{d}{c+f} = Vert_4$ |
| | | | $Tg \times \frac{re}{c+f} = Hor_3$ | $Tg \times \frac{re}{c+f} = \begin{matrix} Hor_4 \\ Hor_3 \end{matrix}$ |
| TOTAL RADIAL LOAD | $\sqrt{F_1^2 + (W_1 - Tg_1)^2} = TR$ | $\sqrt{F_2^2 + (W_2 - Tg_2)^2}$ | $\sqrt{(F_3 - Vert_3)^2 + (Tp_3 + Hor_3)^2} = TR_3$ | $\sqrt{(F_4 + Vert_4)^2 + (Tp_4 - Hor_4)^2}$ |
| THRUST LOAD | Tp | — | Tg | — |
| TOTAL LOAD | $\sqrt{TR_1 + Tp^2}$ | $\sqrt{F_2^2 + (W_2 - Tg_2)^2}$ | $\sqrt{Tg^2 + TR_3^2}$ | AS TOTAL RADIAL LOAD |

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## BEARING LOADS FOR WORM GEARS (Fig. 128)

| FORCE | BEARING NO. 1 | BEARING NO. 2 | BEARING NO. 3 | BEARING NO. 4 |
|---|---|---|---|---|
| F | $F \times \frac{a}{a+b} = F_1$ | $F \times \frac{b}{a+b} = F_2$ | $F \times \frac{rg}{c+d} = F_3$ | $F \times \frac{rg}{c+d} = F_4 = F_3$ |
| S | $S \times \frac{a}{a+b} = S_1$ | $S \times \frac{b}{a+b} = S_2$ | $S \times \frac{d}{c+d} = S_3$ | $S \times \frac{c}{c+d} = S_4$ |
| T | $T \times \frac{rw}{a+b} = W_1$ | $T \times \frac{rw}{a+b} = W_2 = W_1$ | $T \times \frac{d}{c+d} = W_3$ | $T \times \frac{c}{c+d} = W_4$ |
| TOTAL RADIAL LOAD | $\sqrt{F_1^2 + (S_1 - W_1)^2} = TR_1$ | $\sqrt{F_2^2 + (S_2 + W_2)^2} = TR_2$ | $\sqrt{W_3^2 + (F_3 - S_3)^2} = TR3$ | $\sqrt{W_4^2 + (S_4 + F_4)^2} = TR_4$ |
| THRUST | — | T | | F |
| TOTAL LOAD | $\sqrt{F_1^2 + (S_1 - W_1)^2}$ | $\sqrt{T_R^2 + T^2}$ | $\sqrt{W_3^2 + (F_3 - S_3)^2}$ | $\sqrt{T_{R4}^2 + F^2}$ |

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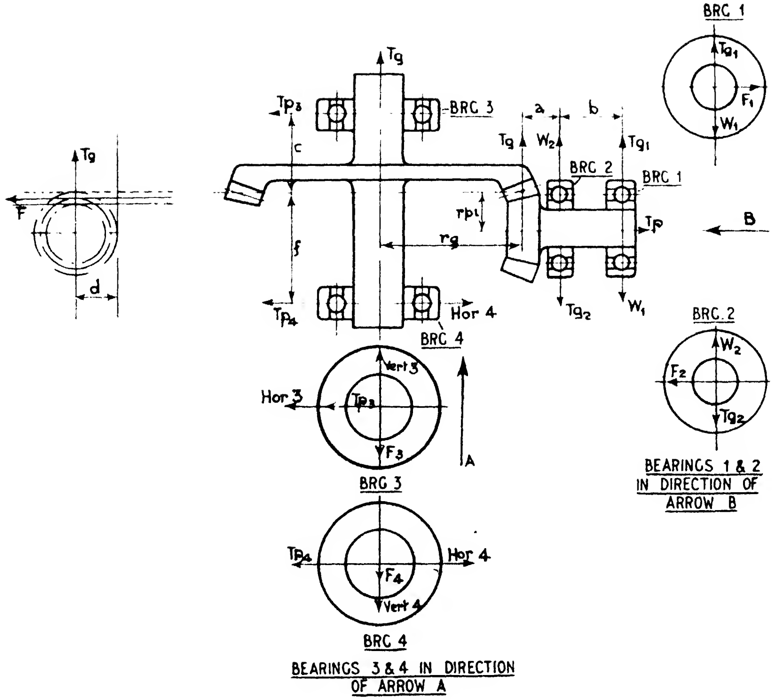



**Fig. 127**

*Bearing loads with hypoid gearing:* The loads on the hypoid-pinion bearings are of a similar nature to those of the spiral-bevel pinion, but those supporting the gear ring differ. There is a small amount of endways slide combined with the rolling action of the tooth in the hypoid gear, due to the tooth profile, which has been mentioned previously.

The following notation and expressions supply sufficient data to enable the bearing loads to be calculated:

$T_p$ = pinion thrust
$T_g$ = gear thrust
$F$ = tangential force
$\alpha$ = tooth pressure angle (drive side)
$\beta = \dfrac{\textit{Pitch cone angle (pinion)}}{2}$
$r_e$ = effective gear radius
$\delta_p$ = spiral angle (pinion)
$\delta_g$ = spiral angle (gear)
$d$ = pinion offset

$$\text{Tangential force } F = \frac{\text{Input torque (lb/in)}}{\text{Mean pitch radius (in) (pinion)}} = \frac{TQ}{rp_i}$$

$$rp_i = \frac{\textit{Pinion pitch diameter} - \textit{Tooth face} \times \sin\beta}{2}$$

$$r_g = \text{mean gear pitch radius} = rp_i \frac{N_g}{N_p} \times \frac{\cos\delta_p}{\cos\delta_g}$$

$$T_p = F\left(\frac{\tan\alpha\sin\beta}{\cos\delta_p} + \tan\delta_p\cos\beta\right)$$

$$T_g = F\left(\frac{\tan\alpha\sin\beta}{\cos\delta_p} - \tan\delta_p\cos\beta\right)$$

$$r_\varepsilon = \sqrt{r_g^2 - d^2}$$

## WORM DRIVE

The worm gear possesses several advantages over the other forms of drive. It has a high overload capacity, whilst the nature of tooth action is such that the effect of wear is less deleterious in comparison with other toothed gearing. On the other hand, the heat generated through the relatively high sliding velocity between worm and wheel necessitates higher working temperatures and consequent attention to adequate lubrication.

One of the reasons for the greater strength of the worm set as compared with the bevel gear is that the mean relative radius of curvature of the contacting surfaces is much greater. This also is true for the ordinary spur gear, where the reciprocal of the relative curvature radius is proportional to the sum of the reciprocals of pitch radii of the two gears, the value depending more upon the diameter of the small wheel than that of the larger gear.

In a bevel gear of high ratio the pinion is small and, whilst the surface stress is high, the torque capacity of the wheel is relatively low. As an approximate comparison, with a bevel and worm of 14 in diameter, the pitch radius of the worm wheel is 7 in and the pitch radius of the worm, which is virtually a rack, is infinity. The relative curvature radius therefore varies as $\dfrac{1}{\frac{1}{7} + \frac{1}{\alpha}}$ which equals 7 in radius. The pitch radius of the bevel wheel is 7 in radius and that of the bevel pinion = 1 in, the relative radius in this case varying as $\dfrac{1}{\frac{1}{7} + \frac{1}{1}} = \frac{7}{8}$ in radius.

With regard to efficiency, however, the worm gear is hardly comparable with the bevel or helical gear, the orders being approximately 94 against 98 per cent respectively. This loss of efficiency is occasioned, it is believed, partly because of the high ratio of mean velocity of sliding to the mean circumferential velocity of the wheel, whilst the power loss through tooth friction is another important factor. If tooth friction only is taken into consideration, and the worm regarded as a coiled wedge of effective angle equal to the lead angle of the worm at mid depth of thread, then the efficiency may be expressed by the formula:

$$E = \frac{\tan \lambda}{\tan (\lambda + \phi)} \text{ with worm driving}$$

and $$E_1 = \frac{\tan (\lambda - \phi)}{\tan \lambda} \text{ with wheel driving,}$$

where $\lambda$ = lead angle of the worm

$\phi$ = angle of friction between worm and wheel.

A lead angle of about 45° gives the highest efficiency, although considerations other than those of a theoretical nature may dictate an appreciably smaller angle, whilst it should be appreciated that the greater the lead angle the smaller must be the diameter of the worm. This consideration is important as naturally with the decrease in worm diameter the bending stresses increase, as does also the deflection of the worm shaft under the influence of tooth load. The high local stresses thus produced lead to a higher coefficient of friction, so that actually a smaller lead angle than $45° - \frac{\phi}{2}$ may possibly give better results in practice. The deflection when transmitting maximum torque is, of course, greater than when operating in top gear, and it is possible to ascertain the amount allowing for this in the initial setting.

However carefully the worm assembly is designed, a certain amount of flexibility in the axle case, worm housing, and in the bearings is inevitable, and conditions of contact between worm and wheel are therefore disturbed.

To counteract this effect the axial position of the worm wheel may be adjusted so that contact in the unloaded condition is towards the leaving side of the wheel tooth. The amount of initial offset is, of course, found by trial and marking. Slight axial movement of the worm wheel in one direction causes movement of the contact bearing in the opposite direction. In this respect a worm having a large lead angle is more sensitive than the smaller angle.

Since it is not possible to check the face bearing when the set is under load it is generally satisfactory if, for a light load, contact bearing is concentrated towards the leaving side and covering about 75 per cent of the available area of tooth flank. Rigidity of mounting is therefore of primary importance and it is desirable to maintain the axial position of the worm wheel rim to within $\pm$ 0·0015, a figure which in practice has been found satisfactory.

To accomplish this the journal bearings must allow only a little radial play and the thrust bearings must be accurate, for it will be appreciated that non-axial end thrust tends to cause the wheel to tilt in the plane containing its own axis and the common perpendicular to the axes, in which case lateral movement of the rim relative to the worm axis will occur. It is for this reason that the bearing-spur ratio to worm-wheel diameter must be as small as possible, whilst in order to minimise the effect of any relative displacement between worm and worm wheel which may occur in a direction parallel to the worm-wheel axis, the bearings must be given adequate support in the form of rigid caps and well-designed bolts. The worm-gear carrier should also derive some support against distortion, by means of a steady spigot machined in the banjo or main-axle casing.

Finally the question of lubrication must be considered. It would be preferable to use a castor-base oil, if replacement at short regular intervals could be maintained. An oil with such a base has a low coefficient of friction, but it also has a tendency towards rapid deterioration. A mineral oil is therefore usually recommended which can retain its lubricating properties at working temperatures of 200° F, a figure which is often found in heavy-duty drives.

Adequate sump capacity is vital and the casing dimensions should permit dipping of one or other of the gears under all circumstances. If lubrication is unsatisfactory, which may be produced by reason of oil-film failure or faulty gear alignment and consequent imperfect tooth contact, wear and abrasive action producing bronze dust will be noticeable. The oil film may fail due to many causes, although

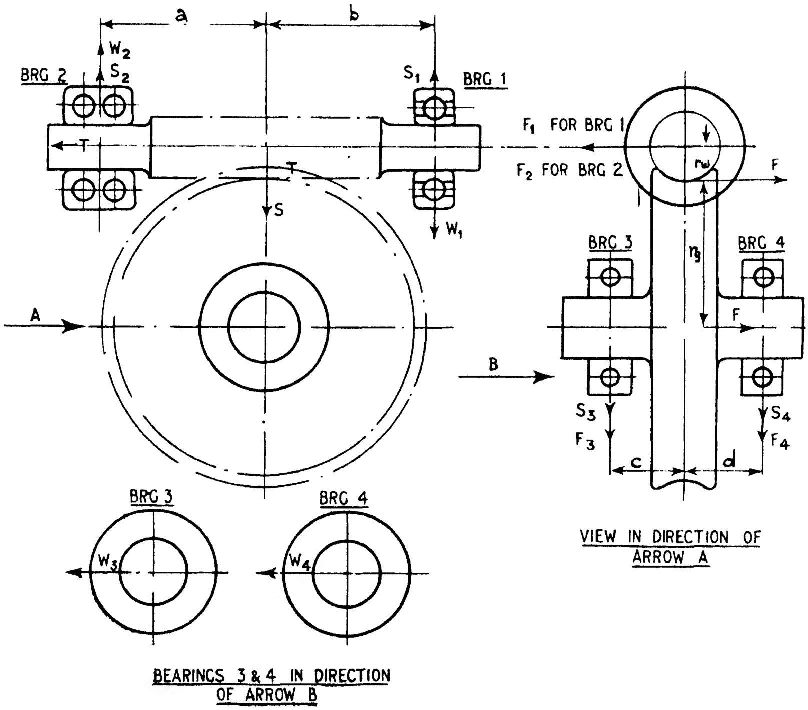



Fig. 129

the most common reason is from inadequate entry gap, which is usually accompanied by high temperatures. The remedy is to form a clearance, by careful filing, between the entering edges of the teeth and by resetting the worm wheel. The entry of lubricant between the thread and tooth will thereby be facilitated.

Should the tooth stresses be of high order "pitting" of the worm-wheel teeth is likely to occur. Such a phenomenon is not unduly alarming, as failures are rare. Pitting occurs early in the gear life and after the initial development it would appear that the rate at which the pitting extends is considerably reduced. For the purpose of bearing-load calculations, the normal tooth force is assumed to comprise (a) a tangential force at the pitch radius of worm F, (b) a force tending to separate the worm from the gear S, and (c) a thrust T produced by the lead angle of the worm (see Fig. 129).

$$\text{The tangential force} = \frac{\textit{Torque input}}{r_w}$$

where $r_w$ = pitch radius of worm (in)

and $r_g$ = pitch radius of worm gear (in)

$$r_g = \frac{1}{2\pi}(N_g \times \beta) \quad \text{if } N_g = \text{number of teeth in gear and } \beta = \text{axial worm pitch.}$$

$$\text{The separating force } S = \frac{F \tan\alpha}{\tan\gamma} \quad \text{where } \alpha = \text{tooth-pressure angle}, \ \gamma = \text{lead angle of worm}$$

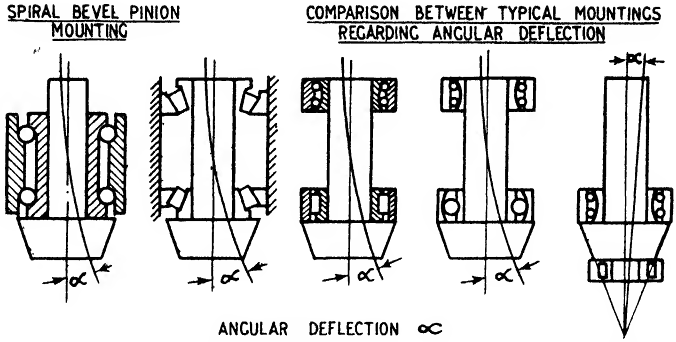
$$\text{The worm thrust } T = \frac{F}{\tan\gamma} \qquad \gamma = \tan^{-1}\frac{\text{lead}}{2\pi r_w}.$$

## MOUNTING OF BEVEL PINION AND WHEELS

A few general observations in connection with the mounting of bevel pinions and wheels, whether spiral or straight teeth, are necessary at this point. It is known that shock loading is not the sole reason for gear-teeth failure, but rather the resulting fatigue, and although all parts of the gear assembly assume some degree of deformation when load is applied, which moves the theoretical tooth contact line, satisfactory mounting can do much to prevent failure from localisation of tooth loads, and stress concentration.

The overhung pinion tends to distort in such a manner as to localise tooth loads on the heel of the tooth, whereas the provision of an outboard bearing restricts distortion to the extent that the pinion merely tends to rotate around its apex (see Fig. 130).

Experiments in the United States by J. D. Almen ("Automotive Industries," 1935) indicate the importance of correct pinion mounting in relation to pinion deflection. In the tabulated results, all bearings had similar capacities and all drives were subjected to the same torque value. The angular deflections were obtained with bearings having tolerances of maximum shaft and maximum housing tightness. These deflections vary, of course, as the bearing tolerance slackens and for the slack limit figures quoted in makers' catalogues the figures in the table become modified as follows:




ANGULAR DEFLECTION α

| | | | | | |
|---|---|---|---|---|---|
| FORWARD LOW GEAR | 107° | ·125° | 140° | 160° | ·005° |
| REVERSE | 215° | 235° | 162° | ·215° | ·040° |

**Fig. 130—Relative distortion due to type of pinion mounting**

| | | | | | |
|---|---|---|---|---|---|
| FORWARD LOW GEAR | 0·135 | 0·15 | 0·206 | 0·225 | 0·01 |
| REVERSE | 0·260 | 0·24 | 0·215 | 0·320 | 0·045 |

In the spiral-type bevel gear the effect of angular deflection is somewhat compensated by the effect of deflection in the direction of pinion axis, but nevertheless it will be seen that wherever it is possible to straddle mount the pinion it is desirable. A further consideration is the correct apportioning of stress between the mating gears, bearing in mind that the allowable stress in the ring gear is higher than that of the pinion, on account of the gear ratio. If, for example, a ratio of 5 to 1 be used, each ring-gear tooth is stressed with a frequency of that of a pinion tooth, from which it will be assumed that the required minimum life of the ring gear would be calculated on the basis of the number of stress cycles of the pinion.

In order to obtain the maximum life from bevel gearing, the gears must be adjusted correctly for tooth contact. The desirable contact area should be that which commences at the toe and which extends approximately 80 per cent towards the heel of the tooth lengthwise. A powdered red lead mixed with light gear oil should be used when marking for contact on the drive side of the gear tooth.

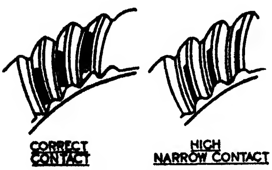



**Fig. 131**

If gears are allowed to operate with an adjustment giving high narrow contact, noise, galling and rolling over at the tooth top will result. In such a case adjustment should be made by moving the pinion in towards the gear. Naturally, the backlash will also be decreased, but this can be corrected by moving the gear away from the pinion to provide about 0·005–0·008 in backlash (see Fig. 131).

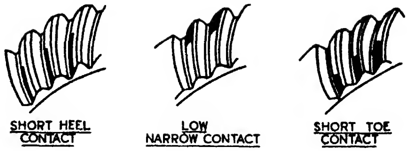



Fig. 132

If low narrow contact is in evidence the teeth will groove at the base. Move the pinion out from the gear sufficient to rectify the tooth contact marking (see Fig. 132).

Short-toe contact is another maladjustment which can be rectified by moving the gear away from the pinion, this having the effect of lengthening the contact in addition to moving it towards the heel. If the short-toe contact is allowed to persist, the gear teeth will chip at their edges, and excessive wear and probable breakage will occur. A short-heel contact gives similar results, but in this case the gear must be moved towards the pinion in order to move the contact area towards the tooth toe (see Fig. 133).

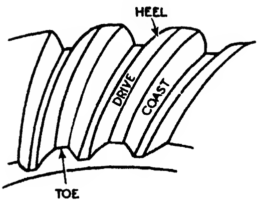



Fig. 133

Comparison between the principle of the hypoid gear and the spiral-bevel gear cannot be made unless it is appreciated that both types are based fundamentally upon pitch surfaces which rotate with constant-velocity ratio about their axes. The spiral-bevel surface produced forms a cone, and when two such gears mate the apex of each cone is coincident. Each rotates about its axis with pure rolling motion (Fig. 134). The " hypoid " gear, which nomenclature is an abbreviation of the " hyperboloid " of revolution, operates not as the spiral bevel, with coincident axes, but about oblique but not intersecting axes. The pitch surfaces of the hypoid are the hyperboloids of revolution and the enveloping form produces a curve to hyperbolic law (Fig. 135).

