AUTOMOBILE BRAKES AND BRAKE TESTING

PITMAN'S AUTOMOBILE MAINTENANCE SERIES

AUTOMOBILE BRAKES AND BRAKE TESTING

DEALS WITH BRAKING PRINCIPLES AND THE CONSTRUCTION, MAINTENANCE, AND TESTING OF BRAKING SYSTEMS

BY

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PREFACE

This book has been written with the aim of giving practical information about braking mechanisms and brake testing equipment as applied to the modern motor car.

The fundamental principles upon which all such apparatus is based are described in the early chapters because a knowledge of these principles is essential to successful work on brakes, whether it be concerned with adjustment, maintenance, or testing. Some parts of the description require an elementary knowledge of mechanics and the use of formulae but, in general, it can be followed by readers who have not had technical training.

Latterly, a close interest in efficient braking has been taken by the Ministry of Transport which led, in 1937, to the provision of regulations conferring authority upon the police to test motor car brakes; frequent references to brake performance are also made in the courts. Another modern trend is to develop garage equipment expressly designed to facilitate brake maintenance and testing.

In view of these facts it has become important for everyone concerned with the servicing and operation of motor cars to have a clear understanding of braking problems. The author will feel satisfied if his book succeeds in contributing towards this end.

MAURICE PLATT

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AUTOMOBILE BRAKES AND BRAKE TESTING

CHAPTER I

GENERAL PRINCIPLES

Good braking is universally accepted as being essential to the safe operation of a motor car. The importance of this aspect of road performance has been stressed in recent years by the congested condition of British roads and by the increasing number of road accidents.

It is the responsibility of the car manufacturer to provide brakes that are effective, safe in operation, progressive and consistent in response to pedal effort, and reasonably easy to adjust. After the car has left the factory, however, the responsibility is transferred to the owner and driver because no brake yet designed will retain its original qualities without regular attention and adjustment. And human nature being what it is, motorists have had to be encouraged in every possible way to see that their brakes are properly maintained in service.

Encouragement has taken various forms. The manufacturer emphasizes the importance of brake maintenance in his instruction books and service manuals; the leading makers of proprietary braking systems issue similar publications, and servicing facilities are available throughout the country. Finally, and more recently, the Ministry of Transport has taken action in the form of revised "Motor Vehicles (Construction and Use) Regulations" which empower a police constable in uniform, or an accredited examiner, to test and inspect the brakes of motor cars selected at random.

In taking this step the Ministry is following the lead of many authorities in the United States, Canada, and certain European countries where, in addition to compulsory testing, minimum standards of brake performance have been in force for some considerable time. So far, no such standards have been proposed in Great Britain, the legal requirement being that the brakes shall be "sufficient under the most adverse conditions to bring the vehicle to rest within a reasonable distance."

There is something to be said for this broad statement, in spite of its ambiguity, because the factors which affect the definition of a "reasonable distance" are, as we shall see, extraordinarily numerous and diverse. So the motorist is left with the responsibility of seeing to it that his brakes are maintained in the best possible condition consistent with their design and that of the vehicle to which they are applied.

Limitations to Braking. No system of brakes can give results beyond a limit imposed by the adhesion between the tyres and the road. Everyone is familiar with this limitation on a slippery surface; when a certain pedal effort is exceeded the wheels lock and slide, and no additional braking effect can be obtained; indeed, there are circumstances in which locked wheels retard a car less effectively than when the brakes are applied to an extent just below the locking point.

The retarding effect which can be obtained from a wheel therefore depends upon the horizontal force which that wheel will stand before ceasing to roll and commencing to slide. The magnitude of this force is mainly dependent upon two factors: the downward load on the wheel and the coefficient of adhesion between the tyre and the road.

Now, as we shall see presently, the rate at which a car loses speed when braked—the deceleration, in short—depends upon the ratio between the retarding force and the weight of the car. It follows that we can take a short cut by saying that maximum deceleration is simply dependent upon the adhesion available between the tyres and the road, irrespective of the weight of the car, provided that the brakes are sufficiently powerful to make use of all the adhesion available.

However, before extending this argument any further, it will be as well to get down to facts by looking at the forces which are exerted when a single wheel is braked. In the following account it is assumed that the wheel is rolling on a level surface; the effects of upward and downward gradients will be considered later in this chapter.

Braking Forces on a Wheel. In Fig. 1 a single wheel is shown complete with tyre, brake shoes, and brake drum; the back plate which normally encloses the drum, together with the axle, is omitted in order to make the mechanism visible. When the driver applies a force to the brake pedal, a pull is exerted through rods and cables to a lever and cam (or some similar device) so as to expand a pair of hinged shoes into contact with the drum which rotates with the wheel.

The maximum braking effect is obtained when the frictional drag exerted by the shoes on the drum and wheel does not quite prevent the wheel from continuing to turn.

If this limit is exceeded, the wheel becomes locked and the tyre "skates" along the road surface instead of rolling.

This line of thought raises an interesting query—namely, why is it that the wheel goes on rolling so long as the braking effect does not exceed a certain limit? The answer is that a force, sufficient to keep the wheel turning against

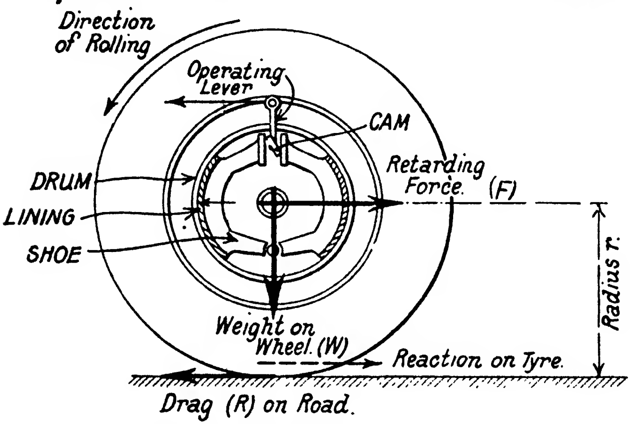

FIG. 1. DIAGRAMMATIC VIEW OF A BRAKED WHEEL, SHOWING THE PRINCIPAL FORCES WHICH ARE ACTING UPON WHEEL, AXLE AND ROAD

the brake, is created by friction between the tyre tread and the road. Usually this frictional effect is simply called tyre adhesion.

Perhaps the easiest way to picture what is taking place is to imagine that the wheel is rolling over a specially prepared strip let into the road, the strip being supported by rollers free from friction so that it can slide endwise. As a further step we can imagine that the strip is restrained by a spring balance of some kind, as shown in Fig. 2.

If the brakes are applied while the wheel rolls over this imaginary apparatus, we shall find that the strip tends to be dragged along in the direction in which the car is moving. Furthermore, our spring balance will show that the drag on the strip is precisely equal to the horizontal retarding

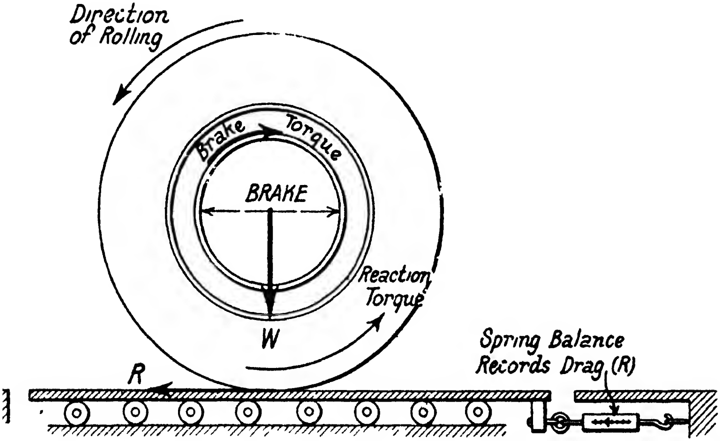

FIG. 2. IF A BRAKED WHEEL ROLLS OVER A STRIP, MOUNTED AS SHOWN, THE DRAG (R) IS EQUAL TO THE BRAKING FORCE AND THE REACTION TORQUE EQUALS THE BRAKING TORQUE

force which is exerted through the hub of the wheel and so, by the way of the axle and springs, upon the car as a whole. The reaction at the road surface, or strip, is the force which keeps the wheel turning against the brake.

Now we might extend our experiment by recording the maximum retarding force which we could get without tyre slip when a series of loads was applied to the wheel in a downward direction. We should find that every time we increased the load the retarding force obtainable would go up in exactly the same proportion; in other words the ratio between the two—force and load—would remain always about the same. Just what this ratio would be depends upon the condition of the tyre and the surface of our specimen strip of road. For a good tyre on a dry surface it is scarcely ever possible to get a ratio better than 1·0, which simply means that the retarding force is never likely to exceed the load on the wheel under ordinary operating conditions. As a rule, a ratio of 0·8 can be taken to represent quite a good result, for dry surfaces.

For example, if the wheel carries a load of 400 lb., a ratio of 0·8 means that the best retarding force at locking point will be 400 × 0·8, i.e. 320 lb. If the load is increased to 500 lb., the braking force can be improved up to 400 lb., and so on. On wet roads the coefficient of adhesion—this is the correct name for our force/load ratio—will be considerably reduced, and consequently the best retarding effect at any given load will decrease in strict proportion. Thus, if the wheel load is maintained at 500 lb. but the coefficient decreases to 0·3 (on a greasy surface), the maximum braking force which we can obtain will be reduced from the dry surface figure of 400 lb. to a mere 160 lb. The best brakes in the world cannot do more than make full use of whatever cofficient of adhesion may be available to keep the wheel rolling.

Before leaving the single wheel, take one more look at Fig. 1 and the forces which are shown there. First there are the horizontal forces already described, namely, the force (R) which drags the road surface, the reaction which keeps the wheel rolling, and which is also exerted between the road surface and the tyre, and the equal retarding force (F) which the braked wheel exerts upon the axle and car. Next there is the downward load (W) which holds the tyre in contact with the road. Using these letters to represent the forces, and supposing that the braking effect is enough to bring the wheel almost to locking point, we can express the ideas which we obtained from our experiment by the following simple formula—

$$\text{Coefficient of adhesion} = F/W$$

Another way of viewing the condition of a braked wheel is in terms of torque, by which is meant the turning effect exerted by forces acting at a certain leverage. Thus the frictional drag of the shoes on the brake drum produces a torque tending to stop the wheel from rotating. The opposite pair of forces (reactions of F and R) produces an equal torque tending to keep the wheel turning.

Size of Brake Drum. This conception is useful when thinking in terms of the diameters of the brake drum and the wheel. Torque is measured by multiplying one of a pair of forces by the distance which separates them. For example, if the rolling radius of the wheel is denoted by r, as in Fig. 1, the torque which keeps the wheel turning is $F \times r$. Supposing that the force is 400 lb. and the radius 13 in., the torque is simply 5200 lb.-in.

It follows that the torque exerted by the brake shoes on the drum is equal in amount although opposite in direction. Consequently, the frictional forces on the drum can easily be calculated from its internal diameter. To take a simple example, if the diameter is 10 in. then each force will have to be 520 lb.

It would be quite right to conclude from this calculation that the bigger the drum the smaller are the frictional forces which are needed between the shoes and the drum in order to produce a certain braking effect. In practice, however, a big drum has various disadvantages; it cannot be made so rigid as a small drum, unless at the expense of excessive weight, and it gets so close to the rim and tyre that the heat created by braking may easily cause damage to the rubber.

Designers therefore choose a drum of reasonable size and arrange the brake shoes and their operating mechanism in such a way that the shoes can produce sufficient drag to provide the torque required. There is no difficulty about doing this and it is therefore a fallacy to suppose that the bigger the drums the more effective are the brakes. Incidentally, the life of the brake linings largely depends upon the pressure which they carry per square inch, and therefore is a matter of lining area. Here again, drum diameter is not a fair criterion because area depends upon width as well as diameter.

Four-wheel Braking. The explanation of the limit to the amount of braking which can be obtained from a single wheel carrying a certain load shows that in order to obtain the best possible braking effect upon a complete car all four wheels must be utilized for braking purposes. This has been common practice ever since 1924 although before that time the great majority of cars were braked upon the rear wheels only. These remarks apply to the pedal-operated system, or what is often called the service brake. The hand lever is sometimes arranged to operate the same four brakes independently of the pedal, or in other cases it applies the rear brakes only. There are a few cars in which the hand lever operates a separate brake on the transmission. In still fewer cases, mainly large and costly cars, it is connected to a separate pair of brakes in the rear wheel drums.

Extending our conception of the braking forces exerted on a single wheel to a car rolling on four wheels, it will be obvious that if the service brakes take full advantage of all the adhesion available, the ratio between the total retarding force experienced by the car with wheels at locking point, and the weight of the car, will be simply our old friend the coefficient of adhesion. Thus under normal dry road conditions we can expect to get a retarding force equal to 0·8 of the weight of the car, and under the very

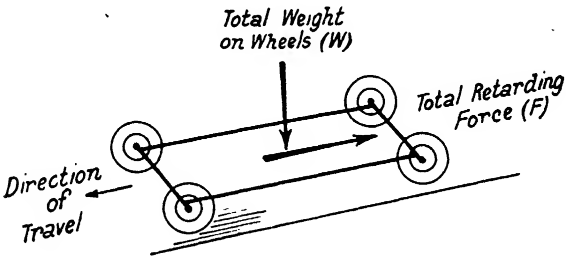

FIG. 3. THE DECELERATION PRODUCED BY THE BRAKES IS PROPORTIONAL TO THE RATIO BETWEEN THE TOTAL RETARDING FORCE (F) AND THE WEIGHT OF THE VEHICLE (W)

best conditions it is not likely to exceed this weight (Fig. 3).

The importance of this idea lies in the fact already touched upon in connexion with a single wheel; namely, that the deceleration, or rate of losing speed, is always proportional to the ratio between the retarding force and the weight of the vehicle to which this force is applied. Now as the ratio cannot exceed the limit represented by the coefficient of adhesion, it follows that there is no fundamental reason why a heavy car should not be capable of stopping at just the same rate as a light car. The only practical requirements are that the brakes should be powerful enough, and that their effort should be so distributed as to make full use of the adhesion which each tyre is ready to provide.

The second of these provisos should always be kept in mind by anyone responsible for the adjustment and maintenance of braking systems. The engineer engaged upon such work has no control over the design of the brakes but, by ensuring that they shall all exert their full effect, he can do a great deal towards improving the deceleration which is obtainable. Furthermore, the risk of locking a wheel and promoting a swerve or skid on a slippery surface is very much reduced by good distribution.

Brake " Efficiency " and Deceleration. We now come to a part of the subject which often causes a great deal of confusion. So far we have thought about the ratio between retarding force and car weight as representing the coefficient of tyre adhesion; in other words, we have assumed that our brakes are capable of creating enough drag on the wheels to use all the adhesion available, even on a dry road. Under practical conditions, however, we know quite well that brakes which are out of adjustment, or suffering from other faults, are not capable of bringing the wheels anywhere near to locking point, and that the best deceleration actually obtainable is therefore well below the result which we know we ought to attain.

Clearly, it is essential to invent some way of setting down the performance which the brakes are actually providing, and the method chosen should enable the performance to be easily compared with some chosen standard of excellence.

At one time the usual plan was to state that the brakes were capable of stopping the car in a certain distance from a chosen speed. The main difficulty was to get everyone to choose the same speed. For example, if one man says that his brakes will stop his car in 38 ft. from 30 m.p.h. while another requires 17 ft. from 20 m.p.h., it is impossible to tell which is the better performance without making a calculation or referring to a table.