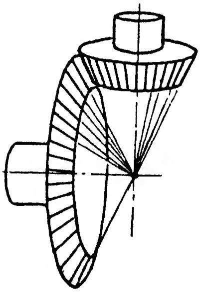


Fig. 134

The main and essential difference between the surfaces of the cones for bevel gears and the surfaces for hypoid gears is that, whereas in the spiral gear the action is purely rolling, the hypoid surfaces supplement this action by sliding action upon one another. The tangency in this case is along a rectilinear element and the rolling action is

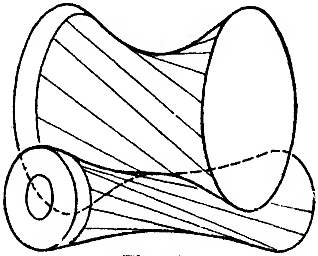


Fig. 135

perpendicular to this element, whilst pure sliding occurs longitudinally.

## HYPOID GEARS

Due to the offset position of the hypoid pinion, the profiles of the opposite sides of the teeth are not symmetrical, the concave side being flatter than the convex side. It is therefore necessary, in order to obtain approximately the same conditions of tooth contact on both sides, to make the pressure angles unequal. These angles have been found to be most satisfactory at $17\frac{1}{2}°$ on the driving side and 25° on the coasting side. Incidentally, for equal conditions of tooth contact on both sides, the following conditions must be satisfied:

(a) Equal arcs of action.
(b) Equal duration of contact.
(c) Similar relative radii of curvature of profile.
(d) Equal freedom from undercutting.

An increase in the spiral angle increases the pinion diameter, thus improving the number of teeth in contact, and in turn this increases the axial thrust in the normal tooth load. However, it is generally understood that the load-carrying capacity increases more rapidly than the axial thrust and the best compromise appears to be effected with a pinion-spiral angle of 50° and a gear-spiral angle of 25°. Due to the unsymmetrical relationship between gear and pinion, these figures mean an approximate offset of $1\frac{3}{4}$ in. Compare this with a normal set of spiral bevels.

For equal smoothness, a spiral angle of at least 45° would be necessary and say a 16° pressure angle for both, whilst the tooth load for the hypoid is 15 per cent higher approximately than the tangential load, the spiral bevel normal tooth load being 36 per cent greater than its tangential loads. The maximum tooth loads recommended for automobile rear-axle sets are 1,600 lb/in of gear face width direct drive, and 4,200 lb/in of face width in low gear.

It will be seen from the foregoing that for the increased diameter of a hypoid pinion, as compared with the plain-spiral pinion, a considerable increase in the tooth strength is obtained. It is recommended that the offset for hypoid should not exceed one-eighth of the gear diameter.

## HYPOID GEAR MOUNTING

Rigidity of mounting must be carefully considered, for flexibility introduces concentration of tooth bearing with corresponding concentration of tooth stress. Within the following limits a hypoid gear set may be considered satisfactorily mounted:

(1) Maximum displacement of the pinion in any direction should not exceed 0·003 in.
(2) Maximum displacement of the gear, vertically or in any direction, should not exceed 0·003 in.

(3) Movement of the gear away from the pinion at point of mesh should not exceed 0·008 in.

If taper-roller bearings are utilised they should be preloaded (to reduce their " elastic yield ") by adjustment during assembly. Straight-roller bearings must have a radial freedom of not more than 0·0004 in.

In view of the inherent weakness in straight-sided splines during recent years the trend has been to develop the stub-type involute spline. Such a spline possesses three main advantages over the earlier straight-side type insofar as the tooth has maximum strength through its minor diameter, the involute spline is self-centreing, which has the tendency to distribute the stresses equally throughout all teeth, and the fact that it is not normally necessary to grind a spline on its involute curve.

The most economical pressure angle is 30 degrees, whilst it is not generally necessary to employ a length of tooth longer than its pitch diameter. In practice, the equivalent of about one-quarter of the teeth are in contact.

The general formulae for shaft and teeth proportions may be derived from the nomenclature:

$T$ = torsional moment (lb)
$f_s$ = allowable torsional shear stress (lb/sq. in)
$f_c$ = allowable compressive bearing stress (lb/sq. in)
$A$ = area of one tooth in contact
$L$ = length of spline (in)
$R$ height of contact on tooth = 0·8 DP or 0·8 D/N
$D$ = minor diameter of shaft
$D_p$ = pitch diameter.

$$T = \frac{\pi D^3 f_s}{16} \text{ and } f_s = \frac{16T}{\pi D^3}$$

If all teeth are in contact the expression for T may be rewritten

$$T = \frac{f_c A N . D_p}{2} \text{ and because } A = Lh = \frac{\cdot 8 L D_p}{N}$$

$$T = \frac{f_c L D_p^2}{2 \cdot 5} \text{ whilst } L = \frac{2 \cdot 5T}{f_c D_p^2} \text{ and } f_c = \frac{2 \cdot 5T}{L D_p^2}$$

For shear stress at pitch diameter where the area in shear is

$$\frac{\pi D_p L}{2} \text{ or } 1 \cdot 570\, D_p L$$

$$f_s \text{ will equal } \frac{2T}{D_p(1 \cdot 570 D_p L)} \text{ or } \frac{1 \cdot 273\, T}{L D_p^2}$$

$$\text{and } L \text{ (length of spline)} = \frac{1 \cdot 273\, T}{f_s D_p^2}$$

The allowable shear stress is assumed to be about 60 per cent of the ultimate tensile and the compressive stress as 140 per cent of that figure so that the ratio of $f_s$ to $f_c$ is 0·428 to 1, but by dividing the results from the above formulae for $f_s$ and $f_c$ we find that the ratio is 0·509 to 1, which indicates that the design of spline is critical in shear.

If the two expressions for fs be combined it will be found that $L = 0 \cdot 25\, D^3/D_p^2$. Since only about one-quarter of the number of teeth are in contact this expression will be modified to form a general basis for proportions to $L = D^3/D_p^2$.

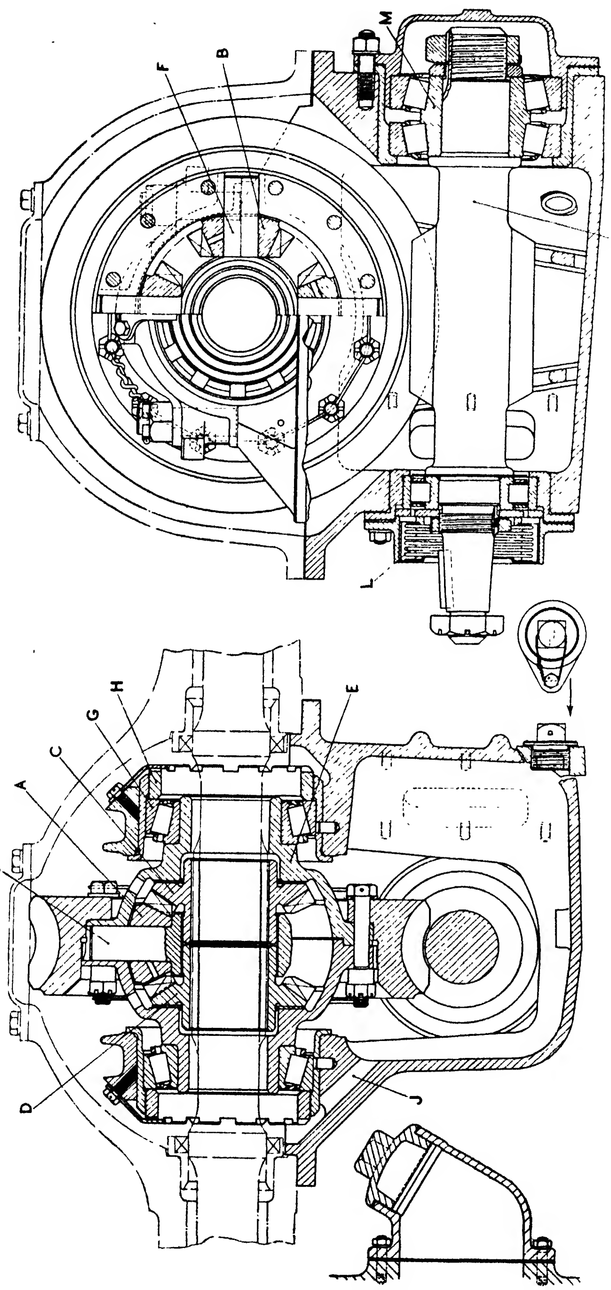



**Fig. 136—Typical heavy-vehicle differential**

A, B, C and D, differential levels; E, differential bevels; F, differential star piece; G, taper roller bearings; H, adjusting nuts; J, oil return groove; K, worm gear; L, bellows oil seal; and M, worm thrust bearings

## CHAPTER 8

# THE DIFFERENTIAL

THE differential is interposed between the two drive shafts of the rear axle in order that the torque or effort applied to each may be equal although the speeds of revolution may differ. This is necessary when the vehicle is turning in a circle, when the outer wheels are required to rotate at greater speed than the inner wheels due to the greater distance they have to travel; but when the vehicle is travelling in a straight line there is no relative motion between any of the differential gears.

In principle the two bevel gears C and D are fixed to the two axle shafts, whilst the bevel pinions A and B are in constant mesh with both gears and are attached to the differential cage E through a pin or two pins in the case of a four-pinion set. This cage is bolted to the final-drive gear. If the cage E is held fixed it will be clear that if the gear C be rotated in a forward direction gear D will be rotated backwards at the same speed (see Fig. 137).

If, on the other hand, the cage be rotated in a forward direction and the wheel C is still rotating forwards in relation to the cage, then D will still turn backwards relative to the cage. Assume the speed of the cage to be 250 r/min forward and the gear C turning forwards at say 5 r/min then gear D still turns backwards at equal speed of 5 r/min. So the actual speed of C is 255 r/min since its forward motion is added to that of the cage, whilst the speed of gear D will be 245 r/min as its backward motion relative to the cage is subtracted from the speed of the cage. The torques transmitted to each of the two axle shafts are equal, and each is equal to half the torque applied to the cage through the final-drive gear.

The total torque is the load on the differential pinion pin, whilst half the torque is the load applied at the pinion teeth, from which dimensions of the pin and gear teeth may be calculated.

The bevel- and spur-gear types of differential are very efficient parts of the transmission, due in the main to the small degree of friction developed and it is because of this efficiency as a compensating device that several undesirable features occur which affect both performance and wheel action at the ground. Should

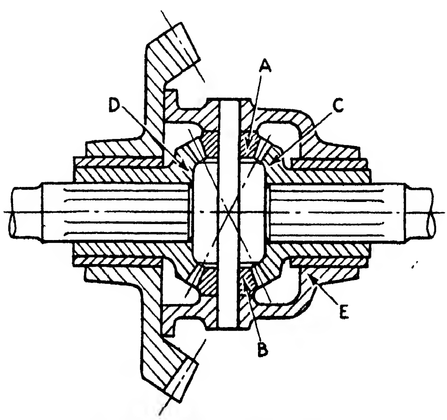


**Fig. 137—Bevel differential**

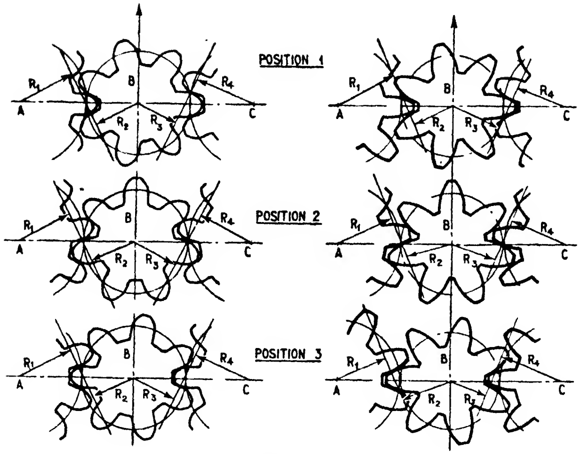



Fig. 138

one wheel momentarily slip, or leave the ground for any reason when the vehicle is riding over an uneven surface, that wheel rapidly increases its speed of revolution owing to the slight internal resistance offered by the differential gear. The following retardation of the wheel speed, as it strikes the ground upon return, imparts by way of the differential an impulsive or jerky acceleration to the other wheel, involving a difference in velocity between the two revolving wheels. This is one of the causes of vehicle skid when there appears to be no apparent reason.

Moreover, should the vehicle be stationary with one wheel resting upon slippery or soft ground, that wheel will spin upon the application of torque as the differential affords no material resistance. In consequence, instead of propelling the vehicle, it merely sinks lower into the ground.

The bevel-gear tooth profiles are partly responsible for the equal division of torque between the driving wheels, as it will be understood that with involute form of tooth the normal line, that is a line normal to the tooth curvature at the point of contact, will pass through the point of contact of the pitch circles and therefore the ratio of tooth leverage between the two axle-shaft gears will be constant with a value of unity.

Reference to Fig. 138 will show a bevel-pinion meshing with two bevel gears at three different stages of revolution around the axis of the axle-shaft gear. $R_1$, $R_2$, $R_3$ and $R_4$ represent the lever arms of each side gear and pinion (since they are drawn perpendicular to the normal line) and the ratio of leverages is:

$$\frac{R_1}{R_2} \times \frac{R_3}{R_4}.$$

It will be noted that for all three positions these arms are similar so that in each case the ratio is unity, or in other words the torque between the two driving wheels is equal.

Refer now to the gears in Fig. 138. The tooth design is such that a considerable variation of leverage is allowed between the dedendum and addendum portion of the teeth, and it is apparent that the lever arms vary in length in each position. This being the case, the ratio of axle shaft gear A varies uniformly to that of gear C when pinion B rotates about its axis, thus: 1 + : 1, 1 : 1 +. Further rotation of B changes the ratio again uniformly from 1 : 1 +, 1 + : 1.

This cycle of torque-ratio distribution may, of course, occur as many times during the revolution of one driving wheel (assuming that the opposite wheel is prevented from rotation) as there are teeth in the axle-shaft gear. It is claimed that with this periodic transfer one wheel cannot spin although, for conditions of equal traction of the driving wheels and straight-ahead driving, the pinion and gears assume the position shown in Fig. 138 (position 2) when torque distribution is equalised.

It is therefore desirable, in a gear having plain involute teeth, to provide some degree of friction in the compensating gear, and numerous devices have been designed to this end. Attempts have also been made to lock the differential as conditions render such action necessary.

## ZF SELF-LOCKING DIFFERENTIAL

One of the friction devices which found great favour on the Continent is the ZF self-locking differential, and its basis is that of very low transmission efficiency, or in other words, it possesses considerable friction. This friction is in turn occasioned by relatively high pressure between the carrier dogs and the curved paths. When the vehicle is starting, having resistance at one rear wheel only, the friction is increased by the back pressure set up by acceleration of the slipping wheel. This high internal friction virtually transforms the differential into a self-retarding transmission gear which does not permit wheel-speed difference. It does not, however, dispense with its compensating qualities (see Fig. 140).

When driving round a bend the differential is controlled by the wheels, the parts transmitting the torque, such as the axle shaft, being subjected to an angular change

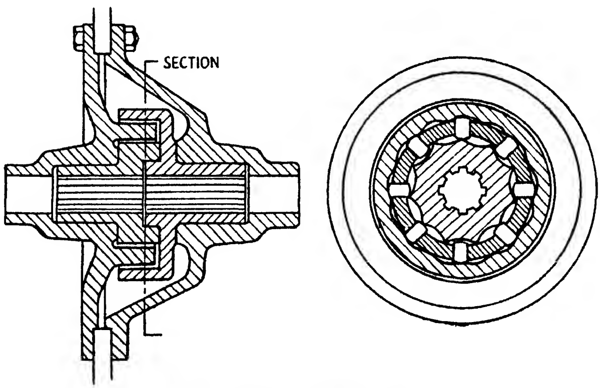



**Fig. 139—ZF differential**

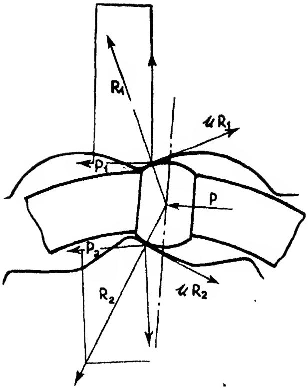



Fig. 140

in their respective position, whilst the driving parts are relatively stationary. The construction of the differential is simple, constituting a cam drive in which one of the axle shafts is connected to the inner and the other shaft to the outer cam body, the bevel or worm final drive being effected by the sliding shoes guided in the central cage. Reference to the sketch will make this clear, whilst the section through the cam and track will illustrate the reason for the high friction values upon which the operation of this differential gear is based.

Another form of differential gear is shown in Fig. 141; this employs spur gears instead of bevel gears. It performs similar duties and is widely popular overseas. Wheels A and B are fixed by keys or splines to the two axle shafts. A long spur pinion E meshes with wheel A, the teeth of E not quite reaching those of wheel B. Another spur pinion F meshing with E also meshes with wheel B. The pinions are carried on pins supported by the differential case, the case in turn forming the mounting for the specific type of final drive—bevel, worm, or spur gear.

Torque is conveyed to gear A through pinion E, and the tooth pressures tend to make E revolve upon its pin. This pressure is opposed by pressure between the teeth of the two pinions at the centre which in turn makes pinion F revolve. Obviously in view of this, a similar condition arises between pinion F and wheel B. Should pinion E not be rotating on its pin, then the two pressures acting upon it must be equal and therefore equality exists between the pressures on the teeth of gear and pinions A and B, and E and F. Torques, therefore, are also equal.

Several sets of spur pinions are usually employed, each pair being equally spaced around the periphery of gears A and B. If the differential case C is held to

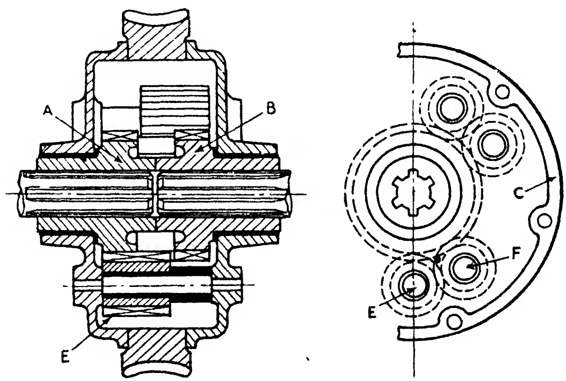



Fig. 141—Spur gear differential

prevent rotation and wheel A is rotated in anti-clockwise direction, then pinion E will rotate in clockwise direction, and pinion F in anti-clockwise rotation, wheel B will obviously rotate clockwise. If, therefore, one of the differential wheels rotates faster than the case, the other wheel rotates exactly the same amount slower than the case; this is a similar action to that which occurs in the bevel-type differential.

We have discussed briefly the question of friction and some of the reasons for its desirable presence and in this connection, in order to prevent or to control the degree of wheel slip, resort is sometimes made to a differential lock. Such mechanisms provide for the locking of one or both differential wheels to their casing or by the prevention of rotation of the pinions on their pins or " stars ", in which case the differential ceases to function and the whole of the torque is then applied to the crown wheel and transmitted to whichever road wheel retains its adhesion with the road surface. The importance of such a mechanism is more apparent in the cross-country vehicle or the tractor than in a light truck or passenger car, when the ground surface may vary between wheel and wheel. The absence of such a lock may under such circumstances render the vehicle totally inoperative, since any applied torque to a slipping wheel merely aggravates that condition.

The splines for the driving of the differential wheels will be assumed to be of the involute type, the basic formulae for tooth proportions being well known as follows:

$$\text{Circular pitch} \doteq \frac{\pi}{\text{Diametral pitch}}$$

$$\text{Addendum and dedendum} = \frac{0{\cdot}50}{\text{Diametral pitch}}$$

$$\text{Circular tooth thickness} = \frac{1{\cdot}570}{\text{Diametral pitch}}$$

$$\text{Pitch diameter} = \frac{\text{Number of teeth}}{\text{Diametral pitch}}$$

$$\text{Major diameter} = \frac{\text{Number of teeth plus one}}{\text{Diametral pitch}}$$

$$\text{Minor diameter} = \frac{\text{Number of teeth minus one}}{\text{Diametral pitch}}$$

whilst they apply equally to internal and external shafts.

We have previously seen the weakness of the straight spline with its undercut at the spline base, but with the involute type torsional tests have shown that the strength is equivalent to a shaft larger than the minor diameter, which indicates that with the fillet the spline teeth actually add strength under fatigue loading.

Moreover the involute provides greater bearing surface and tooth contact and this permits closer fits with decreased noise and greater possibilities of interchangeability. From these aspects alone it is highly probable that the straight-sided spline will eventually be superseded in favour of the involute type.

# CHAPTER 9

# THE CHASSIS

## CHASSIS LAYOUT

ASSUMING that the design of the main units has been agreed, the procedure of installing them in the chassis to the greatest advantage is one which calls for particular skill, for not only must the designer possess a sound knowledge of the loads and stresses encountered and at the same time know how to dispose the units to counteract any circumstances detracting from the best vehicle performance, but he must be able to visualise assembly detail even before that stage in design is attained.

The main aim must be to produce as few a number of chassis parts as is consistent with sound engineering, utilising as many multi-purpose pieces as possible. It will be appreciated that simplicity in design usually produces the most reliable mechanism, and simplicity cannot be achieved if the chassis design is treated in a series of individual sections. It must be considered as a whole.

As an example, it will have been seen in previous pages how the relation of track to wheelbase affects stability and, if the vehicle be of heavy type for cross-country work, how its performance can be affected if a satisfactory wheel loading is not arranged, almost to the extent of rendering the vehicle incapable of movement under certain conditions. Further, it will have also been noted that suspension, braking, and steering are so interdependent that it would be inadvisable to lay out a design of either without giving due consideration to the effects of each one upon the other.

It is therefore somewhat difficult to describe all that takes place during the layout of a chassis, and in attempting to do so it must be presumed that the designer keeps in the foreground the necessity for the checks and cross-checks of dependent units, so vital in the design of a successful chassis.

Most well-designed vehicles possess a certain degree of " character " or " styling " which is the direct result of the particular designer's outlook and method of approach, not only of general layout, but of the smallest detail design, consistency in which is essential to neatness. Neatness in chassis design covers many features, such as correct balance of component location, absence of superfluous parts, simplicity of mechanism, and consistency in design of details of similar nature.

Nothing must be taken for granted in chassis layout, as a collection of excellent individual units is not sufficient to ensure a good chassis; neither does it follow, even if all the individual chassis details are well conceived, that the general design will be all that can be desired, as if the parts most likely to require attention or periodical inspection are arranged in inaccessible positions any

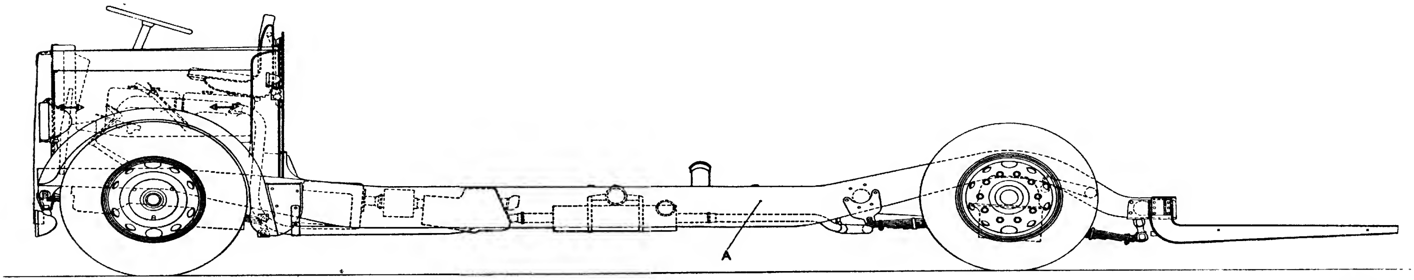



Fig. 142

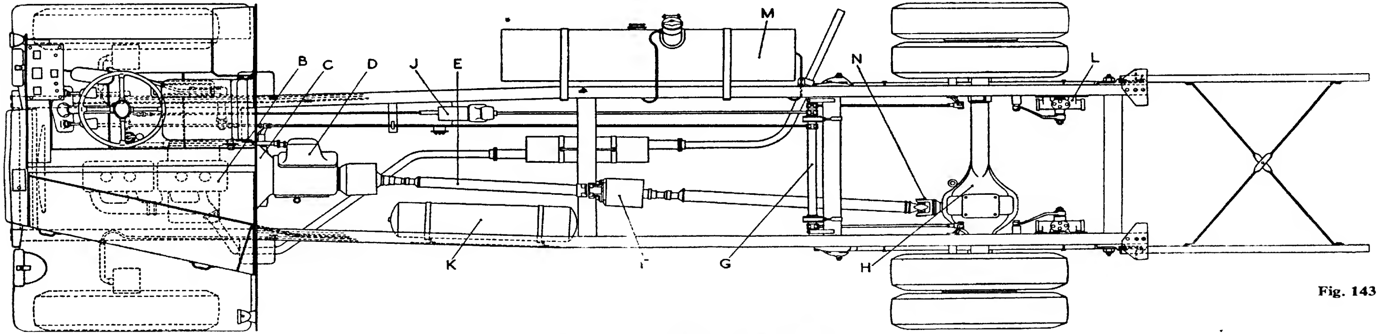



Fig. 143

**Figs. 142 and 143—Typical passenger-chassis layout**

A, frame; B, engine; C, clutch; D, gear box; E, propeller-shaft; F, propeller-shaft centre bearing; G, brake cross shaft; H, rear axle; J, servo motor; K, vacuum reservoir; L, rear shock absorbers; M, main fuel tank; and N, universal joints

[*To face page* 184

merit which attaches to that individual part is liable to become nullified by continued lack of attention, and possible consequential failure.

Accessibility is not confined to simply being able to work on a unit; it means also that a component must be able to be withdrawn without fouling any other part, whilst it should not be necessary to dismantle an adjacent unit in order to gain access to another component.

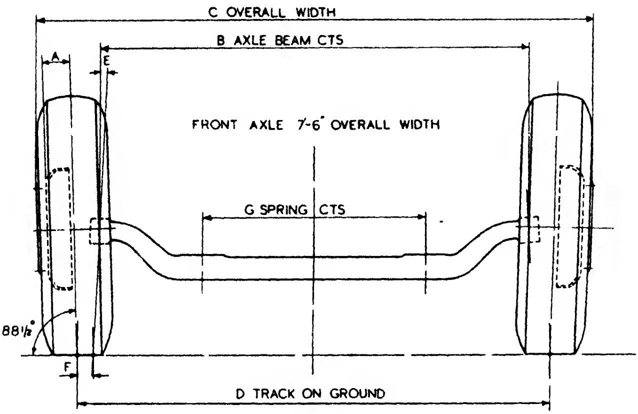
Manoeuvrability, " feel " of control, and operating comfort should take high places in initial considerations. The foregoing represents but a very small proportion of the technical problems to be borne in mind in a successful chassis layout, whilst on the other hand the question which is constantly before the designer, and which often influences a design, is its cost.

The cost of an article depends mainly on the material, labour and overhead charges. The former changes infrequently, whilst labour costs are liable to more frequent disturbances and vary with every different article produced, whilst at the same time they are further affected by the type of labour and its quality. It is therefore essential that materials are carefully chosen in accordance with the requirements of strength, and that suitable, but not superfluous, machining be specified. By suitable machining is meant that type of work which does not require a more expensive type of labour (or a more complicated machine to perform that work) than is necessary for the correct functioning of the part. This, of course, refers mainly to the detail stage, but in the layout of an assembly it must still be borne in mind.

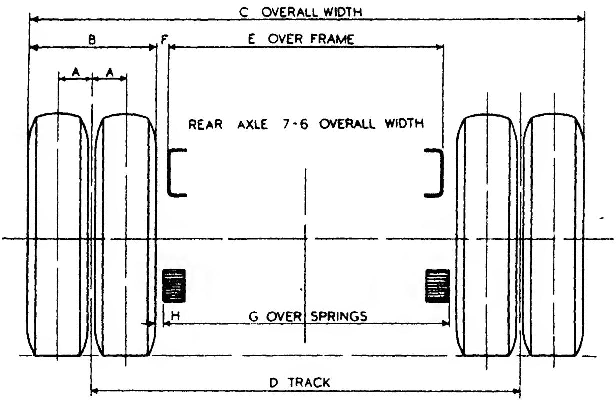
Lastly, there is the question of Acts of Parliament and regulations which affect the operation, construction and equipment of the vehicle. Such legislation was designed to ensure the public safety and is rigidly enforced. Therefore any misinterpretation or deviation from the restrictions set down render the vehicle illegal. It is not possible nor desirable to discuss these in detail, but reference will be made to those relevant when the appropriate stage in layout is reached. A very brief summary of the more important dimensional ones is set out in the following pages and the more important descriptive clauses under their appropriate sections. The actual drawing-office procedure follows somewhat on these lines, depending upon the type of vehicle under consideration, which for this example will be a public service vehicle.

There is, of course, a certain amount of preparatory work to be laid down and the first consideration must be that of tyre equipment, which involves wheels, spring centres, frame width, wheel offset at ground and track. The overall width must not exceed 7 ft 6 in or 8 ft 0 in in special cases and this, once the tyre equipment is decided, confines the main transverse dimensions to within certain limits.

If a complete range of vehicles is under consideration the problem of twinning rear wheels and standardising frame widths for different

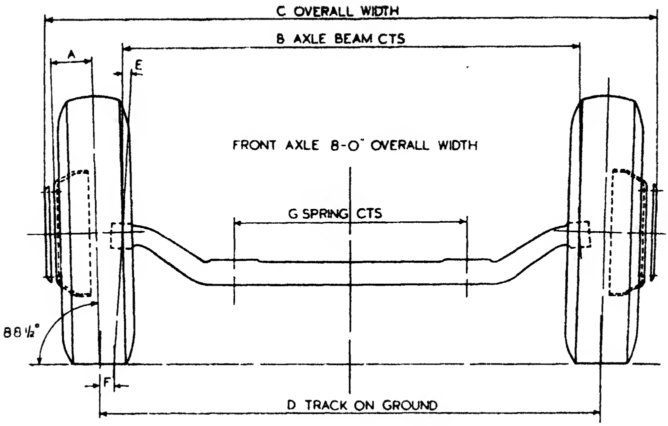



| TYPE OF VEHICLE | TYRE SIZE | TYRE CAPACITY (PLATED) | RIM SIZE | OFFSET A | AXLE BEAM CTS B | OVERALL WIDTH C | TRACK D | KING PIN ANGLE E | F | SPRING CTS G |
|---|---|---|---|---|---|---|---|---|---|---|
| S D BUS | 9 00" x 20" | 3 TONS 10CWTS | 4 PIECE RIM 6 00 x 20" | 5 375" | 69l" | 89 75" | 74 43" | 3° | 1 5" | 36 0 |
| D D BUS | 11 00" x 20 | 4TONS 15CWTS | 4 PIECE RIM 733" x 20" | 4 375" | " | " | 76 43" | " | 2 5" | " |
| | | | | | | | | | | |

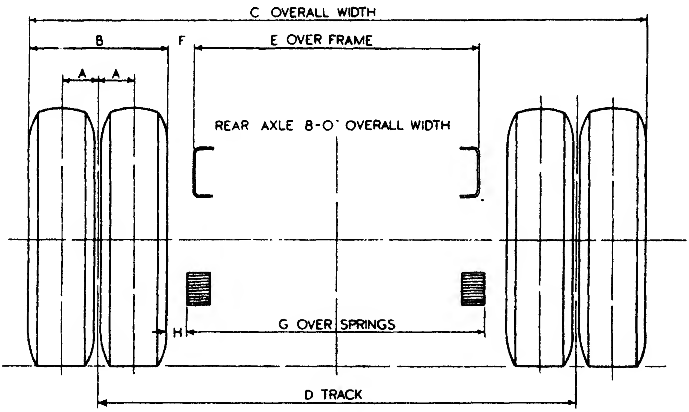



| TYPE OF VEHICLE | TYRE SIZE | TYRE CAPACITY (PLATED) | RIM SIZE | OFFSET A | WIDTH OVER TWINS B | OVERALL WIDTH C | TRACK D | WIDTH OVER FRAME E | TYRE CL F | WIDTH OVER SPRINGS G | TYRE CL H |
|---|---|---|---|---|---|---|---|---|---|---|---|
| S D & D D BUS | 9·00 x 20 | 7 TONS | 4 PIECE RIM 6 00" x 20" | 5 375" | 20 45" | 89 25" | 68 8" | 44 0 | 2 17" | 46 0" | 1 17" |
| | | | | | | | | | | | |

**Fig. 144—Typical diagram, width clearances 7 ft 6 in overall**




| TYPE OF VEHICLE | TYRE SIZE | TYRE CAPACITY (PLATED) | RIM SIZE | OFFSET A | AXLE BEAM CTS B | OVERALL WIDTH C | TRACK D | KING PIN ANGLE E | F | SPRING CTS G |
|---|---|---|---|---|---|---|---|---|---|---|
| S D BUS | 9 00" x 20" | 3 TONS 10 CWTS | 4 PIECE RIM 6 00 x 20 | 5 375" | 69 1" | 94 75" | 79 43" | 3° | 4 0" | 36 0" |
| S D COACH | 10·00" x 20" | 4 TONS | 4 PIECE RIM 6 00" x 20" | 5 625" | " | " | 78 93" | " | 375" | " |
| D D BUS | 11 00" x 20 | 4 TONS 15 CWTS | 4 PIECE RIM 6 00 x 20" | 6 000" | " | " | 78 18" | " | 3 375" | " |
| " " | " " | " " | 4 PIECE RIM 7 33 x 20" | 6 375" | " | " | 77 43" | " | 3 0" | " |




| TYPE OF VEHICLE | TYRE SIZE | TYRE CAPACITY (PLATED) | RIM SIZE | OFFSET A | WIDTH OVER TWINS B | OVERALL WIDTH C | TRACK D | WIDTH OVER FRAME E | TYRE CL F | WIDTH OVER SPRINGS G | TYRE CL H |
|---|---|---|---|---|---|---|---|---|---|---|---|
| S D BUS | 9 00" x 20" | 7 TONS | 4 PIECE RIM 6 00 x 20" | 5 375 | 20 45" | 94 25" | 73 80" | 48 0" | 2 67" | 50 0" | 1·67" |
| S D COACH | 10·00" x 20" | 8 TONS | 4 PIECE RIM 6 00" x 20" | 5 625" | 21 45" | 95 25" | " | " | 2 17" | " | 1 17" |
| | | | | | | | | | | | |
| | | | | | | | | | | | |

**Fig. 145—Typical diagram, width clearances 8 ft overall**

tyre sizes becomes more complicated, and it is desirable to set down each case graphically, tabulating resulting figures as shown in Figs. 144 and 145. Conclusions will be drawn from such a diagram as to the best combination of dimensions for frame widths and clearances, as allied with differing combinations of wheel and tyre sizes; whilst at the same time the wheels and offsets and their rim sizes will be standardised at front and rear.

Tyre sizes will, of course, vary in accordance with requirements of load up to the maximum allowable and in this connection there are several regulations to note :

*Tyres :*

(1) Heavy motor cars must be pneumatic tyred, but solid-tyred vehicles first registered on or before January 2, 1933, are exempt until January 1, 1946. Solid tyres may be used for heavy motor cars weighing over 4 tons unladen and used mainly over rough ground or unmade roads, and also on special municipal vehicles.

(2) Every public service vehicle registered on or after February 10, 1931, must be equipped with pneumatic tyres. (P.S.V. C. of F. 514/41/18.)

(3) On each wheel of a heavy motor car registered for the firsttime on or after October 1, 1938, there must be a plate or a direct marking on the wheel showing the maximum nominal section of tyre for which the wheel is designed (398/41/59/1). No tyre of greater section may be fitted (59 (2) ).

*Axle Weights and Total Weights :*

(1) See tabulated schedule (pages 224-225) for weights, overall dimensions, etc.

(2) In the case of a heavy motor car, motor car or trailer, whether laden or unladen, the weight transmitted to any strip of the surface upon which the vehicle rests contained between any two parallel lines drawn 2 ft apart on that surface at right angles to the longitudinal axis of the vehicle shall not exceed 2 tons (398/41/66).

(3) Two wheels are counted as one if their centres of contact with the road are less than 18 in apart (398/41/3 (4) ).

*Dimensions (Side Overhang) :*

(1) No part of a vehicle other than driving mirror and direction indicator (when in use) may extend more than 6 in beyond the outer face of the outer tyre on the rearmost wheel. Vehicles first registered on or before December 1, 1932, are excluded (514/41/13).

(2) No part of a wheel or its fittings may project more than 3½ in beyond the extreme outer face of the tyre when fully inflated. This does not apply to vehicles registered for the first time on or before January 1, 1932 (514/41/19).

Meantime a vehicle-weight assessment will have been made, probably based on similar parts with the necessary adjustment for any

*Table* 12

| | Unit or Part | Weight No. | | Remarks |
|---|---|---|---|---|
| | | Max. | Min. | |
| | Frame * | 600 | | |
| 1 | Engine | 1,600 | | Includes dynamo, air cleaner and starter |
| 2 | Clutch | 65 | | |
| 3 | Control Gear | 10 | | |
| 4 | Radiator | 110 | 95 | 4-row tube type (max.) 4-row Morris (min.) |
| 5 | Front Spring | 203 | 183 | Plain leaf (max.) Toledo Woodhead (min.) |

*Unladen Weight*
Maximum – Tons Cwt Qr Lb
Minimum – Tons Cwt Qr Lb
*Gross Laden Weight*
Maximum – Tons Cwt Qr Lb
Minimum – Tons Cwt Qr Lb

differences in design or dimensions. If such data is not available, then the figures must be calculated and set out to form a table on similar lines to Table 12. The maximum and minimum weights concern savings and actual figures with alternative designs or equipment, and they serve as a useful reference when preparing data for the BM and stress diagrams.

Before commencing the frame-stress figures, it is advisable to form some preliminary idea of spring centres and deflections and, if not already fixed, decide upon a convenient wheelbase, which in view of legal requirements of overall length, particularly with the P.S. vehicles, is fairly well standardised. Useful figures which form a basis for calculation of spring lengths and assist in the fixing of wheelbase are those which give the average total sprung length of frame in relation to wheelbase as from 0·53 to 0·57, whilst the ratio of front spring length to rear spring length varies between 0·82 to 0·85, the front spring of course being the shorter one.