The same difficulty arises when brake performance is described in terms of the time taken to stop a car from a given speed. Another point is that there are considerable practical difficulties in attempting to measure either time or distance with accuracy, as we shall see in the next chapter.

For these reasons it has become the fashion to use the ratio between the retarding force actually exerted and the weight of the car, as a means for describing the effectiveness of the brakes. We have already seen that the limit imposed by tyre adhesion will very seldom allow this ratio to exceed 1·0, so that this figure can conveniently be taken as the top limit, described as "100 per cent efficiency." If the brakes of a certain car are not capable of producing a retarding force greater than half the weight of the car, the ratio is 0·5 and they are said to have an efficiency of 50 per cent. The same rule applies throughout the range of ratios.

In some ways it is a pity that the word "efficiency" has become so widely used for this purpose. Semi-technical people, and legal gentlemen of no technical knowledge, are apt to imagine that any good braking system should give 100 per cent whereas in actual fact this figure is hardly ever obtained, even on dry roads. Eighty per cent is an excellent figure and few drivers would grumble as long as they were on the right side of 60 per cent. It would be better to call the ratio by some other term such as "brake performance factor."

However, as "efficiency" has become so widely recognized we will have to keep to it. The important things to remember are (1) that 100 per cent is an arbitrary limit, almost unattainable in practice but beaten occasionally in stunt tests; (2) that efficiency figures obtained by trial on the road can only be used to indicate how good, or how bad, the brakes may be provided that there is enough tyre adhesion to enable the brakes to develop the full effect of which they are capable. Thus on a slippery road an efficiency meter can be used to show the deceleration achieved but gives no indication of how efficient the brakes might be on a dry surface.

"Efficiency" and Stopping Distance. Although tables are available from brake makers and other sources which

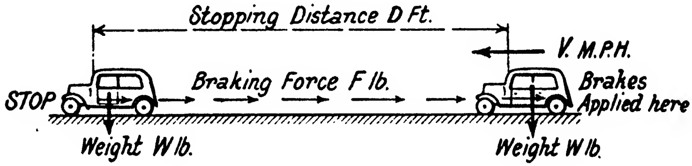

FIG. 4. THE WORK DONE BY THE UNIFORM RETARDING FORCE (F) IN STOPPING A CAR IS EQUAL TO THE KINETIC ENERGY ORIGINALLY POSSESSED BY THE MOVING VEHICLE (see text)

show the stopping distances that correspond to various brake efficiency figures at a number of different speeds (an example is given on page 17, reproduced by permission of Messrs. W. Tapley), it is well worth while to study the relationship between these factors.

A moving car possesses a store of energy in virtue of its weight and speed. When the brakes are applied, this energy is converted by friction into heat at the brake drums. The diagram in Fig. 4 shows a car originally travelling at a speed of V m.p.h., being brought to a stop in a distance of

D ft. by a steady retarding force of F lb.; the weight of the car, also measured in lb., is represented by W.

The usual engineering formula for energy can be simplified, measuring speed in miles per hour and weight in pounds, into the following form—

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

Now the work done by the brakes is simply the retarding force multiplied by the stopping distance through which it acts and must be precisely equal to the energy which has been dissipated. Consequently, we can say—

$$F \times D = WV^2/30$$

By transposing the terms of this formula in the usual algebraical manner we arrive at the following result—

$$D = (V^2/30) \div (F/W)$$

It has been written deliberately in this unusual way because the ratio F/W represents our old friend force/weight. It need only be multiplied by 100 in order to become brake efficiency, which we will denote by E, per cent. So our formula finally becomes—

$$\boxed{D = V^2/.3E}$$

The first thing which this formula reveals is that for any given efficiency the braking distance is proportional to the square of the speed. In other words, supposing a car can be stopped in 40 ft. from 30 m.p.h., then if the brakes remain equally effective it will require a distance of 160 ft. from 60 m.p.h. (speed doubled and distance quadrupled), or 360 ft. from 90 m.p.h. (speed trebled and distance multiplied by nine).

In actual fact, brake efficiencies tend to decrease as the speed goes up so that the actual increase in stopping distances is even greater than this theoretical reasoning suggests. A graph shown in Fig. 5 gives a picture of the range of theoretical stopping distances obtained, at various speeds, corresponding to maintained efficiencies of 40, 60, 80, and 100 per cent.

It will be noticed from the formula, the table, and the graph that at any definite speed the stopping distance is inversely proportional to the efficiency; that is to say, if the efficiency is halved, the stopping distance becomes twice as great, and so on.

Stopping Distance and Stopping Time. It is sometimes necessary to know the time which elapses while a car is being brought to a stop. Here again, the figures can be obtained from the table but it is more useful and instructive to know how these are derived. One method is to calculate time from the distance in which a car can stop from a certain speed. The first step is to convert the speed from miles per hour to feet per second. This is easily done, by proportion, if the fact that 60 m.p.h. is equivalent to 88 ft. per sec. is memorized. Thus 30 m.p.h. equals 44 ft. per sec., 20 m.p.h. equals 29·3 ft. per sec., and so on.

Now while a car is being braked to a standstill, the *average* speed at which it covers the distance is approximately half the speed at which the brakes were first applied. If the deceleration is quite steady throughout, our statement is precisely true; in actual fact, deceleration tends to improve as the speed falls but the difference is not very great under normal conditions. Consequently, stopping time can be calculated on this average speed assumption. The best way of showing this is by example.

Suppose that a car is stopped in 40 ft. from a speed of 30 m.p.h. This speed is actually 44 ft. per sec. and

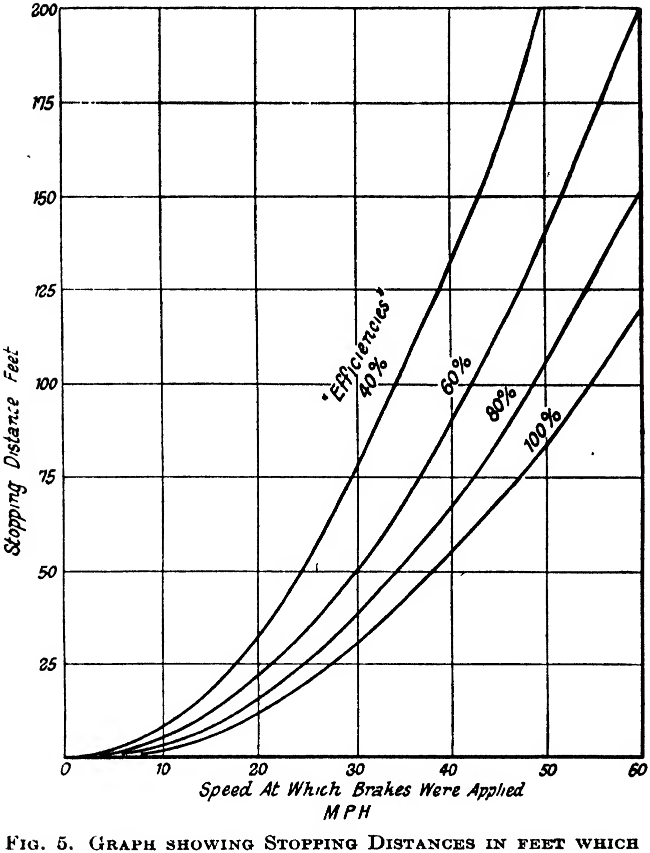

FIG. 5. GRAPH SHOWING STOPPING DISTANCES IN FEET WHICH CORRESPOND TO VARIOUS BRAKE EFFICIENCIES OVER A WIDE RANGE OF SPEEDS

Note the very rapid increase in stopping distance as the speed rises therefore the *average* speed during the braking period is 22 ft. per sec. As in all other calculations involving speed, time, and distance, the time can be worked out by dividing the distance by the speed. The result is therefore $40 \div 22$, or 1·82 sec.

When the efficiency of the brakes is known, the stopping time from any speed can be calculated very easily. As already explained, every efficiency figure represents a definite deceleration, or rate of losing speed, and the handiest way of expressing deceleration is in m.p.h. per sec. This may sound confusing but all that it means is that the car is losing speed at a rate of so many miles per hour during each second of brake application. Thus if the deceleration were 15 m.p.h. per sec., the car would stop in 2 sec. from 30 m.p.h., 3 sec. from 45 m.p.h., and so on, always assuming that the deceleration could be maintained at a uniform figure.

Now on our efficiency scale we agreed that we would select 100 per cent to represent the braking conditions which hold when the retarding force is equal to the weight of the car. The deceleration is then equal to that produced by gravity because, when a weight is thrown into the air vertically, its original speed is being reduced by a force (produced by gravity) equal to its own mass. The deceleration which results is 22 m.p.h. per sec. or, in the engineer's units, 32·2 ft. per sec. per sec.

So, knowing that 100 per cent braking represents a loss of speed of 22 m.p.h. per sec., we can work out the deceleration which corresponds to any other efficiency figure. If the brakes give 50 per cent the deceleration is 11 m.p.h. per sec.; if they give 75 per cent it is 16½ m.p.h. per sec., and so on, in strict proportion.

Summary of Braking Data. At this stage it would seem

CONVERSION TABLE--EFFICIENCY, STOPPING DISTANCES, ETC.

ALL MEASUREMENTS MADE ON LEVEL ROAD

BRAKE EFFICIENCY PER CENT Meter Reading	DISTANCE TO STOP IN FEET From 20 MPH.	From 30 MPH.	From 40 MPH.	TIME TO STOP IN SECONDS From 20 MPH.	From 30 MPH.	From 40 MPH.	RETARDING FORCE Lbs-per-ton	RATE OF RETARDATION Feet per sec per sec	M.P.H. per sec	EQUIVALENT GRADIENTS ASSUMING WHEELS COULD HOLD — HORIZONTAL MEASURE 1 in	Per Cent	ANGLE DEGREES
2	670	1510	2680	45.7	68.5	91.4	44.8	.64	.44	49.8	2	1°-9'
4	335	754	1340	22.8	34.2	45.6	89.6	1.29	.88	24.9	4.02	2°-18'
6	224	503	894	15.3	23	30.6	134	1.93	1.32	16.67	6	3°-26'
8	168	377	670	11.4	17.2	22.9	179	2.57	1.76	12.47	8.02	4°-35'
10	134	302	536	9.1	13.7	18.3	224	3.22	2.2	9.96	10.04	5°-44'
12	112	251	447	7.6	11.5	15.3	269	3.86	2.64	8.26	12.1	6°-54'
14	96	215	383	6.6	9.8	13.1	314	4.5	3.08	7.07	14.14	8°-3'
16	84	189	336	5.7	8.6	11.5	358	5.15	3.52	6.17	16.2	9°-12'
18	74.5	167	298	5.1	7.6	10.2	403	5.8	3.96	5.47	18.3	10°-22'
20	67	151	268	4.6	6.8	9.1	448	6.44	4.4	4.9	20.4	11°-32'
22	61	137	244	4.2	6.2	8.3	493	7.08	4.84	4.43	22.57	12°-43'
24	56	126	224	3.8	5.7	7.6	538	7.72	5.28	4.04	24.7	13°-53'
26	51.5	116	206	3.5	5.3	7	582	8.36	5.72	3.71	26.9	15°-4'
28	48	108	192	3.3	4.9	6.5	627	9	6.16	3.43	29.2	16°-16'
30	44.7	100	178	3	4.6	6	672	9.65	6.6	3.18	31.4	17°-27'
32	41.8	94	168	2.9	4.3	5.7	716	10.3	7.04	2.96	33.8	18°-40'
34	39.4	88.5	158	2.7	4	5.4	761	10.9	7.47	2.76	36.2	19°-53'
36	37.2	83.6	149	2.5	3.8	5.1	806	11.6	7.92	2.59	38.6	21°-6'
38	35.3	79.5	141	2.4	3.6	4.8	850	12.2	8.36	2.43	41.1	22°-20'
40	33.5	75.4	134	2.3	3.4	4.6	895	12.9	8.8	2.29	43.7	23°-35'
42	31.9	71.8	128	2.2	3.3	4.4	940	13.5	9.24	2.16	46.3	24°-50'
44	30.5	68.5	122	2.1	3.1	4.2	985	14.2	9.67	2.04	49	26°-6'
46	29.2	65.6	117	2	3	4	1030	14.8	10.1	1.93	51.8	27°-23'
48	28	62.8	112	1.9	2.9	3.8	1074	15.5	10.56	1.83	54.7	28°-41'
50	26.8	60.3	107	1.8	2.7	3.7	1120	16.1	11	1.73	57.7	30°
52	25.8	58	103	1.8	2.6	3.5	1164	16.8	11.45	1.64	60.9	31°-20'
54	24.8	55.8	99	1.7	2.5	3.4	1209	17.4	11.9	1.56	64.2	32°-41'
56	24	53.8	96	1.6	2.5	3.3	1253	18	12.3	1.48	67.6	34°-3'
58	23.1	52	92.4	1.6	2.4	3.2	1299	18.7	12.8	1.4	71.2	35°-27'
60	22.4	50.2	89.3	1.5	2.3	3.1	1343	19.3	13.2	1.33	75	36°-52'
62	21.6	48.6	86.4	1.5	2.2	3	1389	20	13.6	1.26	79	38°-19'
64	21	47.2	83.7	1.4	2.1	2.9	1432	20.6	14.1	1.2	83.3	39°-48'
66	20.3	45.7	81.2	1.4	2.1	2.8	1478	21.2	14.5	1.14	87.8	41°-18'
68	19.7	44.4	78.8	1.3	2	2.7	1522	21.9	15	1.08	92.8	42°-51'
70	19.2	43	76.6	1.3	2	2.6	1568	22.5	15.4	1.02	98	44°-26'
72	18.6	41.9	74.4	1.3	1.9	2.5	1612	23.2	15.8	.96	104	46°-3'
74	18.1	40.7	72.4	1.2	1.9	2.5	1658	23.8	16.3	.91	110	47°-44'
76	17.7	39.7	70.6	1.2	1.8	2.4	1701	24.4	16.7	.85	117	49°-28'
78	17.2	38.7	68.7	1.2	1.8	2.3	1748	25.1	17.2	.8	125	51°-16'
80	16.8	37.7	67	1.1	1.7	2.3	1791	25.7	17.6	.75	133	53°-8'
82	16.3	36.8	65.3	1.1	1.7	2.2	1838	26.4	18	.7	143	55°-5'
84	16	35.9	63.7	1.1	1.6	2.2	1880	27.1	18.5	.65	155	57°-8'
86	15.6	35.1	62.3	1.1	1.6	2.1	1927	27.7	18.9	.59	169	59°-19'
88	15.2	34.3	60.8	1.1	1.6	2.1	1970	28.3	19.4	.54	185	61°-39'
90	14.9	33.5	59.6	1	1.5	2	2015	29	19.8	.48	206	64°-9'
92	14.6	32.8	58.3	1	1.5	2	2060	29.6	20.2	.43	235	66°-56'
94	14.3	32.1	57	1	1.5	2	2105	30.2	20.7	.36	275	70°-3'
96	14	31.4	55.8	1	1.4	1.9	2150	30.9	21.1	.29	345	73°-44'
98	13.7	30.8	54.7	.9	1.4	1.9	2193	31.6	21.6	.2	492	78°-31'
100	13.4	30.2	53.6	.9	1.4	1.8	2240	32.2	22	.0	∞	90°

(Compiled by Messrs. Tapley, and reproduced by permission)

to be a good idea to sum up the relationships which we have established between all these different ways of estimating the effectiveness of a braking system. At the same time it will be useful to give formulae for calculating one from another.