With this data to hand and the fact that the legal overhang at the rear may be 50 per cent of the wheelbase, it is not a difficult matter to decide upon a suitable dimension, always bearing in mind that the front-axle weight should approximate the ratio of 41/45 per cent that of the total axle weights and the rear of course 55 per cent. It is not advisable to exceed this figure, as any substantial increase may

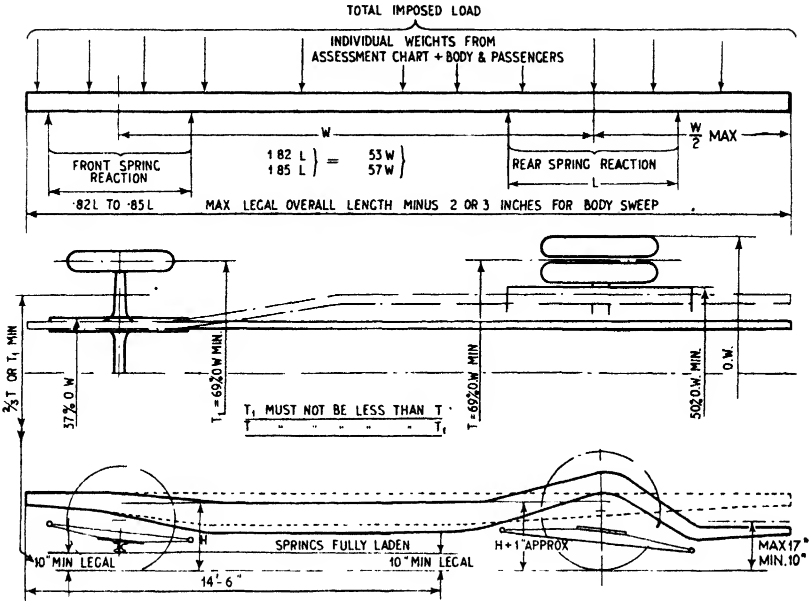



Fig. 146—Chassis weight distribution

affect steering gear problems, and as will be seen later—braking problems.

We therefore can approximate the spring-reaction points, after which it is necessary to set down the weight-assessment figures in scale form on the wheelbase selected and to check axle loadings and weight disposition. The condition is now as shown in Fig. 146 and W may be lengthened or shortened, or alternatively, may be repositioned bodily to so dispose axle weights to their desirable magnitudes.

Having decided these basic dimensions the operation which follows should be to form the bending-moment diagram and stress diagram in the manner described in earlier pages. From the bending-moment diagram the stress will be calculated from the well-known B.M. = fZ expression at many points along the frame side members, and these should be plotted to scale above the combined bending-moment diagram. Should there be any sharp increase or sudden decrease in magnitude, adjustment should be made to flatten out the curve. A public service vehicle frame should be stressed if possible to 6 tons/in$^2$ and never allowed to exceed 10 tons/in$^2$.(See Fig. 147.)

These diagrams are the key to frame development and to the section grading. They also determine the points at which the frame depth may be decreased to the taper ends front and rear. Where it is possible that different wheelbases may be required, diagrams should be produced for all such differences. A certain length of straight portion is formed at the centre of the side rail, since large-wheelbase frames usually have their dies formed in three separate sections, adjustment being made by increasing or decreasing the centre straight section, avoiding any alteration to the front and rear portions.

Having determined the frame section, a side elevation may be produced which includes the springs, usually shown in their fully-laden position. The front and rear wheels should be indicated and drawn at their correct running radius for the estimated tyre loading. In this position the chassis frame should be approximately horizontal and not tail down. This means that when the vehicle is laden it is slightly tail up. General practice usually allows the frame end to be 1 in up when laden, which will allow for the springs settling down and counteract the many other factors which enter into the question.

At this juncture it is desirable to view the frame in plan. The rear width has been decided and the front-wheel track is also known. It is therefore desirable to set down the maximum wheel lock, which will not only show the maximum width frame possible but will allow the elementary steering geometry to be proceeded with in order to ascertain (besides other things) the maximum turning circle, which of course must conform to legal requirements, and the steering-arm and wheel clearance. The steering box can temporarily be located in its vertical position, which will then dispose the length of drop arm required to satisfy any adjustment in ratio between steering box and swivel head. The final position will be settled later.

It will be noted from the stress diagram in Fig. 57 (reproduced for ease of reference in Fig. 147) that the stress is practically constant at the front end at 5 tons/in$^2$ up to a distance behind the spring shackle (point A), therefore having calculated the Z and depth of section for maximum stress the profile grading may be made from the point at which the stress begins to rise and the section made smaller down to the dimensions required for 5 tons/in$^2$. Similarly at the rear the stress begins to fall at point B, but it rises again to C, the magnitude being 7 tons/in$^2$. A similar treatment to that at the front is carried out at the rear.

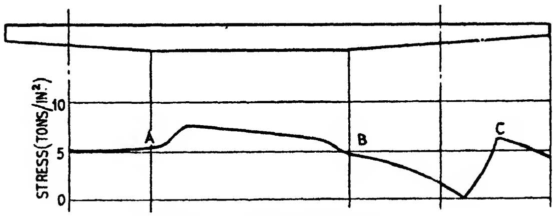



Fig. 147—Frame stress curve

Before the frame can be correctly disposed in relation to the ground, that is in order that it shall be slightly higher at the rear than at the front, and also so that its height from the ground shall not exceed a certain predetermined figure, the road springs or suspension should be calculated. A rough but sufficiently accurate method at this stage is based upon the following formula, with assumed data as tabulated:

| | |
|---|---|
| Weight of vehicle, fully laden | 10 tons 10 cwt 0 qr |
| Front-axle weight laden | 4 tons 10 cwt 0 qr |
| Rear-axle weight laden | 6 tons 0 cwt 0 qr |

Assume 1 ton 10 cwt extra for possible overload. Then if 10 cwt is applied the front-axle loads become

| | |
|---|---|
| Front-axle weight | 5 tons 0 cwt 0 qr |
| Rear-axle weight | 7 tons 0 cwt 0 qr |

Estimated weight of front axle with $10{\cdot}50 \times 20$ in tyres and wheels $= 11$ cwt
Estimated weight of front springs $= 3$ cwt
Total unsprung weight of FA and springs $= 14$ cwt
Sprung weight on FA $=$ 4 tons 6 cwt, that is 2 tons 3 cwt per spring or 4,800 lb (say 5,000 lb).

Estimated weight of rear axle with $9{\cdot}00 \times 20$ in twin tyres and wheels $= 18$ cwt
Estimated weight of rear springs $= 4$ cwt
Total unsprung weight of RA and springs $= 22$ cwt
Sprung weight on RA $=$ 5 tons 18 cwt, that is 2 tons 19 cwt per spring or 6,600 lb (say, 7,000 lb).

In accordance with previous text, satisfactory deflections should be chosen and used as a basis in conjunction with static and dynamic stress to obtain the necessary spring dimensions. To proceed with the example we will work on a front deflection of 3·0 in and a rear deflection of 5·0 in.

*Front Spring :*

Assume eleven plates, $\frac{3}{8}$ in thick, 4 in wide.

Deflection $\delta = \dfrac{WL^3}{41{,}000\,bnt^3}$ where W $=$ weight in tons on spring
L $=$ spring-eye centres
b $=$ breadth of plate (leaf)
n $=$ number of leaves
t $=$ thickness of leaf

$$\text{Therefore } \delta = \frac{2{\cdot}23 \times 50^3}{41{,}000 \times 4 \times 11 \times 0{\cdot}375^3} = 2{\cdot}94 \text{ in}$$

$$\text{Static stress} = \frac{f\delta t}{\left(\frac{L}{2}\right)^2} \text{ tons/in}^2 = $$

$$\frac{2{\cdot}94 \times 13{,}200 \times 0{\cdot}375}{(25)^2} = 23{\cdot}3 \text{ tons/in}^2$$

$$\text{Dynamic stress} = \frac{ft\,(\delta + 2)}{\left(\frac{L}{2}\right)^2} \text{ tons/in}^2 =$$

$$\frac{13{,}200 \times 4{\cdot}94 \times 0{\cdot}375}{(25)^2} = 39{\cdot}1 \text{ tons/in}^2.$$

Note that by the expression $\delta + 2$ is meant that 2 in added to the static deflection is sufficient to cover spring deflection from bump to rebound and this, for a heavy vehicle, is a recognised expression.

These stresses appear to be satisfactory and the deflection is sufficiently approximate to assume that the suggested spring of 11—$\frac{3}{8}$ leaves—4 in wide are acceptable. The recommended dynamic shear stress should not exceed 1,120 lb/in$^2$ and it is therefore desirable to check this stress at the spring eye.

The load per spring eye is 2,500 lb static.

The dynamic load per spring eye is $$\frac{\left(\frac{5{,}000 \times (\delta + 2)}{\delta}\right)}{2} = 4{,}200 \text{ lb.}$$

Sectional area of one plate $= 4$ in $\times \frac{3}{8}$ in $= 1{\cdot}5$ sq in.

Assume two plates under the spring eye

$$\text{Static shear stress} = \frac{\text{Load}}{\text{Area}} = \frac{2{,}500}{2 \times 1{\cdot}5} = 835 \text{ lb/in}^2.$$

$$\text{Dynamic shear stress} = \frac{\text{Load}}{\text{Area}} = \frac{4{,}200}{2 \times 1{\cdot}5} = 1{,}400 \text{ lb/in}^2.$$

This stress for dynamic loading is slightly high and another leaf added under the eye would probably rectify the condition.

$$\text{Dynamic shear stress} = \frac{\text{Load}}{\text{Area}} = \frac{4{,}200}{3 \times 1{\cdot}5} = 935 \text{ lb/in}^2.$$

This appears to be satisfactory for front springs and a similar series of figures should be calculated for the rear springs, which will be found to have, with the foregoing data, a deflection of 5·08 in, a static stress of 29·8 ton/in$^2$ and a dynamic stress of 41·6 tons/in$^2$, with rated load of 7,000 lb at eleven plates $\frac{7}{16}$ in thick and 4 in wide.

We are now at the position where the frame may be mounted and laid on its wheels, or in other words we have arrived at a stage which corresponds to Fig. 146. The next stage concerns adjustment of frame profile to suit rear-axle clearances and to decide upon ground clearances for the front axle, which will again necessarily be controlled by regulations. Bump clearance between front axle and engine sump will be set and buffer dimensions settled. The frame at the front will be slightly rearranged, retaining the calculated section through the requirements of the rear. Floor heights, rear step height and axle bump will necessitate an upswept rear and a rapid drop at the extreme rear end. The profile is modified as shown full in Fig. 146.

Here we will briefly recall the regulations as affecting the design at this stage: There were, prior to October 1, 1936, alternative requirements in the Statutory Orders with which it was necessary to comply, two of which related to track widths and spring-centre distances. In making reference to these conditions at this date it is desired to eliminate the confusion which exists to this day with regard to the interpretation of these two specific requirements which no longer apply to vehicles registered after October, 1936.

## STABILITY OF PUBLIC SERVICE VEHICLES

It is clearly stated in Statutory Rules and Orders 1941 C of F Regulations No. 7 under the heading of Stability of Public Service Vehicles:

(a) Double-deck vehicles registered after July 1, 1931, must pass a 28° stability or tilt test with the upper deck laden, without overturning.
(b) Single-deck vehicles registered after October 1, 1936, similarly must pass a 35° stability or tilt test with full load without overturning.
(c) Single-deck vehicles registered after July 1, 1931, and before December 1, 1932, must pass a 28° stability test.

The obsolete stability requirements were briefly as follows:—

(d) Tracks: The front and rear tracks in either case were required to be not less than 69 per cent of the overall width of the vehicle, and the front track was not allowed to be less than the rear track. Also:—
(e) Springs: The old regulations specifically laid down that " rear springs must be attached to or bear upon the rear axle casing as near as possible to the road wheels ". Also that the width outside of the rear springs was required to be not less than 50 per cent of the overall width of the vehicle and the width over the front springs had to be not less than 37 per cent of the overall width.

It will be recognised that some such requirements as outlined in (d) and (e) above were obviously desirable to preserve some measure of standardised stability, although it was felt that the same degree of stability could be attained by a much simpler form of regulation. Accordingly it was agreed that the two simple clauses quoted under (a) and (b) above should be introduced as the *sole requirements* for new vehicles registered after the dates mentioned.

The clauses under (d) and (e) are retained in the existing regulations in connection with single-deck vehicles registered between the two inclusive dates December 1, 1932, and October 1, 1936, in which case there is a choice of the following alternatives:

(a) Must pass a stability test of 35° or otherwise the vehicles must comply with all the following requirements from (b) to (f).
(b) Pass a stability test of 28°, instead of the now prescribed 35°.
(c) Front and rear tracks in either case be not less than 69 per cent of the overall width of the vehicle.
(d) Front track must not be less than the rear.
(e) Rear springs must be attached to or bear upon the rear axle or casing as near as possible to the road wheels.

(f) Width outside to outside of the rear springs to be not less than 50 per cent of overall width of vehicle.
Width over front springs to be not less than 37 per cent of overall width of vehicle.

In the case of vehicles registered on or after December 1, 1932, transverse springs if fitted at the rear of the vehicle shall be fitted only as supplemental to longitudinal springs and in every case where transverse springs are fitted to a vehicle the system of springing shall be so designed that there is no excessive body sway (514/41/8).

*Turning Circle :* Every p.s.v. except those first registered on or before January 1, 1932, must be capable of turning in a circle not exceeding 60 ft (measured at the extreme outer edge of the wheel track), but if the overall length exceeds 26 ft the maximum turning circle may be 66 ft (514/41/9).

*Step Height :* The top of the tread of the lowest step for any entrance or exit to a vehicle, other than an emergency exit, shall not be more than 17 in or less than 10 in above the ground when the vehicle is empty. All steps shall be fitted with non-slip treads. Fixed steps shall not be less than 9 in wide and shall in no case project laterally beyond the body of the vehicle unless they are so protected by the front wings that they are not liable to injure pedestrians (514/41/28).

*Ground Clearance :* There must be at least 10 in clearance above ground level extending to at least one-third of the front-wheel track from each side of the centre line of the vehicle and rearwards from the foremost part of the chassis (excluding starting handle) up to a distance of 14 ft 6 in in the case of a four-wheeler or 13 ft for a vehicle with more than four wheels. If a rear-driving axle intervenes, the clearances must extend up to but not include such axle. These requirements do not apply to vehicles registered for the first time before January 1, 1933 (514/41/10).

It is inadvisable to mount either front or rear road springs on the sides of the chassis side members, as the resulting twist is difficult to eliminate, and the plan view will be modified in order that the springs will be underneath the frame section.

The adjustment is shown chain-dotted in Fig. 146. In consequence of this frame arrangement a road spring of either flat or slightly negative camber will be indicated. It has been stated previously that this type of spring provides more adequate location for the rear axle than does the highly-cambered type.

Springs may now be worked out and designed in detail. A typical design of spring layout is illustrated in Fig. 148. The cross-bracing members must be next arranged and, after having decided the type to be adopted—whether tubular, channel, or cruciform—a rigid member should be placed at each spring attachment point, the attachment bolts if possible being arranged to pick up the spring bracket holes. The importance of this location is manifest insomuch as these eight

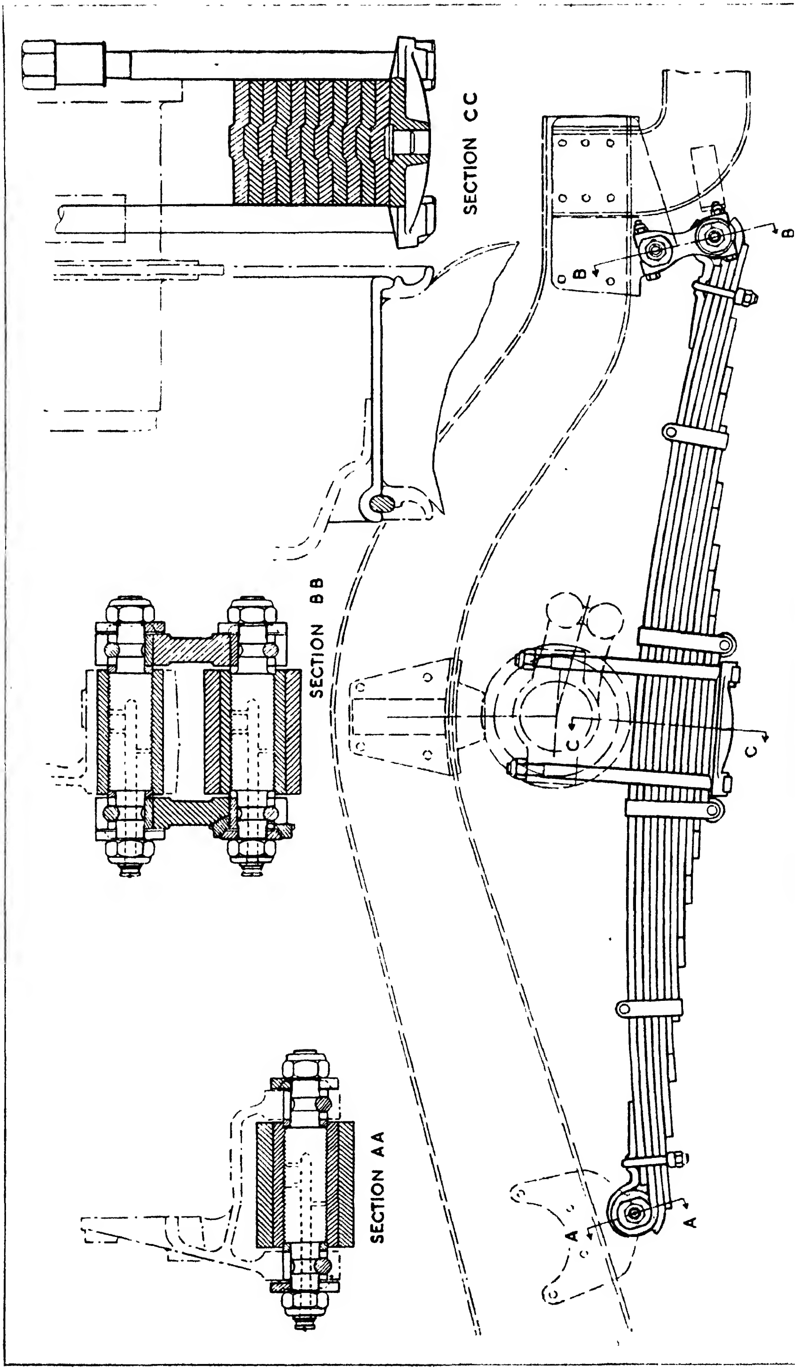



Fig. 148—Typical passenger-vehicle rear spring

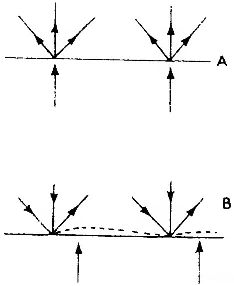
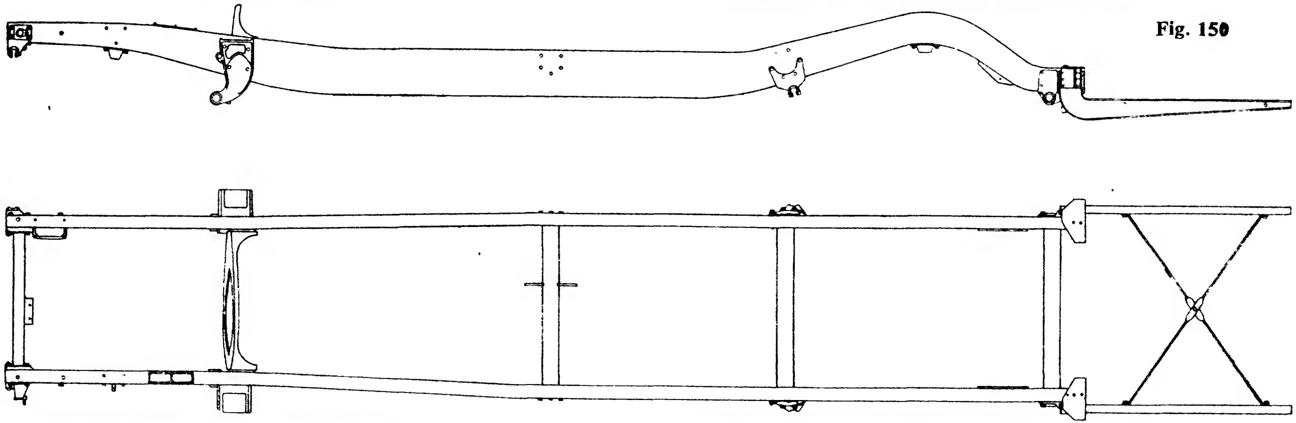


Fig. 149

points are those at which all road-shock loads are transferred to the frame, and therefore the resulting stresses must be distributed through several members and absorbed as early as possible. This is only possible if they are delivered at some focal point. For example, Fig. 149 illustrates the distribution which, in case B, applies the greatest load on the longitudinal member under the worst conditions, those of bending, whilst the effectiveness of the focal members is greatly reduced. Case A shows effective distribution with all members taking their places as either ties or struts and absorbing a fair share of the stresses.

Particular attention must be given to the front cross-member design, as frame rigidity at this point is absolutely essential. Any intermediate members will be spaced for convenience, exemplified by the centre cross bracing, the location of which is usually

Fig. 150



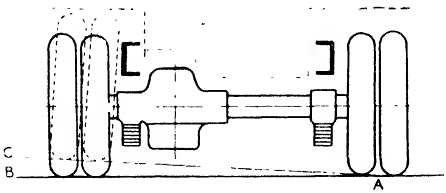



Fig. 151

dictated by the requirements of the propeller shaft with regard to length and its centre bearing.

An example of light but rigid frame design is typified in Fig. 150.

It is now possible to make further progress in the unit layout and a start may be made by fixing the main line of transmission. It has not been possible to advance to this stage previously as it is necessary to fix a suitable transverse position for the rear-axle centre and that could not be done before the rear-frame width had been decided. However, it is appreciated that the offset drive presents no insuperable difficulty and it will be necessary to set out a cross-section view at the rear-axle centre to obtain maximum axle rise and clearances in all directions (see Fig. 151).

It is preferable that the axle " pot " be off centre and positioned under the frame as far as possible in order to provide an unobstructed gangway. This objective cannot be entirely achieved due to the enforced restriction on frame width and bump clearance; thus in consequence the wheel-arch seat tread must be built up to minimise the obstruction. However, when the chassis designer has pared the clearances down to the minimum acceptable to the body designer, the transmission line in plan view may be considered. The effect of angular misalignment of joints upon rotational speed will have been seen in a previous chapter, which, if permitted, sets up a violent period in the shaft and seriously overstresses not only propeller-shaft components, but the vibrations are transmitted throughout the chassis and build up into a most unpleasant noise. Where angular misalignment is absolutely unavoidable, the practice of fitting a vibration damper may be resorted to.

It should be borne in mind during all stages of transmission layout that most probably some section of the propeller shaft will of necessity possess angular disposition in two planes. Endeavour must therefore be made to straighten up the line as much as possible, in both plan and elevation, from gear box to axle. Moreover the axis of axle is variable in elevation on account of spring distortion when braking; whilst, due to vertical movement and normal spring deflection, the shaft rear end joint angle is constantly varying.

Reverting to the plan view, the rear-axle worm or bevel axis will be parallel to the chassis centre line, and therefore it is reasonable to assume that the longitudinal axis of the engine should also be parallel to this chassis centre line, although not necessarily *on* that line. Requirements of pedal spacing, controls, etc., almost invariably necessitate an offset engine—offset to the near side—and whilst this has certain advantages in reducing the propeller-shaft angle, it is often difficult to obtain sufficient accessibility of engine auxiliaries, nearly always mounted on the engine nearside to ensure a clean cab interior.

A front-end layout will be necessary at this stage, setting down frame sides, engine auxiliaries, pedals and steering, and wheels on full lock in both directions. It is not possible to do this adequately without an elevation (side and front) which will include the front-axle cross section in normal and bump positions, allowing the desired clearance under the engine sump. Preliminary layout to assess the longitudinal location of the engine will include temporary radiator and starting-handle positions, allowing for a radiator shell if fitted, but always having in hand approximately ½ in on overall length. This layout requires all the skill of the designer as there are many things to bear in mind, as it forms the basis of the major detail schemes which will be produced at a later stage. Usually there is limited space between the radiator front and the cab rear panel and, as pedal position and seating are very interdependent, it will be appreciated that pedal sweep as affecting radiator position must have a great bearing upon the initial layout. However, these items will be discussed later, but it is necessary to mention a few of the schemes required which bear upon engine location, in order not to minimise the importance of skilled layout at this stage of initial design of transmission.

Although the engine may be parallel with the longitudinal axis of the chassis, it may be, if considered desirable, set at an angle in elevation. Such a practice produces the advantage of reduced transmission angle, and it is very often found possible to provide a straight-line series of shafts, particularly if the rear-axle casing be set on the rear springs so that the worm axis is in line with that of the engine. It will be appreciated that a similar arrangement could be made in plan view,

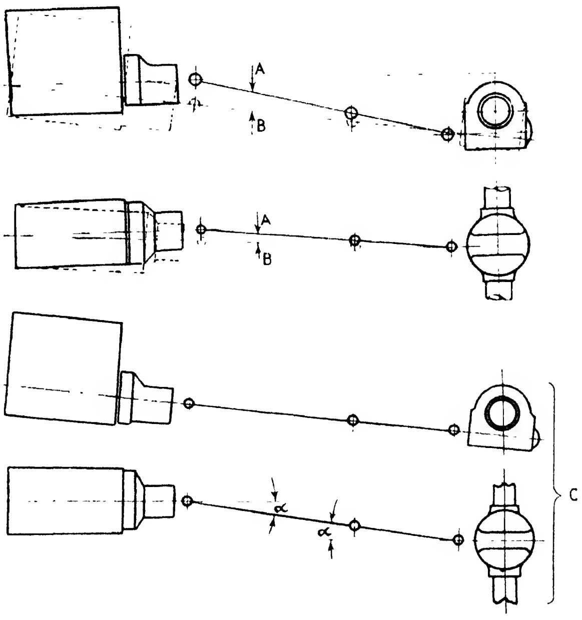



**Fig. 152—Transmission-line alternatives**

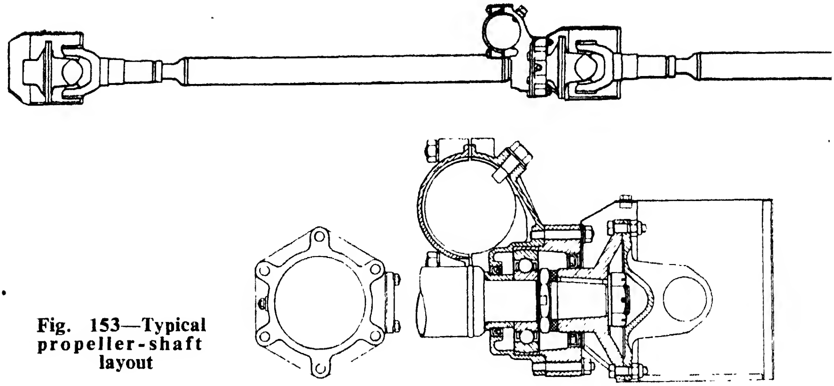


**Fig. 153—Typical propeller-shaft layout**

when the combined angles will be still further reduced. However, this angle in plan view is adopted only in few examples, probably on account of the fact that it possesses the disadvantage of machining and assembly of engine-mounting components in two separate planes, and is therefore costly against its small advantages in shaft angle and cab space. Reference to Fig. 152 will make this clear.

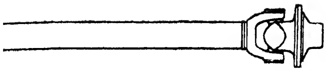
The best compromise is possibly the parallel-plan axis and angular-elevation setting for the engine with the rear axle inclined on the springs as shown in Fig. 152c.

## THE RADIATOR

Having fixed the transmission line, perhaps the most useful unit to investigate next is the radiator and fan cowl if such is considered necessary. Incidentally a close-fitting cowl considerably assists in radiator cooling and if there is no relative motion between engine and radiator, by which is meant that either both units are mounted solidly in the frame or alternatively the radiator is mounted on the engine, then the close-fitting cowl is a recommended inclusion.

It is not intended to discuss the actual detail design of the radiator, as cooling depends upon so many features which it would be impossible to cover in all stages in this chapter. Such items as the type of tube block, design and capacity of fan, cowl clearance around fan blades, to say nothing of the engine characteristics of heat to water in h.p. and water speed, must be considered.

Additionally a great deal depends upon the amount of available space, which controls the radiator frontal area, for it must be appreciated that later the driver's vision line will arise for consideration, and a too-tall radiator will incur an impedance of this most important feature.



It is, however, very desirable to have a good rise of inlet pipe from engine to radiator top tank and a compromise in height must be arranged. As a reasonably practical procedure, it is usual to assume a frontal area compatible with prevailing dimensions at the chassis front end and, if the fan has not already been designed, to calculate the required air flow which will produce the required temperature drop in conjunction with the characteristics of the radiator tube adopted. As an example: Assume a petrol engine develops 100 b.h.p., its heat to water 100 h.p., a temperature drop of say 100° F and a radiator frontal area of say 4 sq. ft. Then the h.p. to be dissipated per sq. ft of matrix per degree F temperature difference $= \frac{100 \text{ h.p.}}{100° \text{ F} \times 4} = 0{\cdot}25$ h.p. This is at a matrix face velocity of 20·5 ft/sec or 14 m/h and assumes a constant water flow of 20 gal/min per foot width of matrix. The curves in Fig. 154 show the comparative capabilities of heat dissipation for 8 and 9 gills per in, whilst also illustrating the corresponding static pressure drop against matrix face velocity; both sets of curves assume a five-row stack.

The quantity of air required to pass through the tube stack $= 20{\cdot}5 \times 60 \times 4 = 4{,}920$ ft³/min. Consider the free air speed at say a vehicle velocity of 10 m/h, which is approximately 15 ft/sec. Then the volume of air at that speed will be $15 \times 60 \times 4 = 3{,}590$ ft³/min. The natural flow of air through the block will be $3{,}590 \times 0{\cdot}4$ or 1,436 ft³/min as 0·4 is the approximate radiator resistance, the fan being left to deal with the difference between the air required and the natural flow, in this case $4{,}920 - 1{,}436 = 3{,}484$ ft³/min.

Reverting to the frontal area and assuming that the fan can deal with this volume of air, then 4 sq.ft is adequate and the boundary dimensions will of course be arranged so that the radiator top tank is not too high to impair vision

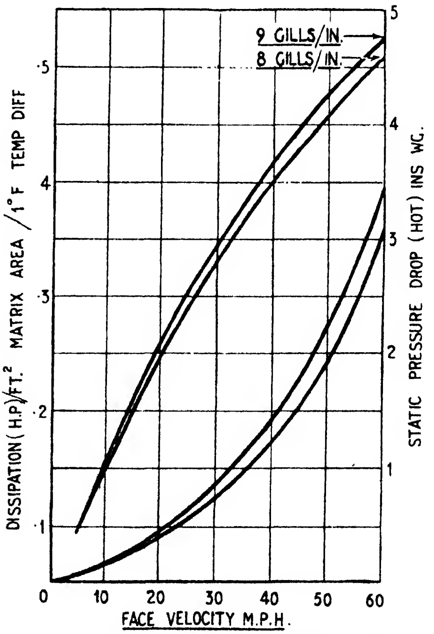



**Fig. 154—Radiator tube cooling curves**

and the block not too wide to sit between the frame side members.

The radiator position longitudinally is virtually fixed by the requirements of vehicle overall length, and thus the engine and fan location depends upon what is considered to be the *optimum* distance of fan from radiator, and in passing it should be realised that this distance is important insomuch as if the fan is too close to the radiator matrix then the efficiency of the combination is considerably reduced, as also is the case if the fan is too far distant from the matrix.

With a petrol engine it is advisable to fit a fan cowl, which somewhat minimises the effects of poor fan location. This optimum position depends upon many factors, such as the type of block tube and gill, the number of tube rows and the characteristics of the fan itself. Sufficient data should be at hand concerning the cooling capabilities of the combination of radiator cowl and fan to enable the designer to locate the fan with confidence.

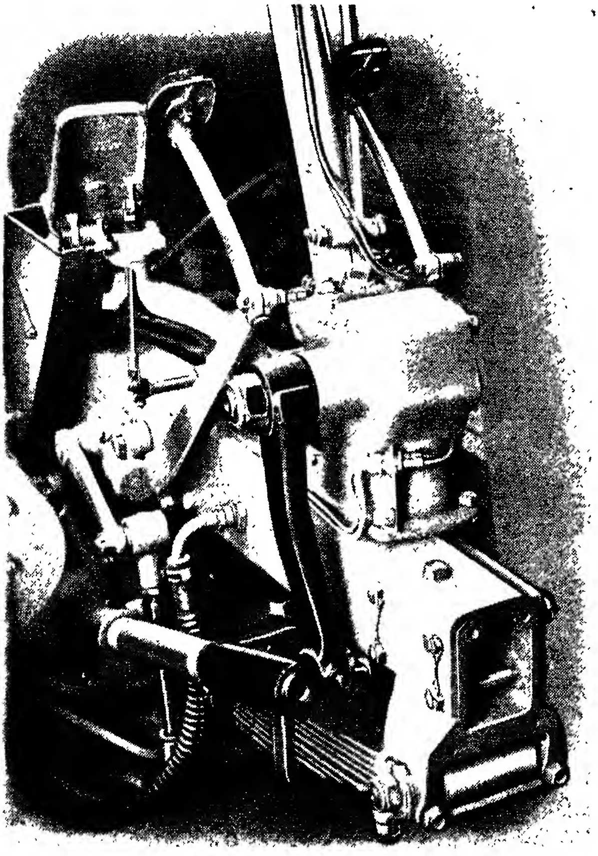


Fig. 155—Pedal spacing

Moreover, since the cooling water is hottest at the radiator inlet, the fan should be arranged as near to the underside of the top tank as is consistent with reasonable cowl design, always taking care that the cowl and metal work are arranged so that air currents are not allowed to enter the bonnet by way of the gap between the matrix and side standards, and thus cause eddies which are liable to counteract the work of the fan by virtue of recirculation.

Fig. 166

Transversely the radiator position is governed by several features, which, combined, serve to fix the dimensions. Firstly, there is its relation to the fan. It is of no avail to have the fan in the top righthand corner, although it is not always possible to arrange it in the absolute centre of the matrix. Secondly, there is the pedal sweep : the radiator must of course be clear of this in a forward-control layout. Finally, account must be taken of the cab-and-bonnet symmetry, before layout of the steering gear and pedals is commenced, which will be the next set of units for consideration.

## PEDAL SPACING

Pedal spacing and its relation to seating is not an easy matter to arrange to suit the differing statures of driver, particularly in the p.s.v. or freight vehicle. It is a much more arduous task than for the private car, wherein the driver occupies a more reclining position, whilst steering box and engine are remote, permitting more freedom in design. However, it is usual to adopt the I.A.E. standard of pedal spacing where possible, that is, from near to offside, clutch, steering column, brake and accelerator in that sequence. With a p.s.v. design the control motion of the pedals is more in the vertical plane than the horizontal plane, so that it is important to note the relative positions of the steering drop arm and the downward sweep of the brake pedal and to arrange that under no circumstances can a foul be experienced when on left-hand full lock (see Fig. 155).

Pedal travel, in the cases of clutch and brake, should not be excessive and general practice favours a figure of approximately 7 in or 8 in travel. Both these amounts will, of course, have been decided in the stages of unit design of clutch and brakes, so that the chassis designer is confronted with the main problem of installation by providing a suitable mounting on the frame (Fig. 156). Care must be exercised in the rod layout, as any relative motion between the clutch lever and the pedal in the case of a flexibly-mounted engine in unit construction with gearbox, or between the brake pedal and brake cam levers, will have unpleasant results in ease of control of the vehicle.

In previous text the road-spring geometry has been set down, and the virtual point of oscillation is of utmost importance in brake layout. A preliminary rod run for clutch and brake control will be laid out as a guide to the possible location of other components. The brake-rod run will be referred to later, but with regard to the clutch control the more salient points concern maintenance and points of adjustment for clutch-liner wear, which, although they must be accessible, should not invite unnecessary attention. The attachment of the rod to the withdrawal lever must be in accordance with the requirements of lever movement, that is, if movement is in one plane only as in the drop-type lever, a fork-and-eye connection is all that is necessary, but if a " bloater "-type withdrawal is fitted a ball joint is indicated.

However, all these details will receive attention when the subsidiary schemes are dealt with after the chassis layout has been completed.

## DRIVING POSITION

A most important point to bear in mind in the initial stage concerns the position of driver's seat, and it is a matter of experience to obtain the required pedal movements whilst at the same time providing a comfortable sitting position. It is essential to remember that the ball of the foot is used to operate the pedals and in the case of the accelerator pedal, particularly if the fixed pad or even roller type is adopted, the ankle and foot must be in a position of repose when operating. It is desirable, therefore, to fix the cab-floor height and set down the offside wheel and tyre on full right-hand lock and maximum articulation, as wheel arch will control pedal spacing demensions on the offside, whilst the engine bonnet will control them on the other or nearside.

After fixing the height of the driver's seat, so that for an agreed eye level the line of vision is good over the radiator top, looking from an angle towards the nearside, the pedal-pad position may be decided for normal position and at full sweep. There are no hard-and-fast rules or methods in laying out the pedals, as the procedure rests upon compromise between space available with wheel on both full locks, steering-box and drop-arm position in both locks, engine-bonnet position, and the relation of pedal pads to seat and floor positions.

A layout which has been agreed to be satisfactory and which provides the minimum of fatigue over long operating periods is set out in Fig. 157. It will be noted that the path of the pedals is not too steep, in which case the driver may be seated in a slightly reclining position, whilst the travel is such that the knee does not straighten out, leaving a certain amount of available movement and strength should it be required. The spacing is also such that the feet occupy a position as normal as possible in the seated position, the accelerator pedal being inclined with the toe tip slightly outwards. The driver must be able to operate all pedals to their full extent of movement without the necessity of leaning forward, and the seat should be adjustable in all directions, so that the feet and hands should occupy a natural and easy position in relation to the body position when simultaneous movement of both is necessary (see also Fig. 155).

With regard to the steering-wheel position, this must be arranged to allow ample clearance between it and the seat, in order that access to the seat is unimpeded.

The pedal-linkage schemes follow and that for the clutch should include the minimum number of relays, whilst due attention should be given to the clutch-operating lever regarding its movement to provide the correct joints or forks at the connecting points.

## BRAKE LAYOUTS

With regard to the brake hook-up, this also is a matter for careful consideration, particularly in the case of the heavy vehicle, upon which servo assistance is required. Many proprietary components are required with this system—usually vacuum type in this country—and the most economical location for these fittings is important in so much as the pipe work forming the circuit must be as short as possible and be free from sharp bends to avoid time lag in operation. It is therefore desirable wherever possible that the servo and vacuum reservoir should be in close proximity, although it is not always economical to mount both on the same side of the vehicle.

The brake rod connecting the pedal to the servo must not upon any consideration be out of line with the servo link for more than a degree or two, a remark which equally applies to the rod from servo to cross shaft. The length of the rod between pedal and servo must have no adjustment other than for its initial setting, as the available travel of the rod must always be equal to the total travel of the servo. Fig. 157 will make this clear. A typical layout showing what is designated the triple system is shown in Fig. 158.