Efficiency (E)—the ratio between the retarding force produced by the brakes, and the weight of the car on which they act, expressed as a percentage.

Deceleration (d)—the rate at which the braked car loses speed, expressed in m.p.h. per sec. Is always proportional to efficiency, and becomes 22 m.p.h. per sec. when $E = 100$ per cent.

Stopping Distance (D)—measured in feet from the point at which the brakes are first applied. Is inversely proportional to efficiency and is proportional to the square of the speed (V m.p.h.) at which the brakes were applied.

Stopping Time (t)—the duration of brake application to a standstill, measured in seconds. Is inversely proportional to efficiency and is proportional to speed.

The following formulae can all quite easily be derived from the reasoning already presented in this chapter but may be found handy for calculating any of the factors E, d, D and t. As before, the speed in m.p.h. is represented by V.

$$D = V^2/.3E \qquad\qquad d = E/4{\cdot}55$$

$$E = V^2/.3D \qquad\qquad E = 4{\cdot}55d$$

$$t = 4{\cdot}55V/E$$

The Effect of Gradients. So far, all our calculations have referred to a car which is being braked on a level road. This will be the usual condition for brake tests but it is useful to know the effect of a gradient (upward and downward) upon stopping distances, as this is a point often argued when investigation follows an accident. There is also the question of whether a brake which gives a certain efficiency on the level will prove capable of holding the car on a steep hill.

The diagram in Fig. 6 shows a car descending a slope set at an angle to the horizontal of a degrees. The weight of the car (W) acts vertically and can be resolved into a force (R) at right angles to the grade and a force (P) acting down the grade. The former is the load carried by the tyres,

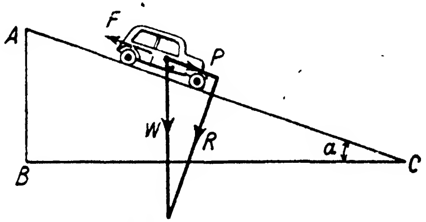

Fig. 6. On a Downward Gradient Gravity Produces a Force (P) which Acts Against the Retarding Force (F) Produced by the Brakes

which is slightly less than their normal load (W); the other force (P) is the one which is tending to make the car gather speed down hill.

If the brakes are applied, they will produce a force acting up the slope which is lettered F in Fig. 6. If F is only equal to P, the car will carry on down the slope at a uniform speed. To bring it to a stop F must be greater than P, and it is obvious that the stopping distance will depend upon the difference between these two forces.

The best way of making the calculation is in terms of efficiency; it is scarcely the right word to use in this connexion but we are already committed to it (see page 11). The triangle of forces, drawn in Fig. 6, shows the relative "sizes" of W, R and P by the lengths of its sides; its angles are obviously the same as those of the triangle ABC which represents the gradient. Thus the ratio P/W is equal to the ratio AB/AC. This is very handy because when we talk of a gradient as being "1 in 5," for example, we mean that the ratio AB/AC is equal to 1/5. So whatever the gradient may be, it gives us the ratio between the forces P and W.

Perhaps it should be added that some civil engineers state gradients as the ratio AB/BC. However, the difference between the two systems is only slight unless the slope is very steep, and therefore need not be taken into account. Students of trigonometry will recognize the two systems as representing the sine and tangent of the angle of slope a.

Getting back to efficiency, so far as stopping distance on the slope is concerned we can take it as the surplus braking force (F minus P), divided by the weight of the car W. This is the same as the subtraction sum $(F/W) - (P/W)$. All that we have to do, therefore, is to express the gradient as a percentage and subtract it from the efficiency obtainable on the level. For example, if a car will normally give 75 per cent but is being braked down a slope of 1 in 5, we must deduct 20 per cent, leaving 55 per cent as the net efficiency available. Reference to the table shows that the corresponding stopping distances, from 30 m.p.h., are roughly 40 ft. on the level (75 per cent efficiency) and 55 ft. down the hill (55 per cent net "efficiency").

The same argument applies to a car which is being braked when climbing a hill, as shown in Fig. 7. The gradient, expressed as a percentage, is added to the efficiency figure which the brakes will give on the level. Thus in the case just quoted, but with the car climbing 1 in 5, the "apparent efficiency" becomes 95 per cent (75 plus 20) and the stopping distance from 30 m.p.h. is reduced to 32 ft.

What are we to do if the sum gives us over 100 per cent, due to the help of gravity on a steep ascent? The table does not go beyond this limit but, as stopping distance is always inversely proportional to efficiency, it can easily be worked out. For example, at 120 per cent the distance is simply half the figure which the table gives for 60 per cent, and so on.

The relationship between gradient and efficiency figures makes it possible to say whether or not a car which gives

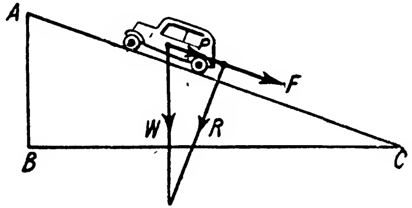

FIG. 7. ON AN UPWARD GRADIENT THE FORCE (P) PRODUCED BY GRAVITY ASSISTS THE RETARDING FORCE OF THE BRAKES (F)

a certain result on a level-road test can be held by the brakes on a steep slope. Thus brakes which will give 25 per cent on the level should retain a vehicle on a slope of 1 in 4. The service system, unless very badly in need of attention, will always be able to look after steep gradients but this is not by any means always true of hand brakes. One point which needs explanation is that some types of brake are appreciably more effective when running forwards than when in reverse. Consequently, the efficiency ascertained by a level-road test does not necessarily apply when the car has to be held from running backwards down a slope. For this reason hand brakes should always be adjusted to give as big a margin as possible.

Transfer of Weight when Braking. The front and rear axle weights of most modern cars, when carrying two or three people, are approximately equal; that is to say, if the car is level and has the front wheels on one weigh-bridge and the rear wheels on another, the same load will be recorded at each end. Remembering that we want to utilize all the adhesion available from each wheel under emergency braking conditions, it would seem sensible to arrange the mechanism in such a way that the front brakes and rear brakes should be equal in effect. This is usually called a "50–50" distribution of front and rear braking; it is used on quite a large number of cars.

When the brakes are applied to a car on the move, however, the front and rear wheels no longer carry equal loads. In actual fact, the greater the deceleration the bigger is the transfer of weight from rear to front. Thus the load holding the rear tyres in contact with the road is reduced while the load on the front tyres is increased to exactly the same extent. It is a "throw forward" effect similar to that which is felt by the passengers in a car when the brakes are forcibly applied.

This transfer of weight during braking accounts for the fact that rear wheels can usually be locked much more easily than front wheels. Some makers distribute the braking front/rear in the proportion 55/45; others have used 60/40. The aim is to get a better emergency stop by postponing the locking point of the rear wheels and by utilizing the bigger retarding effect made possible by the increased adhesion of the front wheels.

Just how much weight is transferred depends mainly upon three factors: the ratio of retarding force to car weight ("efficiency," or deceleration), the height of the centre of gravity, and the length of the wheelbase. The amount transferred is increased in proportion to deceleration; apart from this, it is greatest in a high-built car with a short wheelbase and least in a low-built car with a long wheelbase.

This is made clear by the diagram in Fig. 8 which shows, as an example, a car weighing 3000 lb., this weight being equally distributed under normal conditions. The centre

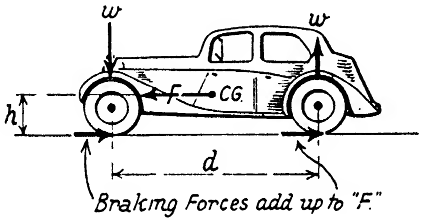

FIG. 8. WHEN A CAR IS BRAKED THE LOAD ON THE REAR WHEELS IS DIMINISHED WHILE THAT CARRIED BY THE FRONT WHEELS IS INCREASED TO AN EQUAL EXTENT (see text)

of gravity or point of balance (*C.G.*) is therefore midway between the axles and, for a car of average build, it will be about 2 ft. above the road surface.

When the brakes are applied, the retarding force (F) acting upon the car as a whole is on the road surface because it takes effect through the tyre treads where these meet with the road. Now the car is trying to carry on without loss of speed and its forward "urge" produces a force through the centre of gravity which is equal and opposite to a retarding force. The distance separating these forces is equal to the height of the *C.G.* (h); their effect on the car is to increase the front axle load by w lb. and to reduce the rear axle load by the same amount. Representing the wheelbase, or distance between axles, by d, the principle of leverages shows that, taking each pair of forces and the distances separating them, the following equation must hold good—

$$F \times h = w \times d$$

Now in the car which we have taken as an example, the ratio h/d is 1/5 so that this equation indicates that the weight transferred will be one-fifth of the retarding force. Thus if the brakes give an efficiency of 80 per cent under emergency conditions, the retarding force will be 80 per cent of 3000 lb. (2400 lb.) and the transfer of weight will be 480 lb. Consequently, the load on the front wheels will increase to 1980 lb. and that on the rear wheels will be reduced to 1020 lb. This is a very different state of affairs from the normal load of 1500 lb. carried by each pair of wheels.

Such is the extreme condition, but after all it is seldom that an efficiency greater than 30 per cent is called upon in the ordinary course of driving. A similar calculation shows that for 30 per cent the retarding force is 900 lb. and the weight transferred is only 180 lb., so giving front and rear axle loads of 1680 and 1320 respectively. The car designer has therefore to decide what brake distribution to adopt in order to get the best compromise, remembering that if he makes the front brakes very much more effective than the rear brakes he will get uneven lining wear (and a slight risk of locking the front wheels on slippery roads) in return for a somewhat shorter stopping distance under emergency conditions.

Incidentally, these are the reasons which have prompted the development of braking systems in which the front/rear distribution varies automatically according to the amount of load applied to the brake pedal; this arrangement, however, has not been at all widely used. However, in the majority of systems the distribution is fixed and anyone attempting to adjust the brakes on a testing machine should ascertain, from the car maker's instruction sheets, the ratio for which the mechanism is designed.

Physical Effort on Brake Pedal. Having described at some length the action of the brakes and their effect upon the moving car, it seems logical to give a little attention to the driver whose physical effort operates the braking system. Various investigators have made tests upon men and women which go to show that, for the short period of two or three seconds involved by emergency stops on the road, anyone can hold a pedal effort of 100 lb. without difficulty. Many people find it easy to hold 150 lb. while some can range up to 300 lb. or even more.

These figures are interesting when tests are being made in the garage using a spring-loaded strut, to produce and record the force on the pedal, while employing a testing machine to ascertain the braking effect which results. Any system which will give an efficiency of 70 per cent in return for a force somewhere between 100 lb. and 150 lb. will be found very satisfactory by the majority of drivers. A higher effort is apt to be tiring. A lower effort is disconcerting, until the driver gets used to it, because he is liable to apply the brakes much more forcibly than he had intended.

A correct seating position is very important for proper brake control, as can be gathered by studying Fig. 9, which shows the toggle action of the human leg. When properly located, quite a small downward muscular effort at the knee, tending to straighten the leg, will provide a big force on the pedal without fatigue. Conditions are not nearly so favourable when the driver's leg is unduly bent, by a seating position which is too close to the pedal, because the toggle effect is reduced. At the other extreme, the disadvantage of being too far from the pedal is obvious but is nevertheless overlooked at times.

A very important aid to good judgment in braking is that the response obtained should be in proportion to the

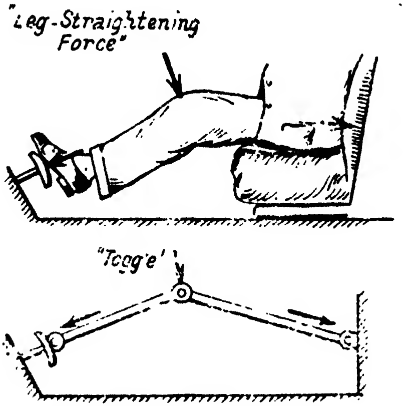

FIG. 9. THE TOGGLE ACTION OF THE HUMAN LEG COMPARED WITH AN EQUIVALENT MECHANISM

(This explains why a correct seating position is important to proper pedal control)

effort exerted. In other words, the retarding force produced by the brakes should increase in step with the force with which the pedal is operated. Thus if 50 lb. gives an efficiency of 30 per cent, 100 lb. should give 60 per cent and 150 lb. should give 90 per cent. No braking systems achieve this ideal, but some approach it much more closely than do others.

In a well-designed system, maintained in good condition, the force required on the pedal before any braking effect is produced will be quite small. This force is required to overcome friction, the return springs, etc., to bring the shoes into contact with the drums. If a poor design, or stiff operating gear, increases this initial effort to any marked extent, it becomes more difficult to use good judgment in brake application.

The Engine as a Brake. Engine braking is very little used nowadays, except on long and steep descents, but deserves a short mention before concluding this chapter. Whenever the accelerator is released, with the car on the move, a retarding effect is obtained because the engine wants to idle but the drive from the back axle forces it to revolve at a considerably higher speed. So the engine exerts a braking effect through the rear wheels, and this effect can be increased by engaging a lower gear in order to raise the engine revs.

It is therefore obvious that the engine can be used to relieve the brakes when descending hills but it is not quite so easy to appreciate the fact that the engine does not assist emergency braking on the level, at any rate when top gear is engaged. The point is that modern brakes slow a car very quickly and, of course, the engine is decelerated with the same rapidity. Consequently, it loses revs. just about as rapidly as would be the case if it were left to slow down on its own account against a closed throttle. Because the "natural rate" of losing revs. is very nearly the same as the rate enforced by the loss of car speed, the engine has very little effect either way. Tests have shown that there is little or no difference in emergency stops whether the clutch is engaged or otherwise.

CHAPTER II

BRAKE TESTS ON THE ROAD

THE subject of brake testing can conveniently be divided into two parts according to whether the tests are to be made on the road or in the garage. A variety of methods is in common use in both cases, ranging from a very simple procedure (in which the judgment of the tester is the most important factor), up to the use of elaborate equipment of various kinds.

The object of the present chapter is to describe the principles and practice involved in testing brakes on the road. In doing so we shall make use of a considerable amount of the information given in Chapter I on the different methods of expressing brake performance in terms of efficiency, stopping distances, and so on. Tests made in the garage, and the testing machines employed for such tests, are dealt with in detail in Chapter III.

Simple Road Tests. The mechanic who is trying to adjust the brakes of a car without any special equipment will first of all endeavour to obtain the right clearance between the shoes and the drums and will then take the car on the road to find out whether he has achieved a balanced result. Thus he may depress the pedal forcibly and notice whether the car swerves to the left or right, or he may try to judge the action of each brake by examining the marks left by the tyres on the road surface. Then the brake shoes are individually adjusted and the tests repeated.

It is not denied that an experienced man can get good results in this way. However, it is at best a "hit or miss" method and, apart from the operator's judgment, does not indicate the effectiveness of the brakes when finally adjusted. Perhaps the most awkward point is that there is little to show what distribution of braking has been obtained between front wheels and rear wheels. True, if there is too much effort at the back, the rear wheels will indicate this by locking very easily, but one can have too much effort at the front without any strong indication of the fact. As a rough and ready means for checking the results, some people brake the car down hill and then test the temperature of each drum by hand—but the human hand is not a very accurate thermometer.