In the case of an oil-engined vehicle, the exhauster is usually engine driven, whilst vacuum is obtained through the induction manifold with a petrol engine. Such a system provides for two cylinders mounted on the front axle, a servo and a vacuum reservoir. The master cylinder controls the two smaller ones, which are coupled to the servo on the vacuum side of its piston by suction pipes, and

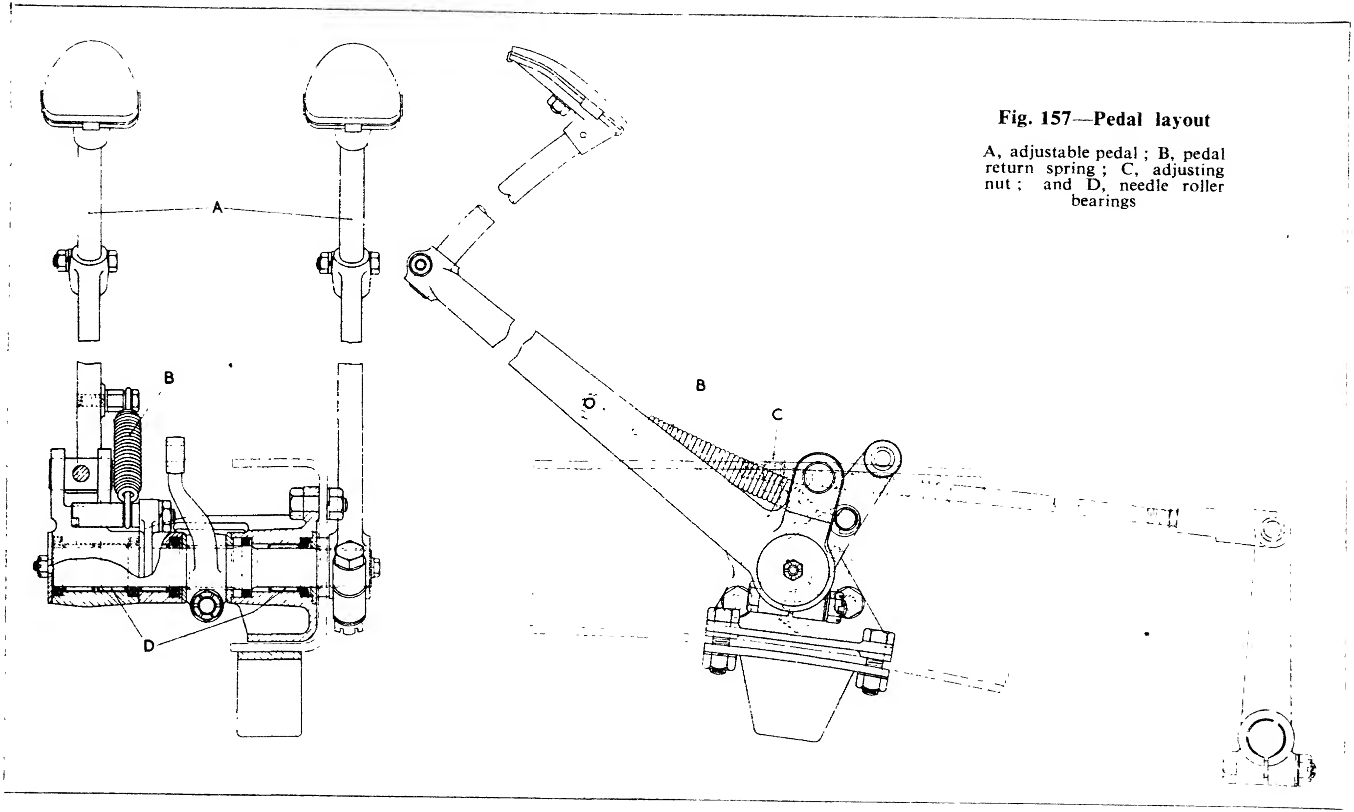



**Fig. 157—Pedal layout**

A, adjustable pedal ; B, pedal return spring ; C, adjusting nut ; and D, needle roller bearings

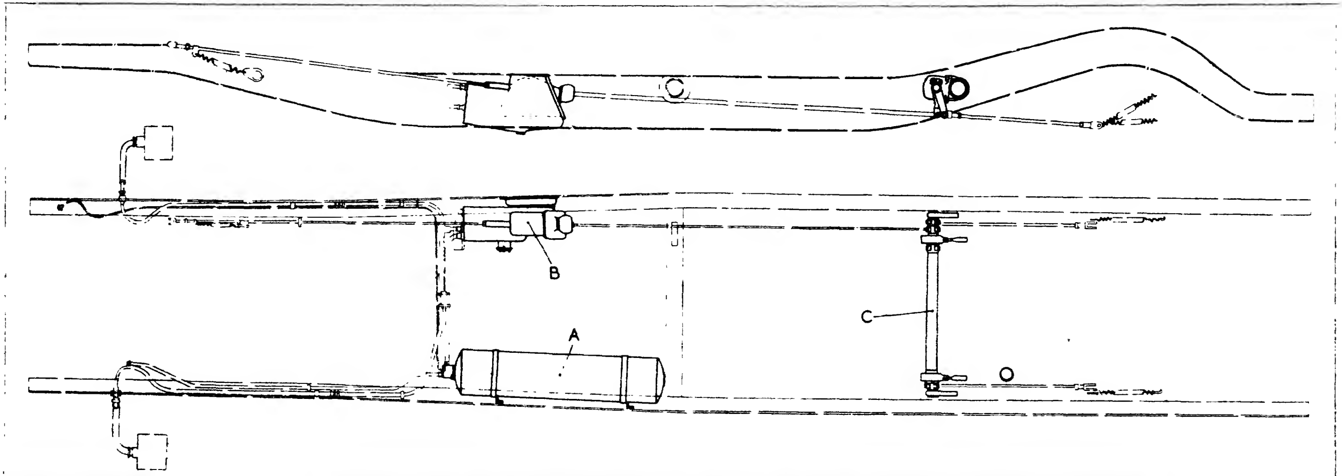



Fig. 158—Triple servo-brake system

therefore whatever degree of vacuum exists in the master also exists in the supplementary cylinders. At the same time it will be appreciated that the proportion of front braking is always relative to that of the rear irrespective of the amount of pedal depression owing to the reaction principle of the distributor valve. The master cylinder applies its effort to the rear wheels only, there being no mechanical connection between front and rear brakes.

Air-brake control is much more widely adopted overseas than in this country, although it affords a high efficiency and permits a stopping potentiality comparable with the private car, with little more physical effort than that required to operate the accelerator pedal. It is inherently flexible and possesses adaptability to remote control and thus it is of value in trailer operation. The calculations differ somewhat from those necessary for ordinary layout,

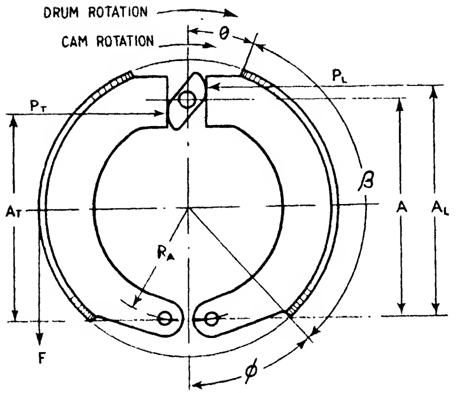



**Fig. 159—Power-factor diagram**

although the basic principles of vehicle braking, of course, still obtain. The stages through which these calculations pass may be of interest and in this connection, and in order to clarify the situation, several assumptions will be made.

It will be assumed that the two anchor pin-type shoes are fitted and operated by a rigid or non-floating cam, also that the liner thickness of $\frac{1}{2}$ in is to be worn fully without the use of packing washers at the shoe-cam face, by an effective cam radius of 1 in (that is, 13 per cent of the drum radius). The fixed-cam type of operation requires that the front cams must rotate in the same direction as that of the drums in order that any tendency towards brake snatch will be eliminated. This phenomenon, incidentally, is probably promoted by the rotational acceleration of the axle bed under braking loads. The inertia of brake parts on rear axle, however, is much lower and thus brakes are not so prone to snatch, so that the rear cams may preferably rotate in the opposite direction to that of the rear drums.

This difference in directional rotation of cams front and rear introduces the necessity for certain power factors, based on the dimensions of the cams and the shoes, since the braking forces at the drum surfaces will differ for a given camshaft torque, as between front and rear drums. An approximate method of calculating these factors, which give results sufficiently accurate to cover practical application, is by the following formulae, and where notation is as Figs. 159 and 160:

(1) Cam rotation similar direction front and rear:

Leading shoe—

$$\frac{F_L}{P_L} = \frac{A_L\,\mu\,(\cos\theta + \cos\phi)}{AR_A - \mu\,[(\cos\theta + \cos\phi)\,R + CR_A]}$$

Trailing shoe—

$$\frac{F_T}{P_T} = \frac{A_T\,\mu\,(\cos\theta + \cos\phi)}{AR_A + \mu\,[(\cos\theta + \cos\phi)\,R + CR_A]}$$

(2) For cam rotations in opposite directions to those of drum:

Factors for leading and trailing shoes are calculated from the same formulae, but with notation as in Fig. 160.

In the foregoing expression $A = \frac{\beta - \theta}{2} + \frac{\sin\theta \cos\theta + \sin\phi \cos\phi}{2}$

and $C = \frac{[(\cos\theta + \cos\phi)(\cos\theta - \cos\phi)]}{2}$.

In the above $\beta$ and $\theta$ are in radians.

If the vehicle were always fully laden and braked to its maximum deceleration then the proportion of brake power would be as the dynamic-load distribution at maximum deceleration, but this is rarely the case, whilst the possibility of varying road conditions must not be overlooked. Over many readings taken on test the average deceleration on service is in the region of 8–9 ft/sec/sec and, assuming the gross vehicle weight to be 11 tons with a front-axle weight of 4¼ tons, then the transference of load at this deceleration, assuming a ratio of height of centre of gravity to wheelbase as 0·3, would be 0·82 tons, making a total of 5·07 tons or 46 per cent of the total weight, which would be the braking-power distribution.

This is considered too high, however, for icy conditions of road surface, where the coefficient of adhesion is much lower than the usual 0·7 used for normal calculations. In point of fact $\mu$ for icy roads is often as low as 0·110 and if this figure be used to calculate the required braking proportion on each axle it will be found that the proportions are nearer the proportions of the static-axle weights and that a percentage of 41·5 approximately is indicated. Under such conditions, at a vehicle deceleration of 8 ft/sec/sec upon an icy road, stability would be preserved.

There are, of course, other considerations of a less serious nature, but nevertheless important, one such factor being the question of rate of liner wear. It is desirable in the interests of service that both sets of liners should wear out at the same time. Front liners wear out less rapidly than those on the rear shoes, probably due to cooler running conditions, and thus the intensity of liner pressure may be increased. The usual increase is 25 per cent more load per unit area, and from this factor the relative widths of shoes may be determined. If, for example, the respective widths of front and rear shoes were 4 in and 7 in then the power distribution would be 41·7 front, which is in approximate agreement with that required for dynamic conditions upon icy roads. It is usually submitted that a deceleration of $\frac{g}{2}$ is sufficient as higher figures are dangerous if standing passengers are carried, and therefore in

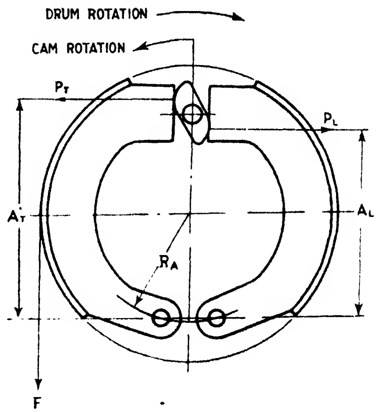



**Fig. 160—Another power-factor diagram**

working out the details of the brakes it is necessary first to ascertain the front-axle deceleration and the torque at the brake camshafts; similarly the rear-camshaft torques.

The air-pressure cylinders are, for calculation purposes, considered to be 90 per cent efficient, whilst the air pressure rarely exceeds 80 lb/in$^2$. Selections will be made of a suitable cylinder size which will give the desired cylinder thrusts through the camshaft leverage to provide the required camshaft torques. The mounting of these components is usually on the frame at the rear and on the axle at the front, and in both cases means of adjustment for lining wear must be provided between the cylinders and the camshafts.

It is necessary at this stage to consider the handbrake in relation to shoe movement and lever travel before the correct cylinder travel can be ascertained. Drum expansion must not be overlooked. It may be occasioned by heat or when under load, and in this connection it is important to distinguish between shoe-centre movement and shoe-centre clearance. Normal distortion of the drums under load is approximately equal to 0·010 shoe-centre movement and, allowing for ovality and necessary clearance, it is not advisable to adjust the brakes closer than say 0·014 shoe-centre movement. If drum expansion due to heat be now considered, which amounts generally to approximately 0·020 in on radius, it will be seen that 0·034 in shoe-centre movement is necessary even before any liner wear exists.

Lining wear for the purpose of calculation is assumed to be 0·011 in per 1,000 miles of city service, so that we now reach a figure for shoe-centre movement of 0·045 in. It should be explained that the average operational temperature of drums on city service work is about 240° F, which represents 0·020 in diameter on expansion, the 0·020 in radius previously mentioned representing expansion on a long gradient.

The allowances for shoe-centre movement are somewhat different when Clayton R.P. automatic adjusters are fitted. These adjusters are designed for two shoe movements, one with the inside link at 0·033 in and the other with the outside link at 0·028 in shoe-centre movement, so that if adjusters are fitted the shoe-centre movement will be 0·033 + 0·020 = 0·053 in (see Fig. 162).

Reverting to the question of handbrake, in order to assess the overall ratio required, it is advisable to allow a little lost movement of the lever due to spring in the system, and as the average hand-lever movement is about 20 in at the grip, and say 2·5 in is absorbed by linkage stretch, etc., then 17·5 in is available for use, although it is not good design to assume that all of this travel should be used. A round figure adopted for heavy-vehicle brakes is 12·5 in and thus total leverage to cam tip is $\dfrac{12{\cdot}5}{0{\cdot}045 \times 2}$ or 136 to 1 for non-automatic adjusted brakes and 118 for automatic R.P. adjusters.

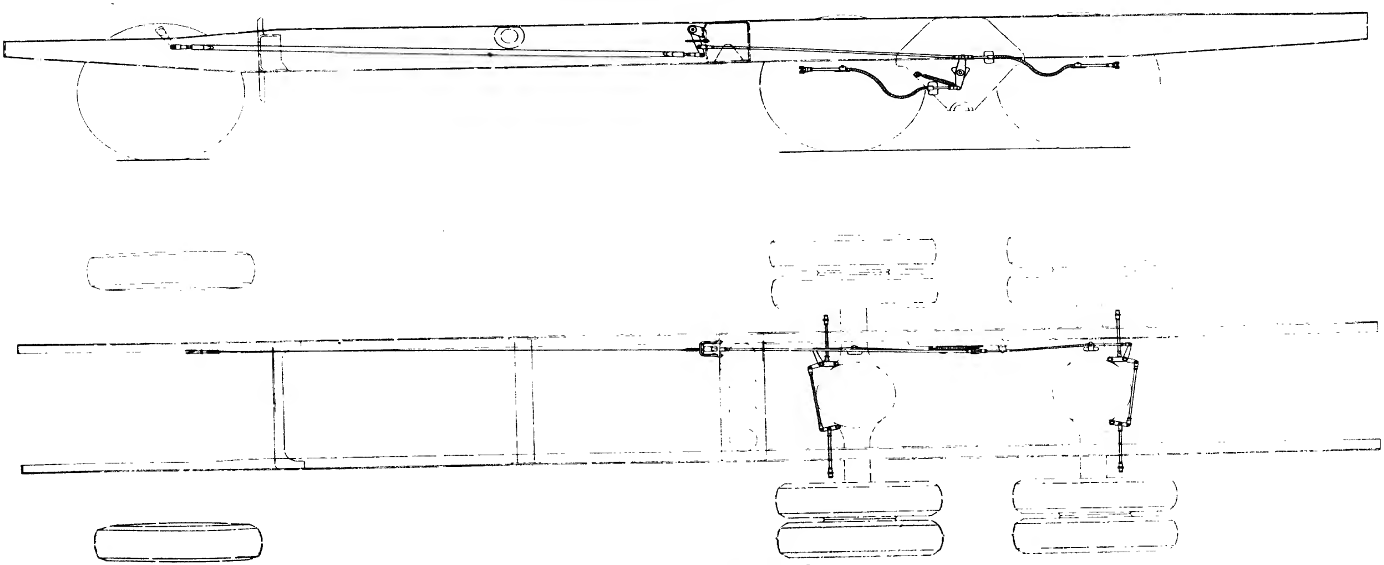


Fig. 161—Typical hand-brake hook-up

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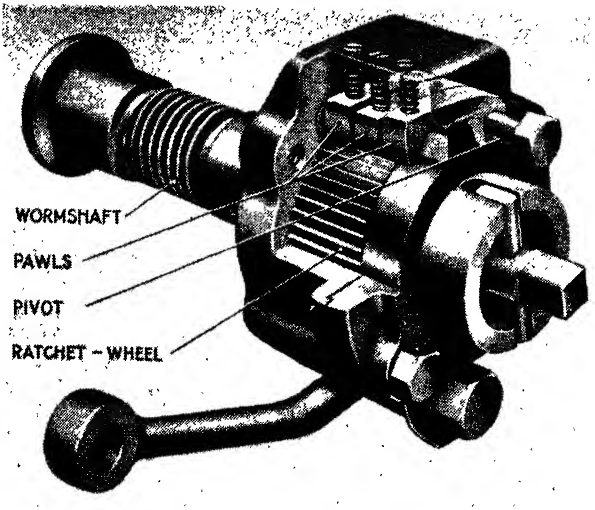


Fig. 162—Brake-shoe adjuster

Incidentally, the handbrake lever should be located so that the driver can exert his maximum effort without the lever becoming too close to his body or the lever grip being in such a position as to make full purchase impossible. The travel of the rear cylinder is arranged adequately to cover the handbrake range and it should be noted with care that the travel must be such that if adjustment for wear has been neglected to the point at which the footbrake begins to be affected, the handbrake will still function. Cylinder travel in the case of non-automatic brakes is $\frac{17 \cdot 5 \times 8 \times 1 \cdot 4}{136} = 1 \cdot 44$ (the figure of 1·4 in this expression represents the leverage multiplication of cylinder thrust and 8 is the assumed length of lever on the brake camshaft) and 1·66 for automatic-adjusted shoes. If the front-brake camshaft lever be assumed at 6·5 in then the front-cylinder travel will be $\frac{1 \cdot 44 \times 6 \cdot 5}{1 \cdot 4 \times 8} = 0 \cdot 835$ for non-automatic and $\frac{1 \cdot 66 \times 6 \cdot 5}{1 \cdot 4 \times 8} = 0 \cdot 965$ for automatic-adjusted brakes.

Deceleration for 100 lb hand-brake effort at 80 per cent efficiency is therefore $\frac{PLFD}{2eW} = \delta$, the 100 lb being the average load which a driver can exert on the lever without undue difficulty, and where

P = effort on hand lever at per cent efficiency
L = leverage in system
F = power factor
D = drum diameter
g = gravity
e = effective wheel diameter
W = vehicle weight (lb).

The maximum demand on the compressor is assessed by taking

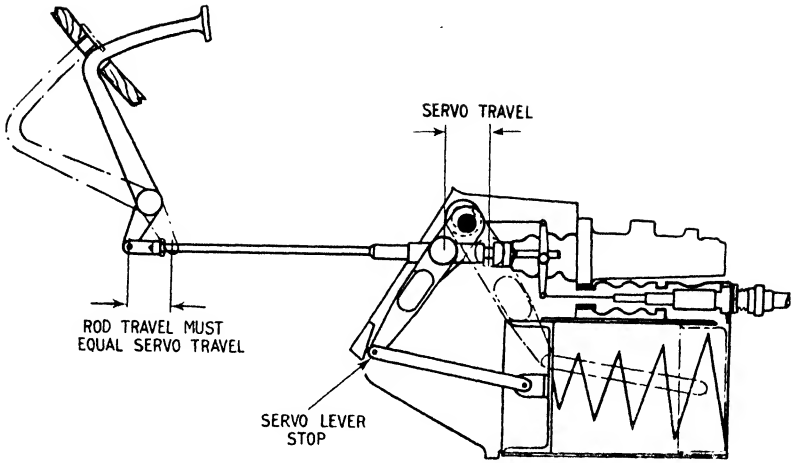


**Fig. 163—Rod and piston travel**

these cylinder travels and calculating the volume of air consumed for a maximum brake application; that is for two front and two rear cylinders, and the volume of air in the pipe line, the amount of air consumed per application is $\frac{V \times (P + p)}{p}$ in$^3$ per application where

V = total volume of air in cylinders and pipe corrected to atmospheric pressure

P = system or line pressure

p = atmospheric pressure (14·7 lb/in$^2$).

The air supply is worst when the vehicle is being driven in top gear, and it is necessary to ascertain whether the selected compressor is capable of maintaining sufficient air supply under such conditions. Data regarding capacity will, of course, be known to the designer relating to compressor output at specified speeds. Most single-stage compressors operate at approximately 1,000 r/min maximum and their nominal capacity is rated at this figure. From a vehicle speed of, say, 10 m/h the actual compressor revolutions will be calculated from the known engine speed and gear ratios which produce this vehicle speed and the output calculated from

$$\frac{C \times r \times V}{R} \text{ ft}^3/\text{min}$$

where C = cu ft/min capacity of compressor at its maximum speed

r = r/min of compressor at vehicle speed

V = volumetric efficiency of compressor at 80 lb/in$^2$, usually in the region of 70 per cent

R = maximum revs per minimum rating of compressor.

The maximum number of applications of the brake which may be made without progressively lowering the reservoir pressure is therefore

$$\frac{\frac{CrV}{R}}{\frac{V(P+p)}{p}} \text{ applications per minute.}$$

An item of importance concerns not only the capacity in volume of the compressor, but the time which is required to replenish the reservoir to its maximum pressure after a brake application has reduced the volume and pressure. The reduction in volume or the cubic feet of air used will be the total volume of all cylinders and pipes at atmospheric pressure or $V \times \frac{3{\cdot}4}{1{,}728} = V1$ and the time required will be $\frac{rV_1\,60}{r_1CE} = t$

where $r$ = maximum r/min of compressor
$r_1$ = r/min (actual) of compressor
$C$ = compressor capacity (rated)
$E$ = compressor efficiency.

It is usual when making the foregoing calculations to assume a normal vehicle deceleration of approximately $\frac{g}{4}$ or 8 ft/sec, the vehicle then remaining stationary with the engine idling.

The figures would not be absolutely accurate as a certain drop in pressure occurs inside the reservoir as brake application continues, unless in order to counteract this the unloader valve setting is raised above the nominal 80 lb/in² setting. The amount depends upon the ratio of cylinder volumes to reservoir and may be estimated from the following expression where

$V$ = volume of reservoir
$Vc$ = cylinder volumes (total)
$P$ = brake air pressure
$p$ = atmospheric pressure

$$\frac{(V+Vc)\,P+p}{V} - p \text{ lb/in}^2 \text{ absolute.}$$

A still further consideration must be given to the number of applications possible before the reservoir pressure falls below an amount deemed safe. Should the engine be stalled for any reason and from the foregoing expressions if $N$ = number of applications.

$$N = \frac{\log \frac{P+p}{P_1+p}}{\log \frac{V}{V+Vc}}$$

where notation is as follows—

$P$ = safe reservoir pressure
$p$ = atmospheric pressure
$P_1$ = unloader valve setting (corrected)
$V$ = reservoir volume
$Vc$ = cylinder volume.

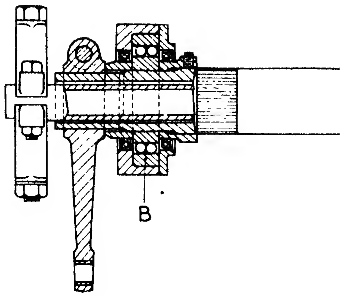


Fig. 164a

From the foregoing notes it should be possible to estimate the approximate specification of equipment required to provide an air-pressure brake system, although, of course, any mechanical weakness would seriously affect the figures obtained. In all brake systems, whether pressure or servo types, rigidity of linkage and brake cross-shaft is essential, as also is the reduction of friction throughout the brake-operating layout ; and in this connection a typical cross-shaft design which provides anti-friction bearings, an adequate shaft diameter and robust levers is shown in Fig. 164. The hook-up between operating levers and pedals and brake shoes is mainly a question of reducing the number of relays and of providing straight rod runs. Study of Fig. 161 will do much to clarify the necessities of correct linkage.

## LEGAL REFERENCES

In connection with hand- and foot-brake layout there are several important regulations with which the design must conform. The actual reading is lengthy and thus reference will be made only to their numbers in the Book of Statutory Rules and Orders No. 8, 1942, but nevertheless they must be studied as not only do they control design to some extent, but they affect the operator of the vehicle. The more important are as follows:

398/41/9
398/41/10P Oct.6/42 P. Sept.4/43 and P. Dec. 7/44
398/41/30, 34, 39
398/41/34
398/41/40(3) /34(3) /39(3)
398/41/30(5) /34(5) /39(5)
514/41/14/15
514/41/2(3).

## STEERING LAYOUT

This brings us to the steering layout and linkage geometry. Having decided upon a suitable dimension for the seat and wheel spacing, the steering gear should be arranged so that the rake of the column is such as to provide a comfortable angle for the steering

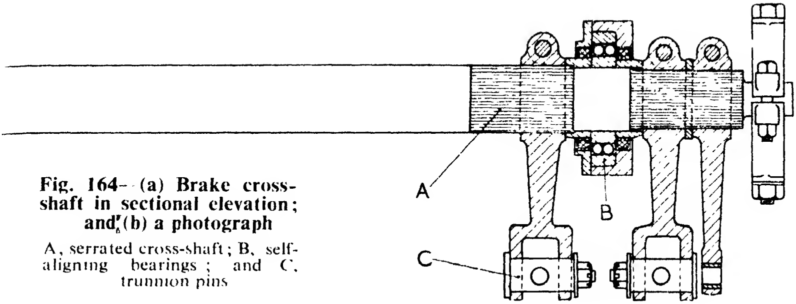


**Fig. 164—(a) Brake cross-shaft in sectional elevation; and (b) a photograph**

A, serrated cross-shaft; B, self-aligning bearings; and C, trunnion pins

**Fig. 164b**

wheel, in the event of the necessity for the driver having to make a full-lock turn quickly for any reason. The frame at the point of steering-box attachment should be as stiff as possible, as the steering mechanism imposes considerable load on the side members, and the method of attachment should be carefully considered.

The spread of the attachment bolts and wide distribution of load on the frame are features of utmost importance. This cannot be overstated, for however accurately the eventual linkage is laid down, if the combination of frame stiffness and rigid attachment are not adequate good steering is impossible.

The steering linkage for a p.s.v. should be arranged to produce an overall ratio from steering box to steering arm of approximately 26 to 29 to 1, giving between five and six turns of the steering wheel from lock to lock. The majority of modern steering gears are designed for a total drop-arm movement of about 80°, of which advantage is taken up to 70° to 72° total drop-arm swing to produce a wheel lock of 40–43°. With the clearance between the tyre and frame decided, the angle of lock on each wheel can be checked, for upon the turning range of the inner wheel depends the angle of set of the track-rod arm.

A diagram is necessary which indicates the error in steering angles in order that the selection of angle of steering arm may be made in which this error is a minimum for all angles of wheel lock. It is known that the angle of the outer wheel differs from that made by the inner wheel when turning, the difference depending upon the relation of the wheelbase to track, and for the purpose of ascertaining the actual angular variation the following expression is usually adopted (see Fig. 165):

$$\cot \beta - \cot \alpha = \frac{W}{T}$$

where W = vehicle wheelbase
T = track at steering-arm pivot
$\beta$ = angle of outer wheel
$\alpha$ = angle of inner wheel.

The relative angular positions in side elevation between the drag link and drop arm assume importance through the articulation of the spring and front axle, as any discrepancy in the steering-drag link geometry produces a virtual lengthening and shortening of the drag link at each spring oscillation, with consequent effect upon the steering performance together with an unpleasant, if not dangerous, movement of the wheel in the driver's hands.

A similar effect will be produced when the spring curls under braking loads, but this is not so pronounced, since correct spring design can reduce this curling practically to a minimum. The road spring does not oscillate at its centre as one would believe but, due to the effect of the clamp plate which almost produces a solid spring centre, it deflects about two points, one on each side of its centre line.

Having obtained these relative angles for various wheel deflections, trial-and-error diagrams are necessary, setting down outer-wheel angles corresponding to a series of inner-wheel angles of say 10, 20, 30 and 40°. Such a diagram is shown in Fig. 165 for a track-wheelbase ratio of 2 to 1. The curve is drawn through the points of intersection and illustrates the degree of deviation from Ackermann layout which is represented by the line a–b. It is usual to lay out several track-rod angles and (in Fig. 165) 15° and 20° have been selected.

It will be noted that for a small angle of wheels from straight ahead direction that of the outer wheel is too small with both the 15° and 20° arms, whilst with the 15° arm the outer-wheel angle in relation to the inner one is correct at about 12° and 39°. The errors are too great at all angles with the 20° arm, and therefore another angle should be tried out in order to ascertain whether there is a more satisfactory solution. The figure also emphasises the fact that had the track-wheelbase ratio been less than 2 to 1 an 18° arm would probably have been practicable.

The method of calculating the virtual centre of spring oscillation has been explained in its appropriate section, and formulae set down which will now be required in the production of the steering-drag link and spring geometry. Fig. 166 shows the diagram for correct layout with no increase or decrease in length of drag link

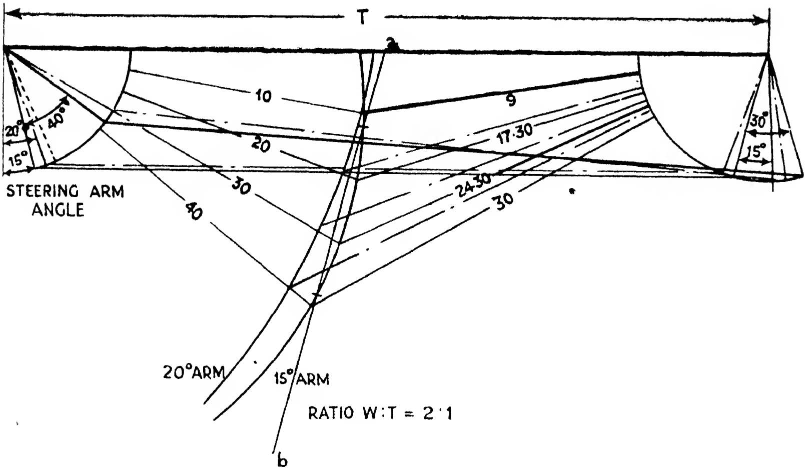


**Fig. 165—Steering-link geometry**

when the road springs oscillate. In consequence, within the prescribed spring deflection, there is no steering interference. Explaining the diagram briefly, AA represents the spring-eye positions when the spring is fully laden, BB is a perpendicular line through the centre of the spring, and O is the point of virtual oscillation of the spring. When the spring is free, that is when mounted on an unladen vehicle, the eye position is represented by A and A denotes the eye position for maximum bump conditions. These positions are obtained by striking a radius with origin at O through the laden spring-eye position and by calculating C and d from the previously given formulae. Naturally if the spring eye is restrained at the forward end, then the spring will oscillate at its centre in a manner which follows the movement of the eye if it were free. Transfer these points to the spring centre as shown at a and b and, bisecting by the usual method of striking an arc from each point, find the line from which the radius Eba will be struck. Along this line mark off a distance EG equal to OA. This is the point of origin for the required radius, whilst the line EO produced should coincide with the eye of the drop arm at b.

It will be found that the interference is very small if a corresponding radius be drawn from b passing through O; O of course represents the ball pin centre on the steering arm and C the centre of the top leaf of the spring. The amount of interference to steering is denoted by the difference in curvature of the two radii EC and BO and shown exaggerated at Fig. 166.

The final position and length of steering drop arm can now be fixed, although it is not absolutely necessary that the drop arm should be vertical when the wheels are straight ahead. Within the confines of the included angle available for drop-arm swing and that obtaining

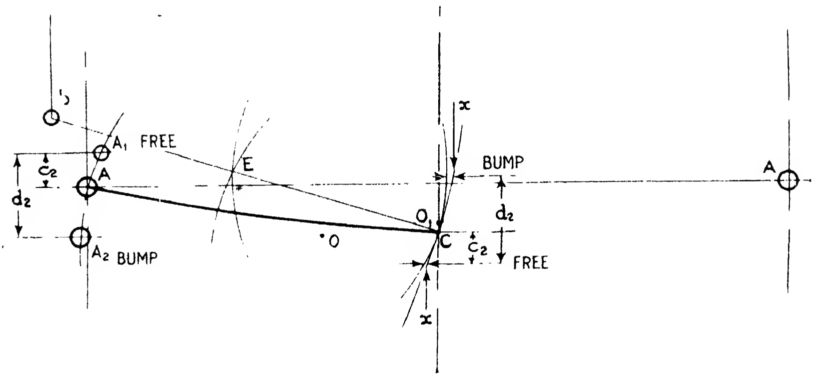


Fig. 166

in the steering box the drop arm may be inclined and thus it is now possible to finalise the steering-gear location. The regulations as affecting steering gear for p.s.v.s state that the gear must be arranged so that no overlock is possible, and so as to prevent any foul occurring between the wheels and any part of the vehicle under any circumstances. Steering arms must be polished, but not plated or painted. Dust covers over joints must be easily removable. Ball-and-socket joints, if used, must not be pendant, whilst if brake or steering connections are secured by bolts or pins, these must be threaded, effectively locked, and, if fitted other than horizontally, their heads must be uppermost (514/41/16) (514/41/17).

## GEAR CHANGE

The position of the gear-change lever next requires consideration. In the case of unit construction of engine and gearbox, and under conditions wherein the combined unit is flexibly mounted in the frame, the gear-change mechanism will be positioned in the engine side, in order that there shall be no relative motion between the gearbox and its control. Should the gearbox, on the other hand, be remote from the engine and clutch, in which event it will probably be rigidly mounted in the frame, then the control mechanism should also be rigidly attached to the frame.

It should be appreciated that in the p.s.v. the engine occupies too great a space to permit the use of a control which is direct on to the selector forks and thus there is usually a relay linkage necessary. The change lever is pivoted and its rocking control included, in the forward selector box, from which the control rod runs to couple with the selector forks, which are included in the rear-gear control case, mounted as a unit on the actual gear casing.

Typical arrangements of such a design are shown in Figs. 167 and 168. The gearbox unit is a sliding-bar type selector having forks attached by wired set screws. A swing lock prevents entry into more than one gear at the same time. Note the ball mounting for the connecting shaft; similarly in Fig. 169, wherein the change-speed lever swings about its ball pivot, rocking the lever end of the connecting shaft. In this example the lever is bodily lifted so that the base of the bottom cup can pass over the guide block for reverse gear engagement. It may be necessary to mount this rear control on the gear-case side, in order to avoid unnecessary relays, rather than to mount the selector forks on the top of the casing. This procedure can be carried through by rotating the box on its housing so that the selector shafts are in line with the gear-change lever casing mounted on the engine side. Difficulty is sometimes experienced in obtaining a straight connecting shaft without some degree of offset since the clutch housing and possibly other accessories provide possibilities of a foul. This, however, is not detrimental to the actual control of the gears providing that the shaft or tube is of adequate section to give sufficient rigidity for the functions it has to perform. There is no fixed standard for gear-lever positions in relation to the gear engaged; this, of course, is on account of the many

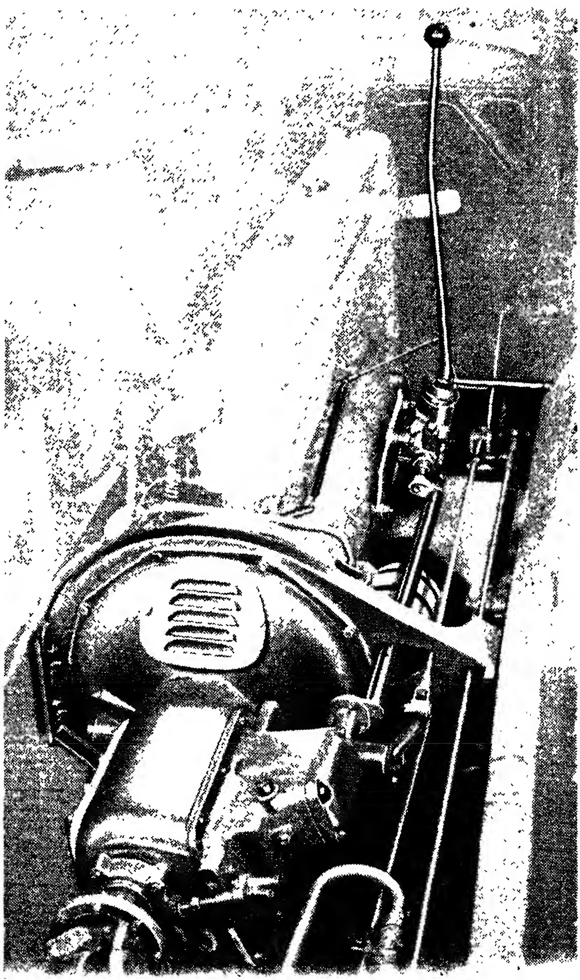

**Fig. 167—Layout of change-speed gear**

different arrangements of gearbox which exist, but the generally accepted " gate " is that suggested in the I.A.E. standards. The reverse-gear position may be anywhere in relation to the other speeds.

The gear lever should be as short as consistent with easy and convenient operating position for the driver, whilst its length and section should not be such as to promote a vibration with possibility of fracture through fatigue. The length is somewhat bound up with the necessary lever ratios required to provide correct gear movement through the selector forks and with the possibility of a satisfactory location on the engine side. The lever movements should be such that the gears completely engage for the whole width of their teeth, and friction in all the rotating and sliding shafts should be reduced to a minimum by careful detail design and the provision of adequate lubrication.

All such details would be taken care of during the subsequent scheming and, since it is the purpose of this section to consider only actual chassis layout, we continue with the location of the fuel tank and system and the exhaust pipe run and silencer.

## FUEL SYSTEM LAYOUT

Although there is little which can be set down regarding the actual fuel-pipe layout, after having established the most satisfactory position for the main tank, the fuel is delivered to the carburettor via pump or autovac in the case of the petrol engine, and through separate injectors with an oil engine.

In the latter design the fuel-pipe system provides the following sequence: Fuel is raised by pump through a filter, thence to the injection pump, from which it is delivered at high pressure to the injectors in the cylinder head. The function of these injectors is to break up the fuel into a fine mist and to distribute it to the combustion chamber, whilst at the same time they also ensure a quick beginning and end to the injection period. The inlet connection is provided with a filter from which the fuel is fed through a passage to the nozzle. This in turn incorporates a valve whose function it is to snap open under the required pressure, thus allowing the mist to be sprayed into the combustion chamber through small orifices in the nozzle tip. A small quantity of fuel inevitably leaks past the valve stem, and this is lead away through passages to the main leak off gallery pipe eventually discharging into the filler neck of the fuel tank.

It would appear that the main points of concern regarding the fuel system layout are:

(a) The incorporation of adequate filter arrangements in the system and in the tank.
(b) Provision of the best-quality tubing free from all scale.
(c) That the pipes are of sufficient diameter to avoid starvation.