Broadly speaking, these are the reasons which account for the fact that so many service depots and garages have installed brake testing equipment during the past few years. It is sometimes necessary to make a final check, on the road, of the balance obtained between nearside and offside brakes, but apart from this the whole job can be done in the garage, quickly and accurately.

Continuing with simple road tests, there is the "stopping distance" trial often made to ascertain the effectiveness of the brakes. It usually takes the form of driving the car at a steady speed and then endeavouring to apply the brakes forcibly at the instant of passing a post or other landmark. The stopping distance is then measured back from the car to the post. If the driver aims at brake application when he himself passes the landmark, the measurement should of course be taken from the position of his seat, not from the front or rear end of the car.

Although it has been so widely used, the test described is most inaccurate and unsatisfactory. This is generally recognized by the police of the many countries which enforce brake tests on cars selected at random; they are now using decelerometers of various kinds for the purpose. In Great Britain such powers were given to the police as from 31st May, 1937, but at the time of writing there has been no indication of the standard of efficiency which will be accepted nor of the equipment which will be used in making the tests.

The term "reasonable distance" is mentioned (see page 2) but this does not necessarily imply that distance will

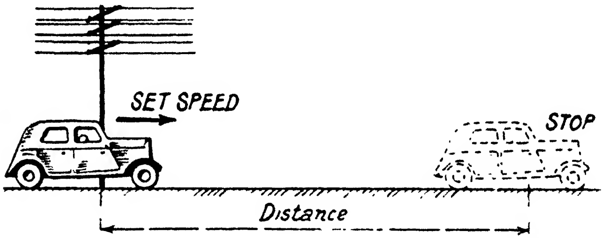

FIG. 10. AN ELEMENTARY FORM OF BRAKE TEST, OFTEN USED BUT SUBJECT TO INACCURACIES AND DIFFICULTIES

The aim is to apply the brakes when passing a landmark at a set speed, afterwards measuring the stopping distance

be measured directly by police testers. From decelerometer readings the equivalent stopping distance from any speed can be calculated, as will have been realized from the account of "efficiency" given in the preceding chapter.

Getting back to attempts to measure stopping distance directly on the road (Fig. 10), the shortcomings of the test deserve a little explanation. First there is the difficulty of applying the brakes at exactly the right instant; it becomes more and more difficult to do so as the speed goes up. Even at 30 m.p.h.—a speed often used—a car is travelling at 44 ft. per sec., so that if the driver is only

$\frac{1}{10}$ sec. "out" in applying the pedal, the braking distance which he subsequently measures will be either 4·4 ft. too long, or 4·4 ft. too short, according to whether he was late or early in relation to the landmark. As the distance is often only 40 ft., these figures represent a plus or minus error of over 10 per cent each way. And not many drivers can get within $\frac{1}{10}$ sec.

Next there is the matter of speed, which should be exactly right at the instant of brake application. It is difficult to hit it off on the speedometer, and the speedometer itself is usually inaccurate. The error involved is large because stopping distances vary in proportion to the square of the speed. Thus, suppose that a car is actually stopped from 27 m.p.h. in 40½ ft. but the driver thinks that he has obtained this result from 30 m.p.h. He looks up a table, such as the one on page 17, and thinks the efficiency is 74 per cent. In actual fact, calculated in proportion to the square of the speed, the stopping distance from a genuine 30 m.p.h. would have been 50 ft., equivalent to an efficiency of only 60 per cent.

A third difficulty met with in measuring stopping distances is to be sure that the road selected for the test is really level. A gradient which does not appear to be severe may nevertheless make an appreciable difference to the results.

Stop Watch Tests on the Road. As an alternative to the measuring tape, a stop watch can be used in order to record the time which elapses from the instant of brake application to the final stop. As already explained, if a car can be pulled up in a certain time from a certain speed, the corresponding stopping distance and efficiency can be calculated or obtained from a table. However, although this method avoids the difficulty entailed in applying the

This plan avoids some of the errors previously described but it is not so easy as it may sound to get the pistol timed to the right instant. Furthermore, the effect of speed errors remains, and the apparatus is rather clumsy to fit to a car.

Some very interesting results were obtained by using two pistols of this kind, one connected to the brake gear and the other operated by an observer seated in the car. The driver was asked to maintain a steady speed of 30 m.p.h., and he had to apply the brakes as rapidly as possible as soon as the observer fired the hand-controlled pistol. In this way two marks on the road were obtained and the distance between them represented the lag which had occurred from the signal to the point at which the brakes were applied. After this, the braking distance could be measured from the car to the second mark in the usual way.

The interesting fact disclosed by these tests was that even an expert driver, alert and expecting a test, required ½ sec. in which to respond and to transfer his foot from accelerator to brake pedal. At 30 m.p.h. the car covered 22 ft. during this period, as measured between the two marks on the road. The subsequent stopping distance was about 40 ft. so that the driver's reaction period added over 50 per cent to the total distance needed to pull up. Other tests have shown that less expert drivers, not fully alert, may need as much as a full second in which to react and apply the brakes. These things cannot be taken into account in carrying out tests of the brakes themselves, where we are interested only in the efficiency of the mechanism, but they are worth bearing in mind when investigations follow accidents, and the probable speed at which a car was travelling is called into question.

Simple Forms of Decelerometer. In the preceding chapter we explained the meaning of the term "brake efficiency" as it has come to be accepted—namely, the ratio between the total retarding force exerted upon a car, and the weight of that car, expressed as a percentage. We also found that every efficiency figure represents a definite deceleration, or rate of losing speed, no matter what the size of the vehicle may be. Thirdly, we discovered that from any stated speed the stopping distance and stopping time which correspond to any efficiency figure can be calculated.

Apart from the obvious handiness of this efficiency idea for making comparisons of brake performance, there is the matter of how best to test brakes on the road; we have already seen the difficulties involved in trying to make direct measurements of distance and time. The question that arises, therefore, is whether it is possible (and preferable) to measure the deceleration of a braked car directly by some form of apparatus designed for the purpose.

This, in brief, is the function of a decelerometer and there is no difficulty in calibrating such a device so as to read brake efficiency straight off a scale. It is true that possibilities of error are present in all such mechanisms but, when they are handled by people of experience who really understand the principles involved, they give results which represent a marked advance over those obtainable in any other way on the road.

One of the simplest forms of decelerometer is shown in Fig. 12; it consists of a glass tube, mounted in such a way that its angle of slope can easily be adjusted, and within the tube there is a loose-fitting steel ball. A succession of brake tests is made with the tube set at different "gradients" until the angle selected is such that the ball just rolls very gently up the tube as the car slows down.

It is not easy to get this result quickly, for which reason the apparatus has not been developed commercially, but it provides a good example of the principle of measuring decelerations and is capable of giving accurate results.

If the slope of the tube is not great enough, the ball will shoot up to the top when the brakes are applied because it wants to "carry on," at undiminished speed, although

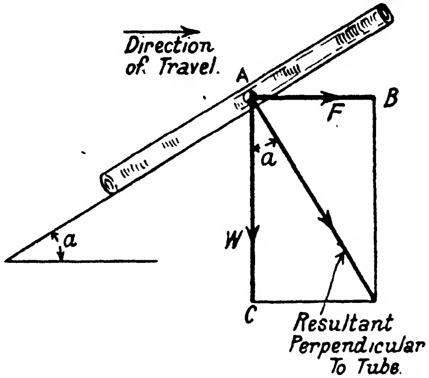

FIG. 12. A SIMPLE FORM OF DECELEROMETER CONSISTING OF A STEEL BALL IN A GLASS TUBE OF ADJUSTABLE SLOPE, SHOWING THE FORCES ACTING ON THE BALL WHEN THE BRAKES ARE APPLIED TO AN EXTENT JUST SUFFICIENT TO HOLD THE BALL AGAINST ROLLING

the car and tube are losing speed. If the slope is too steep, the forward "urge" on the ball will not be great enough to overcome gravity and it will stay at the bottom. When the angle is correct, the gravitational pull down the tube and the desire of the ball to roll forward will be just about in balance.

The forces which are actually opposing one another are as shown in Fig. 12, and their magnitude is proportional to the sides of the rectangle by which they are represented. Now the mass of the ball is in just the same case as the mass of the car; it is experiencing an identical deceleration, the amount of which is proportional to the ratio between the forwardly acting force (F) and the weight on which it acts (W). Consequently, the efficiency of the brake is simply represented by the ratio F/W, multiplied by 100, as for the complete car. From the shape of the rectangle this ratio is seen to be the tangent of the angle a, which is also the angle of slope of the tube. So that, having set the tube to a slope which justs allows the ball to roll upwards during braking, the efficiency of the brakes can be measured by the ratio of the sides of the triangles AB/AC, or by trigonometrical tables giving the tangent of the angle a. For an efficiency of 100 per cent the angle will become 45 degrees and the sides of the triangles (AB and AC) will be equal.

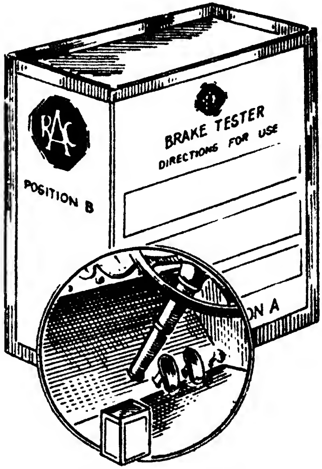

Fig. 13. The "Mustard-tin" Decelerometer, Developed by The Technical Department of the Royal Automobile Club, which Falls Over when a Certain Standard of Braking is Reached

The R.A.C. Brake Tester. Another simple illustration of the decelerometer idea is provided by the so-called "mustard-tin" brake tester brought out by the technical department of the Royal Automobile Club not long ago. It is a tin of rectangular section, carefully made to certain sizes, which can be placed on the floor of the car either sideways or longways, as shown in Fig. 13. The brakes are applied and, if they will tip the tin so that it falls forwards, their efficiency is shown to be above a certain standard. Thus if the tin tips forward from its sideways position, the efficiency is above 46 per cent; if it tips from the longways position, the efficiency must be at least 70 per cent. In this way a very simple check of brake performance is

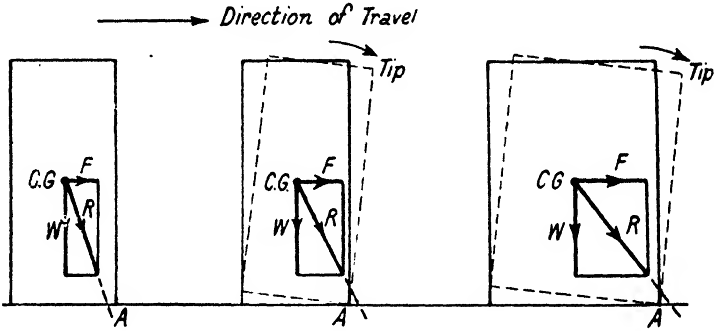

FIG. 14. FORCES WHICH ACT ON THE R.A.C. TESTER

Left to right: Low-efficiency position with braking insufficient to cause tipping; position as before but tipping caused by more effective braking; high-efficiency position with braking sufficient to cause tipping

obtained although the device cannot be used to measure intermediate efficiencies.

The forces acting on the tin, like those imposed upon the ball in the tube, are its downward weight (W) and the throw-forward effect (F) produced by the deceleration. These are shown in Fig. 14, and in the first diagram it is assumed that the brakes are poor so that the force (F) is small. It acts through the centre of gravity ($C.G.$) of the tin, as does also the weight (W), and if a rectangle is drawn with its sides representing the forces (F and W) to scale, their resultant effect upon the tin is represented by the diagonal (R). If this diagonal falls short of the edge of the tin (A) it will not tip forward, whereas if it falls ahead of this edge the tin will tip forward. For this reason the fact that the tin can be made to tip shows that the brakes produce a ratio F/W which is above a certain limit, and that

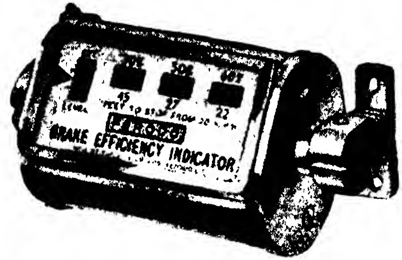

FIG. 15. THE FERODO BRAKE EFFICIENCY INDICATOR
Signals appear in the windows when the braking efficiency reaches 30 per cent, 50 per cent, and 60 per cent respectively

their efficiency is also above this limit. Another diagram in Fig. 14 indicates the increase in the ratio F/W which is needed to make the tin tip when placed longways.

The Ferodo brake efficiency indicator uses the ball and tube principle but is arranged to give just three readings; it is shown in Fig. 15. If the brakes will produce a signal in the first of the little windows, their efficiency must be above 30 per cent: to give a signal in the second and third windows they must do better than 50 per cent and 60 per cent respectively. The signals remain visible until restored to zero by operating a small lever. Before making a test, the instrument must be set with the car on a level road;

an indicator worked by a pendulum shows when it has been placed at the correct angle.

Each signal is controlled by a tilted tube, fixed to a pivot and containing a metal ball; the three tubes are set at different angles. When the braking force is just enough to cause a ball to roll forwards up the slope, the tube becomes tipped when the ball passes the pivot and so brings the signal up into position opposite the appropriate window.

Pendulum Decelerometers. Another way of measuring deceleration is by means of a pendulum, and this is rather an important principle because it is employed in the Tapley brake meter; an instrument widely used for police tests, and by car and brake manufacturers, in many countries. Imagine a pendulum hanging from the roof of a car, as shown in Fig. 16. When the brakes are applied, the bob weight will swing forward because it tries to carry on at undiminished speed just like the metal ball in the inclined tube. If the deceleration is steady, the pendulum will remain at a certain angle where the weight of the bob, and the forward force due to the deceleration, are in equilibrium. Their resultant, represented by the diagonal of a rectangle drawn to represent the forces, will then fall in line with the pendulum itself.

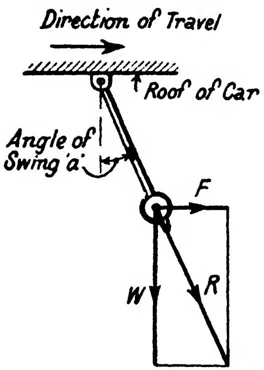

FIG. 16. SHOWING THE FORCES WHICH ACT UPON A PENDULUM DECELEROMETER WHEN THE BRAKES OF THE CAR ARE APPLIED

Just as in the case of the inclined tube decelerometer, the efficiency of the brakes is represented by the ratio F/W and this in turn is the tangent of the angle of swing a. So a scale could be placed behind the pendulum on which the brake efficiency indicated by any extent of swing could be read off directly.

An interesting point about this device is that it is to a large extent self-correcting if the brakes are being tested on a road which is not quite level. For example, on a downward gradient, the pendulum will set itself ahead of the

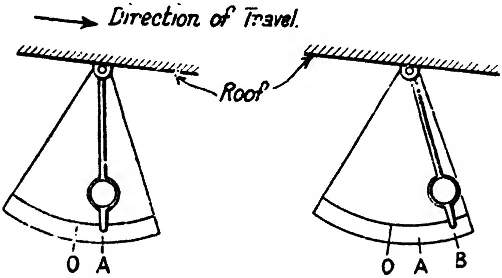

FIG. 17. FOR REASONS EXPLAINED IN THE TEXT, THE PENDULUM DECELEROMETER IS LARGELY SELF-CORRECTING WITH REGARD TO GRADIENT

zero mark on the scale when the car is either stationary or moving at a steady speed. When the brakes are applied, it will move farther along the scale (from position A to position B, Fig. 17) by a distance representing the deceleration experienced. However, this deceleration will be less than the normal figure which the brakes would produce on a level road by an amount depending upon the steepness of the slope. As the distance from zero to position A also depends upon the steepness of the slope, a compensation is obtained automatically. In actual fact, except on really steep gradients, the compensation is almost exactly right so that the position of the pendulum (*B*) is just about the same as that which it would occupy during a test on the level.

Conversely, on an upward slope, the pendulum sets itself at some point behind zero. When the brakes are applied, it has to move from this point to zero, and then goes on beyond zero to its final position. However, the deceleration is now improved by the gradient, to offset the initial "handicap" of starting behind zero, so that once again the final result is just about the same as on the level.

One kind of error which is unfortunately unavoidable, when using any form of pendulum or sloping tube decelerometer, is caused by the pitching of the car on its springs during brake application. As a rule, the car tips forwards and carries with it the body of the brake testing instrument. The result is to shift the scale relative to the recording device in such a way as to add to the true efficiency reading. In the rare cases when the car tips upwards at the front—found in special forms of independent front suspension—the reading obtained is slightly less than the true reading.

In the majority of cars the error caused by pitching is not very great, but this is a point which needs watching in certain cases where "nose-diving" is considerable. It has sometimes been suggested that the difficulty could be overcome by mounting the decelerometer on one of the axles but, apart from the inconvenience of this position, the axle itself is apt to turn under the effect of brake torque. Probably the best way of avoiding error, where extremely accurate results are desired, is to trail the decelerometer on some kind of bicycle-wheel attachment of the kind sometimes used to obtain an independent drive for a calibrated speedometer.

The Tapley Brake Meter. As already mentioned, this well-known efficiency indicator works on the pendulum principle; fluid damping is provided to prevent the pendulum part from swinging erratically during brake tests.

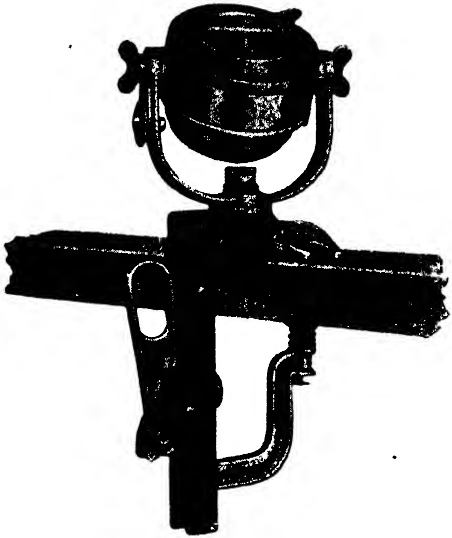

FIG. 18. THE TAPLEY BRAKE METER WHICH IS PROVIDED WITH A QUICK-ACTING CLAMP FOR ATTACHMENT TO A RUNNING BOARD OF THE CAR UNDER TEST

The instrument is so arranged that it can easily be clamped to the car (see Fig. 18), after which it must be carefully set to zero with the car on level ground. A convenient way of making sure that this condition is met is to use the flat surface of a weighbridge. Where this is not possible, and a road which is supposed to be level is employed, the car should be reversed end-for-end in order to check the setting in two positions, up and down. If there is a slight difference due to gradient, an average can be struck midway.

As shown in Fig. 19, the meter consists of a casing which encloses the mechanism and circular scales (7), there being a glass window (8) through which readings are taken. The pendulum (1) is swung on anti-friction pivots (2) inside an inner casing (3) and is completely surrounded by fluid which provides a damping effect. The curved base of this

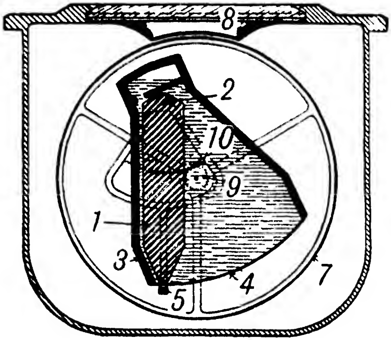

FIG. 19. A CROSS-SECTION THROUGH THE TAPLEY BRAKE METER WHICH SHOWS THE DAMPED PENDULUM AND OTHER PARTS DESCRIBED IN THE TEXT

inner casing is a thin sheet of brass (4) and beneath it there is an iron armature (5) which swings from the same centres as the pendulum. The armature follows all the movements of the pendulum because the latter part is magnetized. As it swings, the motion of the armature is conveyed by a pinion (9) and toothed sector (10), with a ratio of 1 : 6, to the circular scales. An adjuster, which can be operated externally by a key, enables the fluid damping to be supplemented, if necessary, and a second adjustment allows the instrument to be set to zero with accuracy.

There is an external catch which can be placed in either of two positions. When "free" it allows the pendulum to swing to and fro, but when in the operative position it holds the pendulum at the maximum angle reached during the trial. Consequently, when about to make a test, the trigger is set to the second position and, after bringing the car to a standstill, the efficiency can be noted at leisure from one of the scales which is calibrated from zero to 100 per cent. The other scale shows the equivalent stopping distance in feet from 20 m.p.h., but it is not, of course, necessary actually to use this speed when making a test. After a trial the catch is moved to allow the pendulum to return to zero and must be shifted once again before making a second trial.

It is true that the deceleration is apt to fluctuate slightly during the braking period but, as the meter needs about ¼ sec. to respond, it actually strikes an average through the fluctuations and does not record temporary "peaks" in the deceleration curve. On the other hand, if the brakes are of a kind which tend to be much more effective near to a standstill than at higher speeds, the meter may not give a true average because it will indicate the maximum deceleration reached near to the end of the test period. Where this trouble is suspected, it can be checked by applying the brakes at, say, 40 m.p.h. and releasing the brake pedal at about 15 m.p.h. This will eliminate high readings near to a stop, and the results can be compared with those obtained by braking right down to a standstill.

Other Forms of Decelerometer. Although we have dealt with the decelerometers which are most widely used, this chapter would not be complete without mention of other types. One of these (an example is the Siemens) employs a coloured fluid which rises in a vertical tube to an extent depending upon the deceleration produced by the brakes; the calibration is usually in feet per second per second. The conversion to "efficiency" is rather clumsy because 100 per cent is equivalent to the effect produced by gravity, which is the odd figure of 32·2 ft. per sec. per sec. (see page 16). To convert the readings to the efficiency scale it is necessary to divide them by 32·2 and to multiply the result by 100.

The Bendix brake people make a neat decelerometer which registers the equivalent stopping distance from 20 m.p.h., on a card, regardless of the speed from which the test was actually made. There are also costly instruments which are really only required for special research work. The results are recorded as a graph which shows the deceleration from instant to instant, throughout the braking period, on a strip of material driven by a clock mechanism.

"Incredible" Stopping Distances. Extravagant claims are still often made for brake performance and are sometimes used as arguments against (1) the choice of a maximum figure of 32·2 ft. per sec. per sec. (a stopping distance of 30·2 ft. from 30 m.p.h.) as 100 per cent on the efficiency scale; (2) the use of decelerometers in general. It is not denied that this arbitrary limit of 100 per cent has apparently been exceeded in a few rare cases, but, in the vast majority of instances, claims to have beaten the limit are based upon tests of stopping distance of a kind which could not possibly yield accurate results. We have already described the many sources of error involved in such tests.

These errors really provide the best argument in favour of the decelerometer. It is not a perfect instrument, and it needs care and skill in handling to give good results; even so, it does appear to be the best means available to us for checking brake performance on the road.

An exaggeration of the powers of motor car brakes can only do harm by making drivers over-confident. It has always to be remembered that when a car is in service the brakes never give as good a result as is possible when they have just been adjusted. Furthermore, the usual talk of stopping distances omits entirely any reference to the distance covered while the driver is realizing the emergency and acting upon it. As we have seen, this distance may easily prove to be as great as the actual stopping distance.

CHAPTER III

BRAKE TESTS IN THE GARAGE

During the past few years there has been a great increase in the use of brake testing machines in garages and service depots. There is no doubt that they are of great assistance to the mechanic who is effecting adjustments to the car and, in many cases, subsequent correction by road testing is not found to be necessary. Some of the machines will give a good indication of brake performance, either by showing the total retarding force which the brakes produce or by giving results directly in terms of brake efficiency per cent. Efficiency can, of course, be derived from the former test results by calculating the ratio between the retarding force and the weight of the car.

These results have to be treated with a little caution because, in some machines, the retarding force recorded (or the efficiency, as the case may be) is not so great as can actually be reached on the road. The reason is found in slip between the rollers of the machine and the tyres of the car which may set an artificial limit to the readings, particularly in the case of heavy vehicles.

There are two general classes of garage machine which can broadly be named "static" and "dynamic." In the static variety the car is stationary and the machine turns the wheels against the resistance created by the brakes. In the dynamic testers the moving car is braked with the wheels running over four track plates let into the floor, and the drag produced on the plates is recorded.

Before describing some of these testing machines it will be best to deal with the working principles involved in each class—static and dynamic.

Static Brake Testers. The idea behind machines of this kind closely resembles that of the simple and time-honoured trial in which the car was jacked up, the brake pedal held down, and a mechanic "felt" the braking effect by pulling on the periphery of each wheel in turn. In a rough and ready way he was actually measuring brake torque, and that, in effect, is the result achieved by the static tester, albeit in a much more satisfactory fashion.

When a machine is employed to replace human effort, the

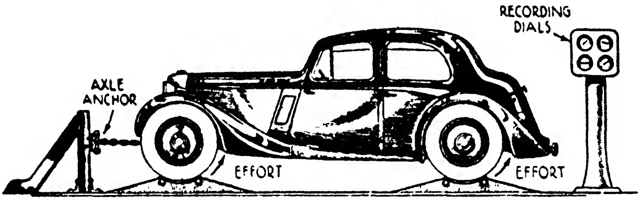

FIG. 20. IN THE STATIC TYPE OF BRAKE TESTER THE WHEELS OF THE CAR ARE DRIVEN BY ROLLERS WHILE THE BRAKES ARE APPLIED
(Sketch reproduced by courtesy of "The Motor")

torque which can be applied to the braked wheel becomes very much greater, and an accurate means of measurement replaces guesswork. As a rule, the machine is so arranged that the tyres of the car rest upon rollers which are driven by electric motors (Fig. 20). There may be two sets of these rollers, so that a pair of brakes on either axle can be tested simultaneously, but in the more popular (and more expensive) lay-outs four sets of rollers are employed. Apart from the facility of testing all four brakes at once, better results are obtainable. An anchorage prevents the car from being driven backwards by the rollers.

As explained in Chapter I, and illustrated in Fig. 1, page 4, the drag on the track of the tyre which is needed to turn the wheel against the brake is equal to the retarding force which the brake applies to the car. Some of these static machines measure the drag in pounds, while others are ingeniously arranged to measure both the drag and the weight on the wheel, and to give a result which represents the ratio between the two. The methods employed are best described in relation to the machines in question.

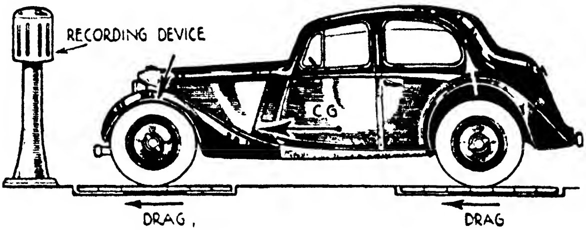

Fig. 21. The Alternative Dynamic Form of Brake Tester Records the "Road Drag" Produced by the Braked Wheels of a Moving Car
(Sketch reproduced by courtesy of "The Motor")

The dynamic type of tester, as already explained, records the drag produced when a moving car is braked (Fig. 21) and really reproduces in a more elaborate form the simple idea of the spring balance and strip of "road" which was illustrated in Fig. 2 (page 5). Tables are usually provided from which brake efficiency can be read off when the test results and the weight of the car are available.

Owing to the transfer of weight from rear axle to front axle, which occurs when a moving car is braked, the distribution of the retarding forces as recorded by a dynamic tester differs from that which is recorded on a static tester. Other discrepancies may occur through the fact that the operating gear is affected by relative movements between the axles and the car, during braking with the vehicle in motion, which are not reproduced in the course of a static test. On the other hand, for brake maintenance purposes, the static test is usually to be preferred. Apart from enabling the brakes to be closely adjusted for equalized action, the static machine shows at once any fluctuations of retarding force, which may be caused by an oval drum,

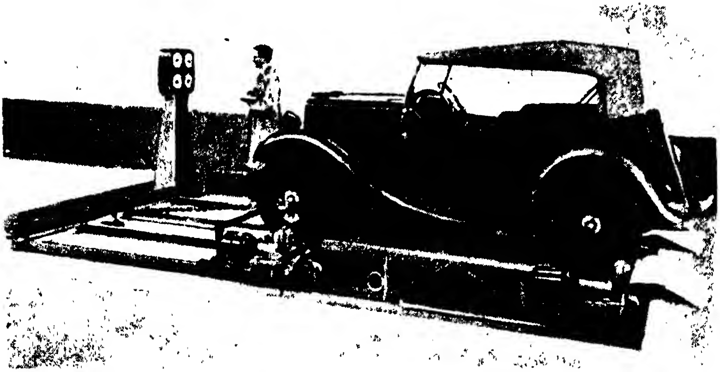

Fig. 22. The Bendix-Cowdrey Brake Tester of the Universal Type Tests all Four Braked Wheels at One Time and Records the Results on Grouped Dials

or any tendency for one of the brakes to drag after the pedal has been released.

The Bendix-Cowdrey Brake Tester. Although called by the makers a "dynamic" tester, the Bendix-Cowdrey machine is properly classified as "static" by the definitions given in this chapter: thus the car is stationary although the wheels revolve. Each tyre rests upon a pair of rollers, coupled by a chain and sprockets, one of which is driven by an electric motor through gearing. This gearing is mounted in a cradle, and the apparatus is so arranged that the torque reaction on the cradle can be measured. By a suitable calibration the scale which "weighs" the drive is able to show, in pounds, the retarding force applied to the wheel; this is, of course, equal to the force at the tyre tread needed to turn the wheel against the brake.

In the "Universal" machine (Fig. 22) four pairs of rollers are provided, with a convenient arrangement of controls, and there are four grouped dials to show the retarding effect of each brake; they read up to 1500 lb. per wheel so that the machine will handle any vehicle up to a weight limit of about 6000 lb. It can be adjusted to suit variations of wheelbase. Charts are provided to enable stopping distances from various speeds to be ascertained from the known weight of the vehicle and from the sum of the retarding forces shown on the dials. Incidentally, as in all other static brake testers, provision is made for anchoring the vehicle; without this, the drive on the braked wheels would move it backwards off the rollers. There are various other types, ranging up to very large sizes for heavy trucks.