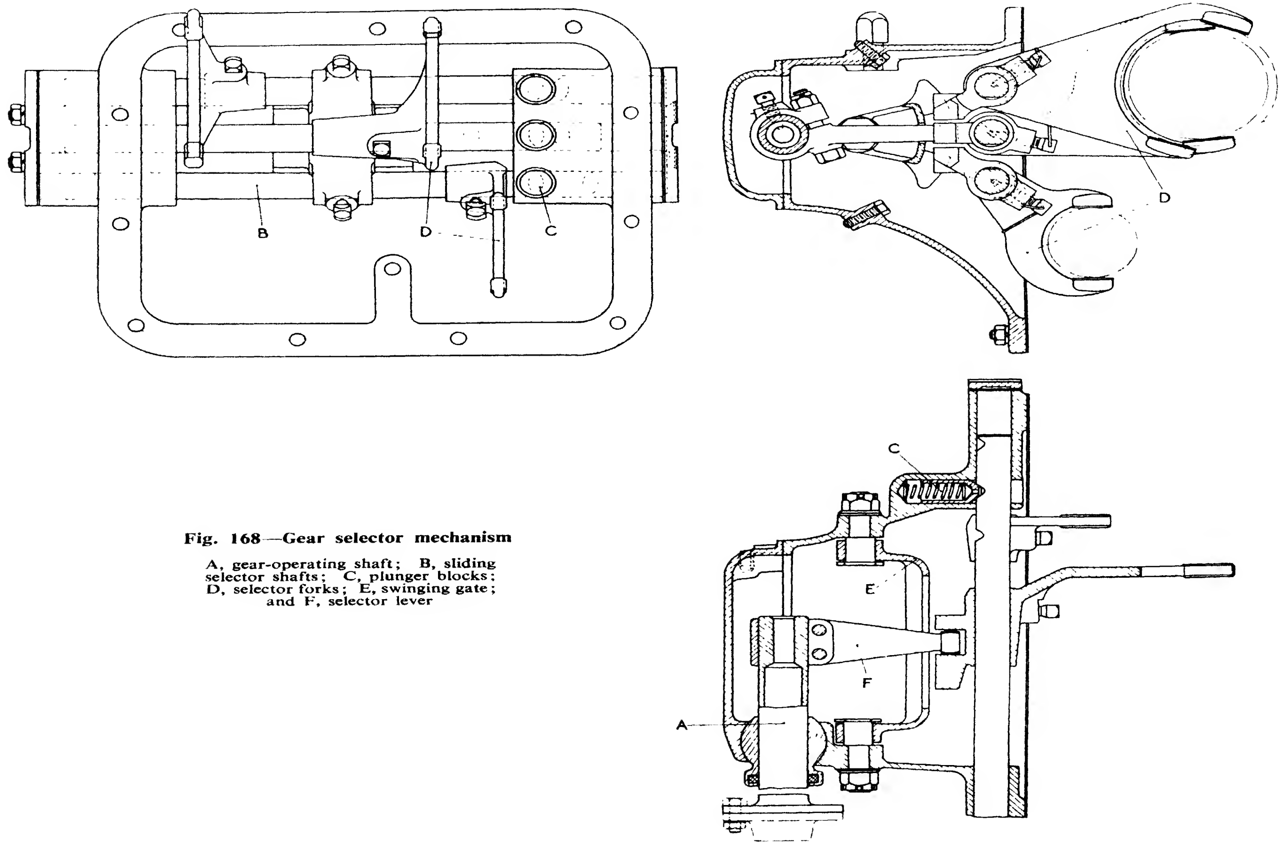


**Fig. 168—Gear selector mechanism**

A, gear-operating shaft; B, sliding selector shafts; C, plunger blocks; D, selector forks; E, swinging gate; and F, selector lever

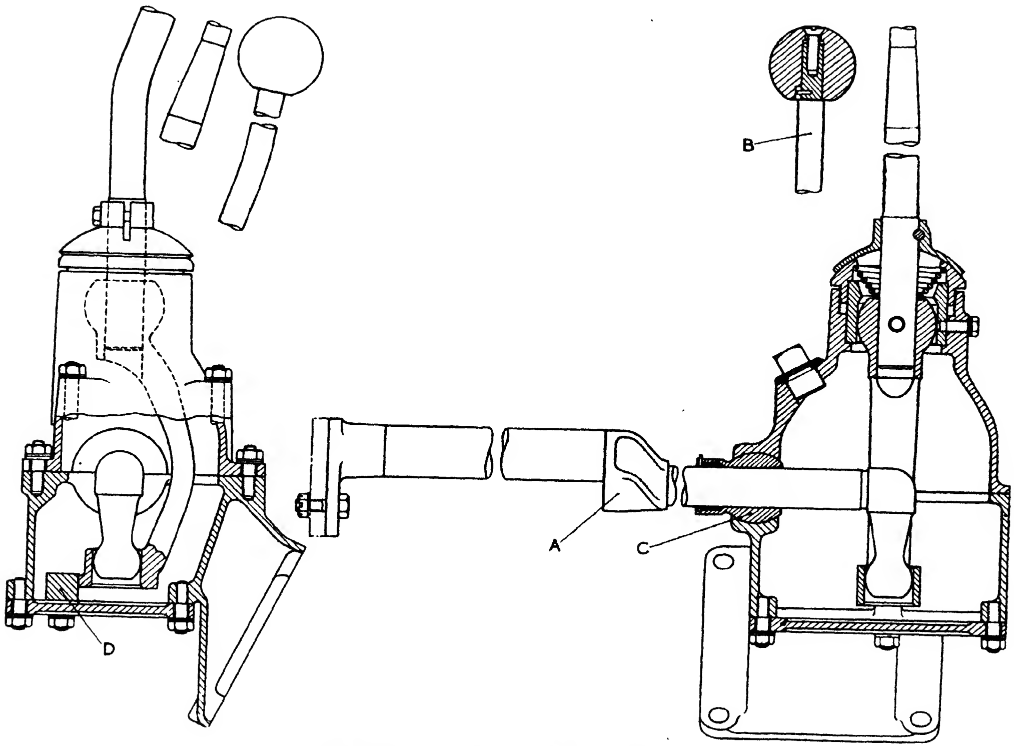


**Fig. 169—Gear selector lever assembly**

A, gear operating shaft; B, change-speed lever; C, aligning bearing; and D, reverse stop

(d) Also that the number of pipe joints and connections are as few in number as possible.
(e) The greatest care must be taken to avoid long lengths of pipe which can vibrate without support, and the correct location of and provision for expansion coils.
(f) That no part of the system is too close to the exhaust manifolds or pipes.
(g) The tank capacity is sufficiently large to contain enough fuel without replenishment in the case of a p.s.v. for one day's running, and in the case of a heavy goods vehicle for 450 miles maximum.

The constructional features which call for maximum consideration may be summed up as follows:

(a) Adequate fixing for tank to chassis; and in this respect it is suggested that it is safer to support the tank rather than to sling it from the side member.
(b) Provision of baffles to prevent surge, and cleaning apertures large enough to admit a hand in the main fuel tank.
(c) Whether autovac or fuel pump is fitted either should be of ample capacity to deal with the required flow of fuel.
(d) That the suction pipe in the tank is positioned so that it is impossible under any circumstances other than an empty tank to suck air, and that breathing or pressure release is not overlooked.
(e) The filler cap must be accessible when the body is in position.
(f) That orders and regulations are satisfied. These orders are set down below.

No fuel tank may be placed under any part of any gangway which is within 2 ft of any entrance or exit. Every tank must be so arranged that no fuel from it can overflow on to any woodwork, or accumulate where it can be ignited readily. The fuel-supply line must incorporate a cock for cutting off the fuel and the means of operation thereof must be visible and at all times readily accessible from the outside of the vehicle. The " off " position must be clearly marked on the outside of the vehicle. All filling points must be outside the body with caps which can be securely fixed. The vent hole, if any, must be of the non-splash variety and protected from danger of penetration by fire. All carburettors and similar fittings are required to be placed and shielded so that no leaking fuel can fall on any part of the vehicle capable of igniting it, or into any receptacle where it might accumulate (514/41/20, 21, 22).

The law relating to the exhaust system of a p.s.v. states that " the exhaust pipe must be kept clear of any inflammable material on the vehicle and placed or shielded so that no such material can be thrown on to it from any other part of the vehicle. The outlet of the pipe has to be on the offside of the vehicle and far enough to the rear to prevent as far as possible fumes from entering the interior of the vehicle " (514/41/23).

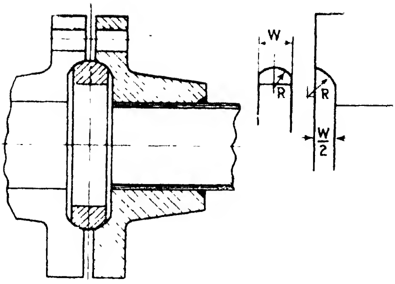


Fig. 170 Exhaust manifold joint

The offside outlet, however, does not apply to a goods vehicle, but with a p.s.v. it means that as the manifolds on the engine will almost surely be on the nearside a cross-over is necessary and it is usual to exhaust behind the rear of the fuel tank. An endeavour should be made to avoid sharp bends in the pipe, particularly at the point of attachment to the manifolds, as the heat of the gases impinging will eventually burn the pipe. A long sweep is therefore desirable.

As a matter of interest, it is often the custom to lag the pipe with asbestos, but such a practice is a fallacy as the asbestos performs its function in keeping the heat in the pipe and prevents radiation, and as a consequence the pipe material deteriorates much more rapidly than if radiation were permitted.

The type of silencer or expansion chamber will have been decided on prior to engine test, but having decided upon one which reduces the noise to a minimum and which also produces little or no back pressure, it will be necessary to position it as near to its optimum distance from the exhaust manifold as possible. In this respect it should be appreciated that an expansion chamber has within quite small limits one position in the pipe line at which it functions best in conjunction with particular engine characteristics and it is usually possible to arrange this. Another point which should not be overlooked is the fact that such a chamber also acts as a gas cooler, whilst too long a length of tail pipe not only is an encumbrance but will have an adverse effect upon the matched engine and silencer characteristics.

The silencer chamber should preferably be flexibly mounted on the chassis to allow for expansion throughout the system, and the pipes must of course possess a sectional area at least equal to that of the manifold outlet. The actual pipe run can best be checked and suitable supports arranged on the prototype vehicle. A point of refinement in the method of jointing the pipes to the manifold, and at the silencer inlet and outlet, is shown in Fig. 170. This may also be adopted at any pipe joint. It removes the necessity for a gasket between the flange faces and makes an effective seal between the insert circumference and the flange recess. The insert radius is similar to that of the flange recess, but in order to ensure definite wedging faces against gas leak the radius is offset on the flange recess, which of course produces a gap between the two flange faces. This is shown in Fig. 170 also.

## ELECTRICAL EQUIPMENT

Apart from the minor layouts, the general lines of the chassis will have taken shape, leaving such units as engine controls, chassis lubrication, etc., to be completed, and in these cases there is little which can be said regarding method of layout. There is, however, the electrical equipment and wiring, which is in the nature of a specialist's problem.

For a p.s.v. it is usual to fit 24-volt equipment for lighting and starting, together with batteries of approximately 185 ampere-hour capacity. These batteries occupy considerable space and are heavy in construction. It appears to be a matter of personal preference regarding their mounting and location, but the tendency is to mount them in removable steel containers attached to the chassis side frame. They must, of course, be accessible with the body in position. A very neat layout can be attained by housing the switchgear, cut-out voltage regulator, fuses and battery isolator as a complete unit, whilst the connections are accessible and the electrical installation sound. Provision for the horn and head-lamp dipping switch can be made on the steering column.

The wiring system may preferably be harnessed in separate assemblies, and on the latest arrangements pin-and-socket connectors are included at points where quick detachability is desired. Instruments should be placed so that easy vision is obtained by the driver and they should be adequately illuminated for night driving.

# CHIEF STATUTORY REQUIREMENTS AFFECTING MOTOR VEHICLES

| | Trolley Buses | | §Locomotives | | §Tractors | Heavy Motor Cars | | | Motor Cars | | | M. Cycles | Trailers |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Single Deck | Double Deck | Light | Heavy | | Goods | P.S.V.'s Single D | P.S.V.'s Double D | Goods | Passenger up to 7 Seats | P.S.V.'s over 7 Seats | | |
| 1. DIMENSIONS | | | | | | | | | | | | * | |
| Height (Z) | 11′ 0″ | 15′ 10″ | — | — | — | — | 15′ 0″ | 15′ 0″ | — | — | 15′ 0″ | — | — |
| Length, 4 Wheels | 27′ 6″ | 26′ 0″ | 27′ 6″ | 27′ 6″ | 27′ 6″ | 27′ 6″ | 27′ 6″ | 26′ 0″ | 27′ 6″ | 27′ 6″ | 27′ 6″ | — | 22′ 0″ |
| „ over 4 Wheels | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | 30′ 0″ | — | 22′ 0″ |
| „ Articulated | — | — | — | — | — | 33′ 0″ | — | — | 33′ 0″ | — | — | — | — |
| „ with Trailer s (Y) | — | — | 137′ 0″ | 137′ 0″ | 60′ 0″ (W) | 60′ 0″ | — | — | 60′ 0″ | 60′ 0″ | — | — | —(V) |
| Overhang, Rear | ½ W.B. | ½ W.B. | — | — | 6′ 0″ | ½ W.B. | ½ W.B. | ½ W.B. | ½ W.B. | ½ W.B. | ½ W.B. | — | — |
| „ Side | 0′ 6″ | 0′ 6″ | — | — | — | — | 0′ 6″ | 0′ 6″ | — | — | 0′ 6″ | — | — |
| Width, on any Road | 7′ 6″ | 7′ 6″ | 9′ 0″ | 9′ 0″ | 7′ 6″ | 7′ 6″ | 7′ 6″ | 7′ 6″ | 7′ 6″ | 7′ 6″ | 7′ 6″ | — | 7′ 6″ |
| „ on Specified Roads | 8′ 0″ | 8′ 0″ | — | — | — | — | 8′ 0″ | 8′ 0″ | — | 8′ 0″ | 8′ 0″ | — | — |
| 2. GROUND CLEARANCE | 0′ 10″ | 0′ 10″ | — | — | — | — | 0′ 10″ | 0′ 10″ | — | — | 0′ 10″ | — | — |
| 3. GUARD RAILS TO WHEELS | Yes | Yes | No | No | No | No | Yes | Yes | No | No | Yes | No | No |
| 4. MARKING | | | | | | | | | | | | | |
| Name and Address | No | No | No | No | No | No | No | No | No | No | No | No | No (B) |
| Speed | Yes | Yes | No | No | Yes | Yes | Yes | Yes | No | No | Yes | No | — |
| Speed Discs | — | — | (E) | (E) | Yes | Yes | (D) | (D) | (D) | No | (D) | No | (C) |
| Unladen Weights | Yes | Yes | Yes | Yes | Yes | Yes | Yes (G) | Yes (G) | No | No | Yes (G) | No | (F) |
| 5. SPEED LIMITS | | | | | | | | | | | | | |
| (i) Without Trailer | | | | | | | | | | | | | |
| On Pneumatics ... M.P.H. | 30 | 30 | 12 | 5 | 20 | 20 | 30 | 30 | 30 (J) | Nil | 30 | Nil | — |
| „ Soft or Elastic ... „ | 20 | 20 | 12 | 5 | 20 | 20 | 20 | 20 | 20 (H) | Nil | 30 | Nil | — |
| „ Other Tyres ... „ | — | — | 5 | 5 | 5 | 5 | — | — | 5 | — | — | — | — |
| (ii) With Trailer | | | | | | | | | | | | | |
| On Pneumatics .... M.P.H. | — | — | 12 | 5 | 20 | 20 | — | — | 20 | 30 | — | Nil | — |
| „ Soft or Elastic .... „ | — | — | 12 | 5 | 12 | 12 (J) | — | — | 12 (J) | 20 | — | Nil | — |
| „ Other Tyres .... „ | — | — | 5 | 5 | 5 | 5 | — | — | 5 | — | — | — | — |
| 6. TILTING ANGLE | 25° | 25° | — | — | — | — | 35° | 28° | — | — | 28° | — | — |
| 7. TURNING CIRCLE | (O) 60′ or 66′ | (O) 60′ or 66′ | — | — | — | — | (O) 60′ or 66′ | (O) 60′ or 66′ | | | (O) 60′ or 66′ | | |

| | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 8. TYRES | | | | | | | | | | | | | | | |
| Pneumatic | Yes | | Yes | | No | No | No | Yes | Yes | Yes | Yes (K) | Yes (K) | Yes | Yes | Yes |
| Soft or Elastic | — | | — | | Yes | Yes | Yes | — | — | — | — | — | — | — | — |
| 9. WEIGHT | L.P. | H.P. | L.P. | H.P. | | | | | | | | | | | |
| Axle, 4 Wheels .... Tons | 8 | 8 | 8 | 8 | — | — | — | 8 (R) | 8 | 8 | 8 (R) | | 8 | — | 6½ (S) |
| ,, Over 4 Wheels.. ,, | 8 | 8 | 8 | 8 | — | — | — | 8 | 8 | 8 | 8 | | 8 | — | |
| Distribution on 2-ft. Strip..Tons | 10 | 10 | 10 | 10 | (U) | (U) | 10 | 10 | 10 | 10 | 10 | 10 | 10 | — | 10 |
| Gross Laden = (Zc) | | | | | | | | | | | | | | | |
| Without Trailer | | | | | — | — | — | — | | — | — | | — | — | — |
| 4 Wheels .... Tons | 12½ | 12½ | 12½ | 12½ | 14½ | 20½ (L) | 9 | 12 (N) | 12 | 12 | 12 | | 9½ | — | — |
| 6 ,, .... ,, | 14½ | 14½ | 14½ | 14½ | — | — | — | 19 (N) | 14 | 14 | 19 | | 13 | — | — |
| Over 6 Wheels ... ,, | — | — | — | — | — | — | — | 22 | — | — | 22 | — | — | — | — |
| With Trailer | — | — | — | — | 40 | 40 | 22 | 32 (Zd) | — | — | 22 | 22 | — | — | 13 (M) |
| Unladen | | | | | 11½ | 15½ (T) | 7¼ | — | — | — | 3 | 3 | 2½ | 8 cwt | — |

## ANNOTATIONS

- * Tricycles of over 8 cwt unladen weight classed as motor cars
- § Vehicles not constructed to carry load
- B Trailers not forming part of motor vehicle to have name and address
- C Yes, if part of articulated vehicle
- D Yes, if on soft tyres
- E Yes, if not drawing trailer
- F Yes, if over-run brakes fitted
- G Including number of seats
- H 30 m.p.h., if under 1 ton
- J 20 m.p.h., if articulated
- K If over 1 ton
- L 18½ tons, if not on soft or elastic tyres
- M Based on 2-axled trailer, 6½ tons each axle
- N If steam or solid-fuel propelled, 12 tons increased to 14 tons, and 19 tons to 20 tons
- O For lengths up to 26 ft and over, 26 ft respectively
- P Based on 140 lb per person; see O
- R 9 tons for steam or solid-fuel propelled
- S 8 tons, if part of articulated vehicle
- T 17½ tons, if on soft or elastic tyres
- U Three-quarters of total weight on any two wheels
- W 100 ft if unladen
- X Gongs, bells and sirens prohibited except for special purposes
- Y Motor vehicle drawing a trailer not now restricted to 26 ft
- Z Trolleybus height over trolley base
- Za "L.P." = low pressure, means not over 85 lb/sq. in. "H.P." = high pressure
- Zc Laden weight of public service vehicle is total of vehicle with water, oil, and fuel, plus 140 lb per seat and 140 lb for driver and conductor each. Same basis applies to trolleybuses "fully equipped for service"
- Zd 32 tons gross only permitted (with trailer) if trailer fitted with power-assisted brakes operated by driver and if operative with engine stopped. Articulated vehicle not included. Otherwise 22 tons gross

NOTE: The weights under heading 9 are in force at present (January, 1948), but they may be increased in some instances in the near future.

# INDEX

# INDEX



## C

## D

## INDEX

## R

# INDEX



## S

## T

## U



## V

## W



## Z