The Heenan and Froude Brake Tester. This is another well-known static machine, the outstanding feature of which is that it automatically obtains the ratio between the retarding force on each wheel and the load carried by that wheel; in other words, it gives direct readings of brake "efficiency." There is therefore no need to know the weight of the vehicle. Each wheel is carried by fluted rollers (1, 2), the spindles being supported in a carrier (3) located by links (4, 5), as shown in Fig. 23. The rollers are driven by an electric motor through a chain (6). Torque reaction at the tyre makes the rollers swing away from the normal position, beneath the centre of the car wheel, against a

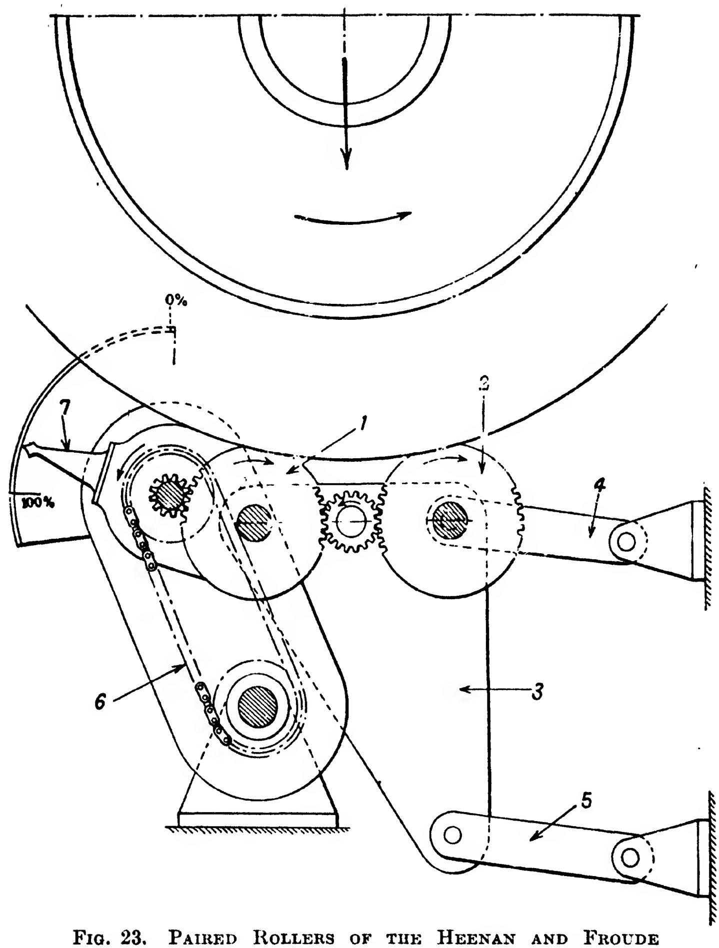

FIG. 23. PAIRED ROLLERS OF THE HEENAN AND FROUDE BRAKE TESTER WHICH RECORDS THE RATIO BETWEEN RETARDING FORCE AND WHEEL LOAD (see text)

In an alternative and earlier design a single roller is used, but the principle is the same spring (not shown). On the other hand, the weight on the wheel tends to shift the rollers in the opposite direction.

The net result is that when this machine is driving a braked wheel the position taken up by the rollers depends upon the ratio between braking force and downward load and this ratio is shown on a scale in terms of efficiency per cent. A pointer (7) "stays put" at the maximum reading attained.

The machine is supplied with eight rollers, to test all the brakes simultaneously, and a depresser can be obtained with which a definite and constant load can be applied to the brake pedal. A neat refinement is available in the shape of an illuminated chart on which the efficiency of each brake is clearly recorded. Lines on the chart help the operator to secure the best possible adjustment, while curves enable him to read the stopping distances which correspond to the efficiencies recorded by the machine.

The G.E. Brake Tester. This machine is made by Joseph Bradbury & Sons Ltd., a concern well known for equipment used in servicing brakes. It is normally arranged to take all four wheels of a car, but a two-wheel model is also available. Each unit comprises a pair of rollers, to carry the tyre, driven by an electric motor: brake effort is recorded on an hydraulic pressure gauge calibrated to read from 100 lb. to 1600 lb. In the four-wheel machine the gauges are conveniently grouped with a control by means of which all four pointers can be thrown back to zero. To interpret the results in terms of brake efficiency the sum of the readings must be divided by the weight of the vehicle.

The Tecalemit Tester. This is an ingenious static machine which, like the Heenan and Froude tester, shows brake efficiencies directly although the result is obtained in a different manner. Each of the car wheels is supported by a group of rollers, driven electrically and mounted in a trolley. The trolley wheels rest upon rails which are of a specially curved shape. The car is anchored against endwise movement in the usual way.

When the drive is exerted upon the braked wheels, by the motor-driven rollers, torque reaction forces the trolleys to creep forward and, in order to do so, they have to climb the curved rails. At a certain point a balance is reached between the weight on the wheel, tending to push the trolley backwards, and the reaction which is pushing it forwards. The position of the trolley therefore shows the efficiency of the brake and is indicated on a dial; four such dials are of course provided.

The Bean Brake Tester. This clever machine, which is of American origin, is handled in England by E. P. Barrus

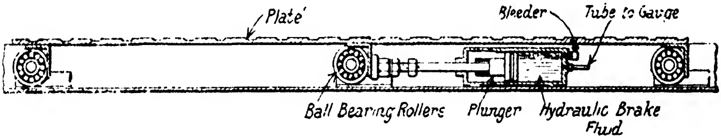

FIG. 24. IN THE BEAN BRAKE TESTER EACH CAR WHEEL RESTS UPON A PLATE OF THE KIND SHOWN IN SECTION IN THIS DRAWING

Braking force is recorded hydraulically from the drag on the plate

Ltd.; it can be used either for static tests or for dynamic tests. In the former case the braked wheels rest upon four steel plates, with non-skid surfaces, which are carried by rollers mounted on ball bearings. Each plate (Fig. 24) is connected to a plunger working in a cylinder filled with Lockheed brake fluid, and pipe lines connect the cylinders to four pressure gauges.

By means of a pneumatic ram (attached to the front end of the car) the vehicle can be dragged forward, whereupon the gauges show the pull which each braked wheel exerts upon the plate by which it is supported. The ram is provided with a convenient remote control which can be handled by the operator, seated in the car, who is using the brake pedal. It is a two-way control which enables the action of the ram to be reversed in order to shift the car backwards for a re-trial. To obtain brake efficiency the dial readings are added together and then divided by the weight of the vehicle.

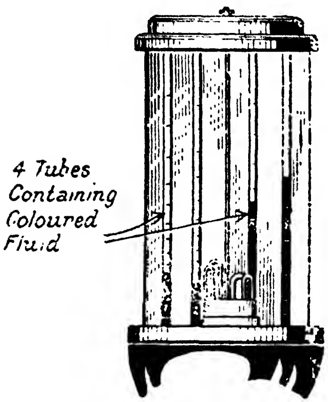

FIG. 25. FLUID RISING IN TUBES IS USED TO RECORD BRAKING FORCES (ON THE CAR WHEELS) IN THE BRITISH WEAVER DYNAMIC BRAKE TESTER

In order to use the machine as a dynamic tester the ram is removed and the car is driven on to the plates at a moderate speed. As the wheels pass over them, the brakes are applied and the forward drag (which equals the braking effort) is recorded by the four gauges. The readings obtained naturally differ from those afforded by a static test on the same machine because of the transference of weight from rear to front which occurs when a moving car is braked.

The British Weaver Brake Tester. This machine is the best known of the dynamic group and provides a ready means for demonstrating to car owners the action of their brakes and any fault requiring attention. It is suitable for installation in the open, and has been put into use by many garages. The car is driven over four steel plates, the brakes being applied while the wheels roll over them, and instead of using dials the drag on each plate is recorded by a coloured liquid which rises in a vertical tube. There are four of these tubes (Fig. 25), built into a tower, and they indicate braking effort in pounds for each of the wheels. A chart is supplied with the tester to facilitate the interpretation of the results in relation to the weight of the vehicle.

This machine is arranged for flush fitting in the floor and has a two-way action so that the car can be driven on from either end. The plates are provided with wire mesh surfaces which give a good grip even under wet weather conditions. Three sizes of machine are available.

It is not claimed that this chapter has exhausted the subject of garage brake testers, as a wide variety of machines is available and continues to increase. However, the leading examples which we have described demonstrate the general principles employed in all testers so far developed for garage use.

CHAPTER IV

FUNDAMENTALS OF THE BRAKING SYSTEM

TEN to twelve years ago a very great variety of braking systems was in current use, each of individual design and requiring special knowledge for its correct maintenance and adjustment. Luckily, the work of service depots and garages has been greatly simplified since those times, by the wide adoption (by car manufacturers) of three proprietary braking systems—Bendix, Girling, and Lockheed, to place them in alphabetical order. Furthermore, in all three systems semicircular shoes of various kinds are employed, arranged to hinge outwards into contact with the inner surfaces of drums which rotate with the wheels. The shoes are mounted in pairs upon back plates, which are secured to the axles, and which serve also to enclose the drums.

Before dealing with the three braking systems in question, it is desirable to devote a chapter to the general mechanical principles involved in obtaining a retarding effect upon the rotating drums, from an effort exerted by the driver upon a pedal.

Two stages need consideration—first, the method used to transfer the pedal effort to the shoes, and secondly, the way in which the loaded shoes act upon the drums. For various reasons it will be easier to select the second of these stages for prior treatment.

Each shoe is faced with a segment of "friction material" which includes in its composition a considerable proportion of asbestos, resistant to heat. So when the shoe is forced outwards, the material attached to the shoe is pressed into rubbing contact with the rotating drum—a part made either of iron or of steel. The first step in our investigation is therefore to discover what arises from frictional contact of this kind

An Experiment with Friction. Suppose an experiment is made with the simple apparatus shown in Fig. 26. The slider, of weight *W* lb., is made from iron (or steel) to

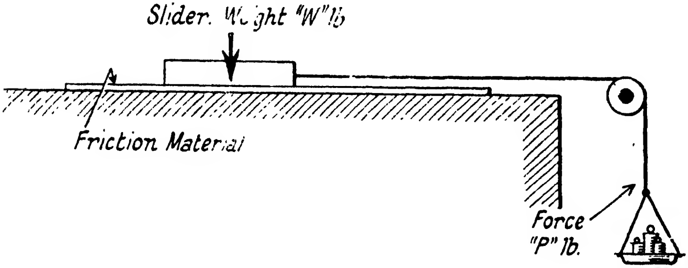

FIG. 26. A SIMPLE EXPERIMENT IN WHICH AN IRON SLIDER REPRESENTS THE BRAKE DRUM AND A FLAT STRIP OF FRICTION MATERIAL REPLACES THE USUAL SEGMENTAL LINING

represent a brake drum, and it moves along a flat, horizontal surface, faced with friction material, when towed by a cord which passes over a pulley. By making a number of tests we might find that a force of *P* lb. would just suffice to keep the slider moving at a steady speed. Then we could assert that the "coefficient of friction" of the fabric against iron is represented by the ratio P/W.

We should find that this coefficient remains steady over a wide range of loads, speeds, and temperatures; that is to say, if the load is doubled, the towing force must also be doubled; if the load is trebled, the force is trebled, and so on. The *ratio* remains the same, just like our old friend the coefficient of adhesion between the tyres and the road. But instead of amounting to 0·8 or (80 per cent) more, the coefficient for brake linings is usually between 0·3 and 0·45, according to its composition. Provided the surfaces are smooth, it makes little or no difference whether the metal against which the material operates is of iron or of steel; the coefficient remains much the same.

Pursuing our experiments with the slider, we might discover that oil or water on the surfaces will greatly reduce the

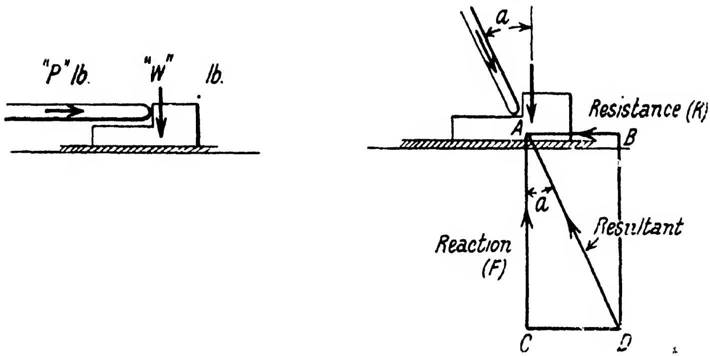

FIG. 27. ANOTHER TYPE OF TEST IN WHICH THE SLIDER IS PUSHED INSTEAD OF PULLED

The view on the right shows that there is a limiting angle for the "push" beyond which the slider cannot be moved no matter how great a force is used

coefficient of friction. Rust or road grit will cause it to increase, temporarily, until they become worn away. These effects apply equally to the brakes of a car.

Another kind of test is shown in Fig. 27. Here the slider is being pushed, instead of pulled, by means of a rod. If the rod is horizontal we shall get the same ratio of P/W as before, although P will now represent the push of the rod instead of the pull of the string.

Next, suppose that the rod is gradually lifted, while continuing to push the slider. We shall find that the force required becomes greater and greater, not because of any change in the coefficient of friction but simply for the reason that a part of the "push" is adding to the load on the slider while, at the same time, only a diminishing proportion of the "push" is available in the form of an effective horizontal force. The really interesting point, however, is that when a certain angle is reached *no amount of "push"* will shift the slider. This is called a self-locking, or jamming, effect and is worth thinking about because of the similar trouble sometimes experienced with brakes.

Throughout the experiment the forces acting on the slider are the reaction at right angles to the surface (F) and a frictional resistance (R) which acts horizontally and is equal to the "push"; these are shown in the second sketch in Fig. 27. The reaction (F) is equal at all times to the downward force on the slider. Now the two forces (F and R) produce a resultant which can be represented by the diagonal of the rectangle $ABDC$ if the sides AB and AC are drawn (to scale) to represent the forces R and F, respectively.

Because the ratio R/F is always the same, it follows that the angle a, at which the resultant acts, is always unchanged throughout the experiment. So when the rod with which we are pushing comes in line with the resultant force, or goes beyond, we can no longer shift the slider no matter how hard we may try. There is, in fact, only one way out—namely, to reduce the coefficient of friction, so as to alter the ratio R/F, and with it the angle a. Even so, if we continue to increase the slant of the rod another self-locking point will be reached, corresponding with the new coefficient.

The angle *a* is sometimes called the "angle of friction" because the tangent of this angle is simply the coefficient of friction for the surfaces in question.

Self-application of Brake Shoes. The sketch in Fig. 28 shows a way in which our experiments with a slider "on the flat" can be applied to the curved surfaces of a brake. Here the tangential force (*P*) is applied to the periphery of a brake drum and the radial force (*F*) is pressing a hinged shoe, carrying a lining, against the drum. We must imagine the drum as being so thin that the tangential force acts at practically the same radius as the friction surfaces. If this force is increased until the drum turns steadily, the ratio *P*/*F* can be measured. To our surprise it turns out to be a great deal higher than in the case of the slider experiment.

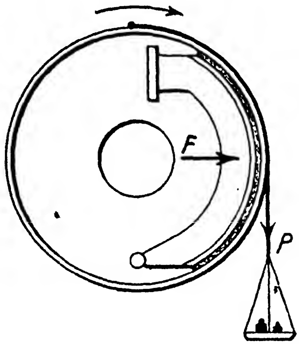

FIG. 28. REPRODUCING THE PREVIOUS SLIDER EXPERIMENT (SEE FIG. 26) BUT USING A BRAKE DRUM AND SHOE

This would disclose a self-applying action in the shoe (see text)

The reason is not to be found in any change in the true coefficient of friction, which will not alter as long as our materials are the same as before. The real explanation is that the force (*F*), which is *applied* to the shoe, is being supplemented by the *self*-applying action of the shoe. And so, if we could find a way of measuring it, the actual load on the surfaces is bigger than the applied force; the ratio *P*/*load* will give us our previous coefficient of friction whereas the ratio *P*/*F* is a false coefficient.

There is no point in making the complicated analysis needed to show all the ins and outs of this servo action in what is called the "leading" shoe. The reason for self-application can be made clear by considering two small pieces of the lining, as in Fig. 29, in the light of our slider experiment (Fig. 27).

The little piece of lining (A) at the "toe," or leading end, of the shoe is experiencing a radial reaction and a

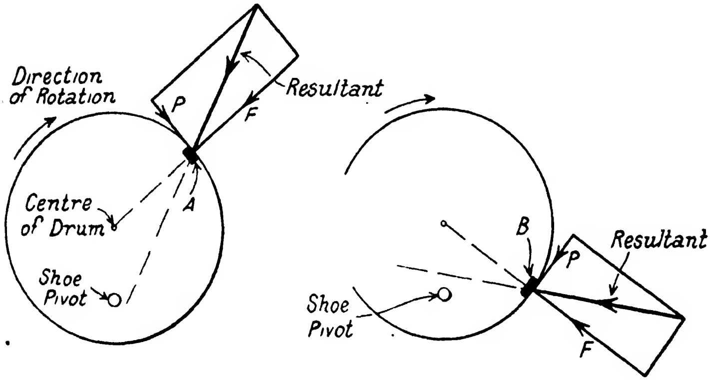

FIG. 29. SELF-APPLYING EFFECTS MADE CLEAR BY STUDYING SMALL PIECES OF BRAKE LINING (A AND B) AT THE TOE AND HEEL OF A LEADING SHOE

tangential drag, shown to scale by the forces P and F respectively. The ratio between these forces is the coefficient of friction and determines the angle at which the resultant force is acting. In the position shown, this resultant acts to the right of the shoe pivot and therefore has a leverage tending to press that piece of lining against the drum—a self-applying action, in short.

Fig. 29 shows a piece of lining (B) at the other end of the shoe. In this case the resultant force produces a "self-freeing" effect, acting against the force which is pressing the shoe to the right. So we could continue, analysing the lining piece by piece, and finally balancing the credits and debits in order to find the degree to which the shoe as a whole possesses this curious servo effect.

Without going to so much trouble, the two specimens *A* and *B* demonstrate certain valuable facts which are worth remembering; in each case the converse is also true—

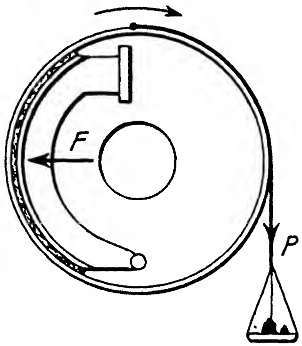

FIG. 30. AN EXPERIMENT WITH A TRAILING SHOE WHICH SHOULD BE COMPARED WITH THE TEST OF A LEADING SHOE ILLUSTRATED BY FIG. 28

1. Extending the leading end of the shoe will aggravate jamming troubles.

2. Raising the location of the shoe pivot will increase the servo effect and may lead to jamming.

3. A change of lining to one giving a lower coefficient of friction will reduce the servo effect.

The Trailing Shoe. So far we have considered only one shoe; what about its counterpart? To answer this question requires a repetition of the experiment of Fig. 28, but using, this time, a trailing shoe in place of a leading shoe. This is shown in Fig. 30, the drum turning in the same direction as before.

As may be anticipated, the effect is reversed—whereas the leading shoe had self-applying properties, the trailing shoe tries to pull away from the drum. So in order to get a certain drag on the drum, we find that—

1. The trailing shoe requires a greater outward force than

would normally be expected, because of its self-releasing action.

2. The leading shoe needs a smaller outward force than would normally be expected, because of its self-applying action.

One other point must be mentioned in this connexion—namely, that when the rotation of the drum is reversed, the trailing shoe becomes the leader and what was formerly the leading shoe becomes the trailer. On the road, this condition is important when a car begins to roll backwards down a steep gradient.

The Simple Two-shoe Brake. In a conventional two-shoe brake, the "expander" takes the form of a cam which can be turned by means of a lever connected to the pedal; the lay-out was shown diagrammatically in Fig. 1. The cam spindle and the shoe pivots are carried by a stationary back plate, mounted on the axle.

In this kind of brake the cam cannot "follow up" the efforts of the leading shoe to provide a servo effect. Indeed, it is obvious from the design that both shoes are bound to move outwards equally and to wear at equal rates. So the net effect is just about the same as if the shoes had no self-applying or self-releasing characteristics. This means that the ratio between the *total* tangential drag on the drum, and the *total* load on the shoes, is approximately equal to the coefficient of friction.

Therefore, in order to obtain the high braking forces needed, the heavy shoe loads required must be obtained from a reasonable pedal effort by employing a big leverage between the two. Just *how* big it must be is best shown by an example, based upon the popular type of 8–10 h.p. British car and assuming a gross weight, fully laden, of 2500 lb.

If we want to get a braking efficiency of 80 per cent we must provide for a total braking force of 2000 lb., or 500 lb. per wheel, assuming equal axle loads. This force is applied at the rolling radius of the tyre—say 12 in.—whereas our braking force acts on the drum at a radius of, say, 4 in. On the principle of leverage, the drag on the drum will have to be three times as great as the retarding force, namely, 1500 lb.

If the coefficient of friction of the brake linings is 0·4, the shoe load required is greater than the drag on the drum in the ratio 10/4; it is therefore—

$$10 \times 1500 \div 4 = 3750 \text{ lb.}$$

So if the total leverage between the shoes and pedal is 100 to 1, it will need 37½ lb. on the pedal *per brake unit* to get a braking efficiency of 80 per cent; that is to say, a total pedal effort of 150 lb. The shoes themselves give a leverage of 2 to 1; hence we need a ratio of 50 to 1 between the pedal and the shoe tips, provided by all the levers in the system, of which the pedal is one.

Leverage and Pedal Travel. The unfortunate aspect of leverage, which otherwise seems to be giving us something for nothing, is that the distances which we have to handle at the two ends of the system tend to get beyond reasonable limits. A ratio of 100 to 1 is practical; it gives, for example, a 1½ in. pedal travel against a shoe travel of 0·015 in. In other words, if the clearance between the shoe linings and the drums is "fifteen thou." the pedal will have to be moved 1½ in. in order to bring the linings into contact with the drums.

Now at the braking end of the system we cannot cut the clearance too fine, and at the pedal end we must not make the travel too great. A distance of 1½ in. is not much in itself but is extended by the movement needed to take up all the slack, and to stretch the operating gear before a full load on the brakes is obtained (see Fig. 31). There is, additionally, some loss of energy in the system.

On top of this, there must be enough spare space between the pedal and the toe-board to provide a reasonable interval between brake adjustments. With a leverage of 100 to

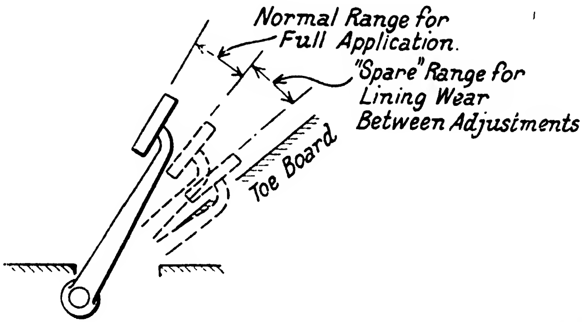

FIG. 31. THE TOTAL TRAVEL AVAILABLE FOR THE PEDAL CAN ONLY PARTLY BE USED FOR BRAKE APPLICATION BECAUSE ALLOWANCE MUST BE MADE FOR "PEDAL DROP" DUE TO LINING WEAR BETWEEN ADJUSTMENTS

1, the pedal travel is increased by no less than 1 in. when the linings wear to the extent of $\frac{1}{100}$ in.

All this goes to show that leverage cannot be increased *ad lib.*; in few cars is it greater than 150 to 1. Consequently, to get a reasonable pedal pressure in a heavy car the designer employs some form of "servo" effect—usually by utilizing the self-applying action of the leading shoes in various ways.

Before leaving the question of mechanical leverage, the two cases shown in Fig. 32 are worth study. Each mechanism gives a leverage of 50 to 1 between the pedal and the shoe tips, as in the example already described. In one case it is made up of a 5 to 1 leverage at the pedal and a 10 to 1 leverage in the shoe expanding mechanism; in the other case these leverages are reversed, as shown.

Each system gives the same shoe loading in return for a given pedal load, but in case *A* the operating rods, cables,

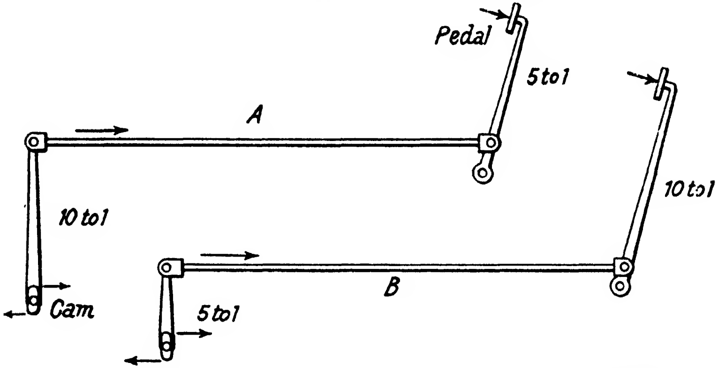

FIG. 32. AN EXAGGERATED CONTRAST BETWEEN TWO SYSTEMS WITH THE SAME OVERALL LEVERAGE

The modern tendency is to use type *A* rather than type *B*, maximum leverage being allocated to the shoe expander—cam or otherwise

etc., are only half as heavily stressed as in case *B*. That is one of the main reasons for the modern tendency to concentrate as much of the total leverage in the shoe expanders as may be possible, and to reduce the leverage adjacent to the pedal to a corresponding extent. And owing to the limitations of the lever and cam, wedge-type expanders have become popular.

The Hydraulic System. A detailed description of the Lockheed hydraulic braking system is given on later pages but the principles involved can appropriately be mentioned here. Instead of using rods, levers, and cables to transmit the pedal effort to the brake shoes, a column of fluid is employed. The pedal operates a plunger, fitted to what is called the "master" cylinder, and between each pair of brake shoes there is a "wheel" cylinder containing two opposed pistons as shown in the diagrammatic lay-out in

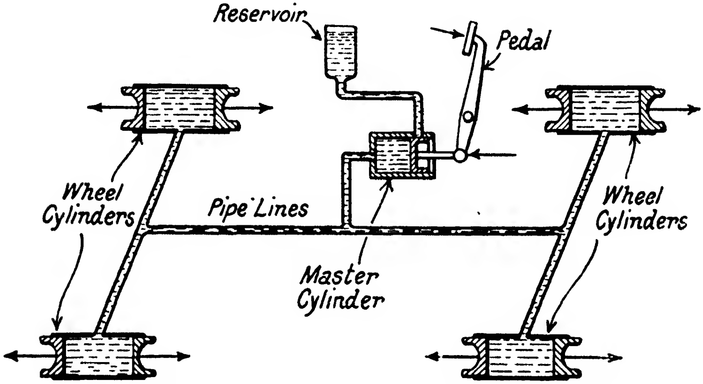

FIG. 33. DIAGRAMMATIC LAY-OUT OF AN HYDRAULIC BRAKING SYSTEM

The pistons in the wheel cylinders act upon the ends of the brake shoes (not shown)

Fig. 33. The cylinders are connected up by pipe lines, and the whole system is filled with a special fluid. A reservoir provides allowance for any expansion and contraction; this must be replenished at intervals.

When the pedal is depressed fluid is displaced from the master cylinder, and the pressure created in the system forces the pistons in the wheel cylinders to move apart and so to expand the brake shoes against the drums. The shoes are pivoted in much the same way as in a mechanical system. One of the advantages of the hydraulic braking system is that the only connexion between the chassis and the wheels consists of flexible tubing, so that the brakes are not affected by wheel movements. Another point is that the pressure throughout a fluid in a closed circuit is uniform, so that the loads on the shoes are automatically equalized.

The fundamental law of leverage does, however, apply equally to hydraulic braking systems. In this case the "leverage" takes the form of a difference between the areas

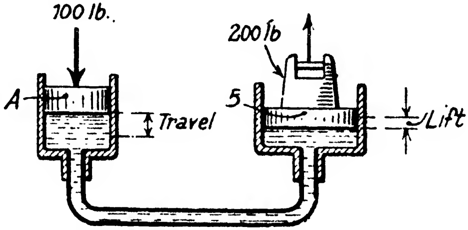

FIG. 34. THE PRINCIPLE OF LEVERAGE HOLDS GOOD FOR HYDRAULIC SYSTEMS AS INDICATED BY THE RELATIVE MOVEMENT OF THE TWO PISTONS (*A* AND *B*) IN THIS SIMPLE ILLUSTRATION

of the plunger operated by the pedal and the pistons which operate the shoes. For example, taking the simple single-unit system shown in Fig. 34, if piston *A* has an area half as great as that of piston *B*, then a force of 100 lb. acting upon piston *A* will produce an effort of 200 lb. at piston *B*. Once again, we do not get something for nothing, because the movement of piston *B* will be only half as great as that of the operating piston *A*.

Brakes Released by Springs. In all these systems, mechanical and hydraulic, means must be provided to release the brakes when the effort is removed from the pedal or hand lever. This is invariably done by means of springs which act in tension between the pairs of shoes, so pulling them towards one another. Sometimes auxiliary tension springs are provided elsewhere in the system.

It is very important that all the brake shoes should return to the "off" position because, if any one of them continues to rub against the drum, the friction created will cause overheating and may affect the performance and fuel consumption of the car. On the other hand, very powerful pull-off springs are not desirable because they have to be overcome by pedal effort before any useful load can be transmitted to the shoes.

There is therefore every incentive to the designer to reduce the friction in the operating system to a minimum so that the pull-off springs have less work to do and need not be excessively tensioned. For similar reasons, anyone concerned with brake maintenance should always take care to ensure that the various parts of the operating system are working freely and are properly lubricated. Parts often neglected are the bearings which carry the main cross-shaft and those which support the spindles of the expanding mechanism adjacent to the brake shoes.

Methods of Brake Adjustment. As the linings become worn in service, the clearance between shoes and drums increases, and consequently the pedal movement needed to apply the brakes gradually grows larger. Eventually, pedal travel is limited by the toe board, but long before this dangerous state is reached the wear of linings should call for compensation by means of the adjusters provided. In all modern brakes the adjusters are fitted close to the shoes, whereas in some of the older systems a single-point adjuster was often provided adjacent to the pedal. It is interesting to examine the reason for this change of design because it throws some light on the basic principles of brake adjustment.

The diagrams in Fig. 35 show a pedal connected to a simple cam expander with an adjustment of the single

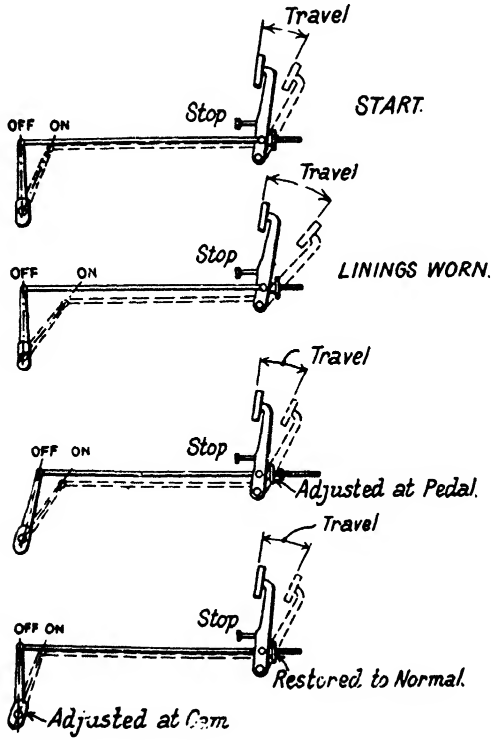

FIG. 35. FOUR DIAGRAMS WHICH ILLUSTRATE ALTERNATIVE METHODS OF BRAKE ADJUSTMENT

At the pedal (three upper views) and at the shoe operating mechanism (lowest view) point type; this takes the form of a nut on a screwed pull rod. The first diagram shows the positions of the parts when the linings are new, and the second diagram shows how wear extends the travel of the pedal and the cam lever.

If the pull rod is shortened to bring the range of pedal travel back to its original position (third diagram) leverage

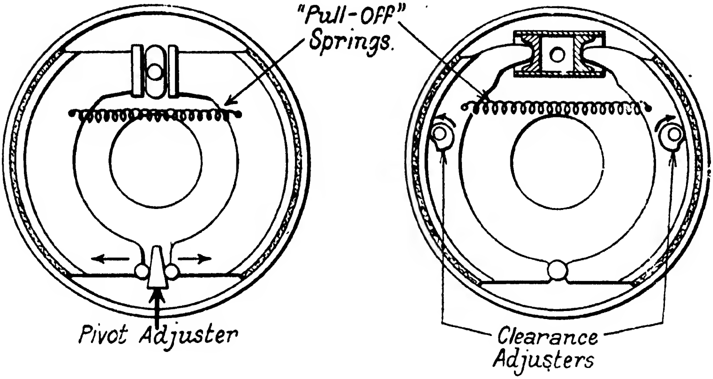

FIG. 36. AS WEAR OF THE LININGS OCCURS IN HYDRAULIC SYSTEMS (*right*) ECCENTRIC STOPS CAN BE USED TO BRING THE SHOES CLOSER TO THE DRUMS WHEN IN THE "OFF" POSITION

For mechanical systems (left) one plan is to increase the distance separating the shoe pivots

is lost because of the angles at which the cam lever and pull rod are now forced to operate. Consequently, it is preferable to adjust the expander instead of the pull rod; this principle is made clear by the fourth diagram, in which the cam lever can be adjusted on the cam spindle; many alternative mechanisms give similar results.

One alternative which has become widely used is to provide an adjuster which increases the distance that separates the shoe pivots. This brings the shoes closer to the drums while leaving the action of the operating system unaffected. In hydraulic systems, fluid added from the reservoir will compensate for lining wear up to a point but eventually a "fresh start" must be made. It is effected by turning eccentric stops against which the shoes return when the brakes are released; the result is to reduce the clearances between the shoes and drums, as shown in Fig. 36.

Apart from compensating for wear of the linings, brake adjusters provide a means for ensuring that each brake shall do its proper share of the work of retarding the car. This is particularly important in the large majority of mechanical systems where wide differences in shoe clearance are liable to cause equally wide variations in braking effect.

CHAPTER V

CONSTRUCTIONAL FEATURES, ADJUSTMENT AND MAINTENANCE

At the beginning of the preceding chapter we referred to the fact that the majority of British cars employ one or other of three proprietary braking systems. It is therefore essential that anyone concerned with motor car brakes should possess a working knowledge of these systems. For this reason the present chapter is entirely devoted to their constructional features, adjustment, and maintenance.

Widely distributed service facilities are available for these brakes and the makers also operate systems of shoe replacement which avoid much of the relining work formerly done in garages. Further references to these services will be found on pages 127 and 128.

The following account should be read as an extension of the general description of brake mechanisms given in the preceding chapter.

The Bendix Brake

As two entirely distinct and separate types of brake are made by Bendix Ltd., we must explain at the outset that by "Bendix brake" we mean the original "Duo-servo" design. In various forms this has been (and is) widely used on many makes of car, and it was first developed in its present cable-operated form during 1930. Some years later, the Bendix-Cowdrey braking system was brought out by the same manufacturers; this will be described and illustrated independently on later pages.

Commencing, then, with the Bendix brake, the outstanding feature has always consisted of utilizing the self-applying qualities of a leading brake shoe to the greatest practicable extent. To this end the two shoes, instead of being mounted on pivots fixed to the back plate, are connected by a link in which the adjusting mechanism is incorporated. When the leading (or primary) shoe is expanded, and exerts its self-applying action, it tends to be dragged round by the drum and so produces a force which adds to the pressure exerted by the secondary shoe against the drum.

The expander is operated by an internal lever and a cable passing through a flexible conduit secured to the back plate (see Figs. 37 and 38). It has a "floating" action, first bringing the primary shoe into contact with the drum while reacting against the end of the secondary shoe. The drag on the primary shoe completes the loading of the secondary shoe. If the pedal pressure is increased, the expander again takes its pivot, or reaction point, from the secondary shoe, holding it against the drag while adding to the load on the primary shoe.

A brake of this kind gives a powerful response and an excellent stopping distance in emergencies. It does, however, require really careful adjustment in order to avoid ineffectiveness on the one hand and too powerful a servo effect on the other hand. And to keep the servo action within bounds the makers specify linings with a moderate coefficient of friction; on no account should high-friction linings be substituted.

As the shoes are not hinged on pivots, it is clear that some special means must be provided to locate them properly while allowing them the movement needed for brake operation. In the "Double Anchor" (D.A.) form of Bendix

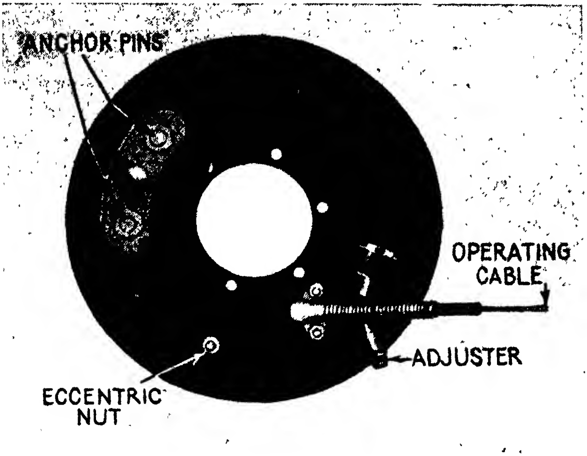

FIG. 37. THE BACK PLATE OF THE BENDIX "DOUBLE ANCHOR" TYPE OF CABLE-OPERATED BRAKE, SHOWING THE OPERATING CABLE, ADJUSTER, ETC.

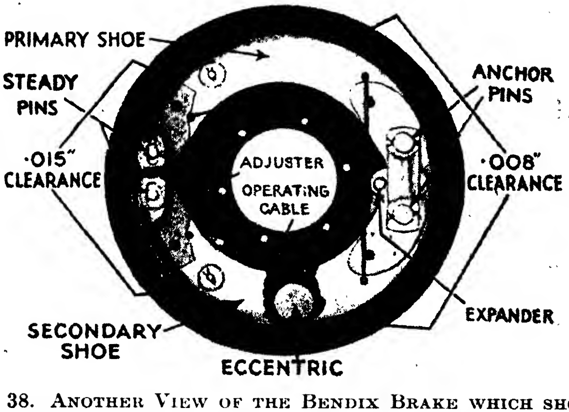

FIG. 38. ANOTHER VIEW OF THE BENDIX BRAKE WHICH SHOWS THE SHOES AND OTHER PARTS. FOR FORWARD TRAVEL THE DRUM TURNS ANTI-CLOCKWISE, IN THIS VIEW

brake, two anchor pins form the locating device. Each fits through a slot in the web of a shoe, adjacent to the expander (Fig. 38) and is firmly secured to the back plate with latitude for adjustment. The correct positioning of the anchor pins is the first step towards successful adjustment and is of paramount importance to the correct action of the brake. If it is not properly effected, no amount of fiddling with other means of adjustment will produce the desired results.

The makers set the shoes and their anchor pins on a jig before the braking unit leaves the factory, but resetting is required, after assembly on a car, in order to compensate for slight variations in axle mountings and brake drums. It is done in the following way—

First, the nuts on the anchor pins must be loosened slightly, after which the adjuster between the shoes is turned until the linings are in close contact with the drum. Each anchor pin should then be given a straight "end-on" blow with a soft hammer, after which the nuts can be tightened to within half a turn of the "fully home" position.

Next, the adjusting screw is slacked back until the right clearance is obtained between shoes and drum. An eccentric stop, provided for locating the secondary shoe in the "off" position, may also be adjusted to centralize the shoes in the drum, after which the nut by which the stop is secured to the back plate must be fully tightened.

Feelers are now employed to check the shoe/drum clearances and, if necessary, the anchor pins may be shifted slightly by striking them with a soft hammer. The aim is to obtain about twice as much shoe clearance at the adjustable ends as that which exists at the operating ends. A good result is represented by clearances of 0·015 in. and

0·008 in. respectively—$\frac{15}{1000}$ and $\frac{8}{1000}$. Finally, the anchor pin nuts must be tightened to the utmost extent, using a spanner not less than 16 in. in length in order to obtain the requisite leverage.

In the small sizes of Bendix brake, used on light cars, a slightly different design is employed. It embodies a single anchor pin and is therefore often known as the "S.A." Bendix brake. There is an eccentric adjuster and a shoe adjuster, as before, and the procedure required is similar to that already described for the D.A. brake. The only important difference is that the S.A. brake requires a uniform clearance, all round, between linings and drum; the recommended figure is 0·008 in. ($\frac{8}{1000}$).

Bendix Brake Adjusters. When the shoes have been properly positioned in the first instance, lining wear can usually be compensated for by using the adjuster provided between the primary and secondary shoes. This is operated from behind the back plate in several different ways. The simplest mechanism comprises a screw, with left-hand and right-hand threads, which carries a notched collar. By inserting a screwdriver through a slot in the back plate, and using it as a lever against the slot, the collar can be turned (Fig. 39). To expand the shoes, the screwdriver handle is moved towards the axle, and vice versa.

In order to suit cars in which this simple device would not be accessible, two other mechanisms are in use for operating the screwed adjuster between the shoes. One employs spur gears operated by a flexible spindle terminating in a hexagon head (Fig. 40). To expand the shoes the head is turned towards the frame with a box spanner —i.e. clockwise for the off-side brakes and anti-clockwise for the near-side brakes.

The "crown-adjuster" is another type of mechanism and is more positive in action than the flexible spindle. It is operated by a hexagon head, projecting behind the back

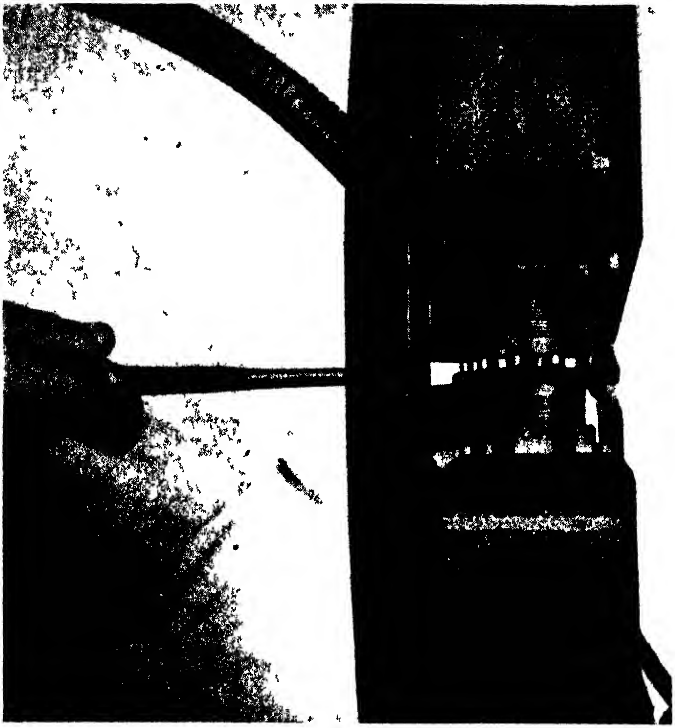

FIG 39 ONE FORM OF BENDIX BRAKE ADJUSTER WHICH IS OPERATED BY USING A SCREWDRIVER THROUGH A SLOT IN THE BACK PLATE

plate, and is turned clockwise, to expand the shoes, in all cases—i.e. in the same direction as when tightening a nut.

The Bendix Operating System By far the most important of the various "don'ts" which mechanics have to remember is that the lengths of the operating rods and/or

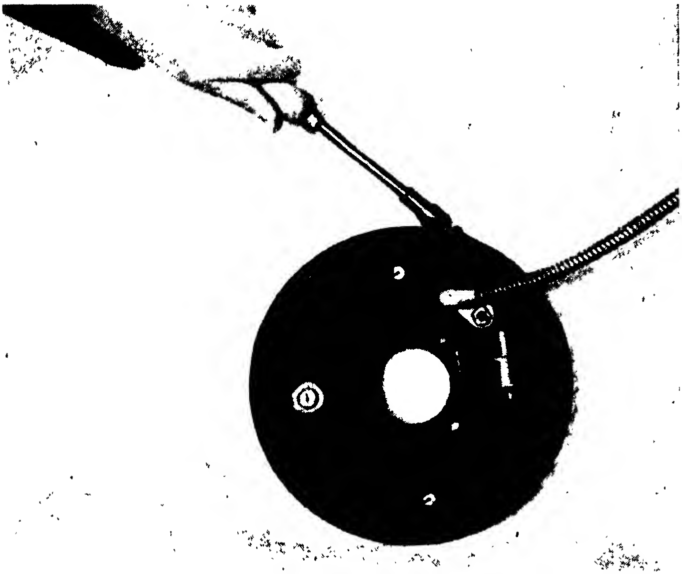

FIG. 40. ANOTHER KIND OF BENDIX BRAKE ADJUSTER WHICH IS ARRANGED FOR OPERATION BY A BOX SPANNER

cables should never be altered; Bendix brakes are designed for adjustment *at the shoes* and cannot satisfactorily be handled in any other way. The only exception arises when new cables are being fitted throughout.

Of course, trouble in securing a good result when adjusting brakes may be due to undue friction in the operating gear, or loose abutments where the flexible conduits are secured.

There are many points at which stiffness will develop if the car has been neglected, such as the bearings which carry the cross-shaft and pedal shaft, the knuckle joints (and their pins) at lever ends, etc. If the brakes fail to return freely to the "off" position, when pedal and hand lever are released, these points should be checked and lubricated. There is also the possibility of front brakes becoming adversely affected by grease from the hubs, or rear brakes by oil from the axle, necessitating the renewal of linings, and sealing devices. Any brake is affected permanently by lubricants, and temporarily by water, but the duo-servo shoe mechanism is, by its very nature, sensitive to these unwelcome influences—simply because the drag on one shoe is being relied upon to load the other shoe.

A typical Bendix operating system is shown in Fig. 41. The main cross-shaft can be operated either by the pedal or the hand lever, the pull-rod from each of these controls embodying a slotted link so that they can work independently. From double levers at the ends of the cross-shaft, rods extend to the conduit abutments near to each axle; from these points the effort is transmitted to the brakes through cables. A detail of importance is the provision of a steady bearing on the cross-shaft between the levers operated by pedal and hand lever. This ensures that even if the cross-shaft should fracture it would still be possible to operate two of the brakes by means of one or other of the controls. Apart from this widely used system, there is an alternative design in which levers replace the cross-shaft.

The Bendix-Cowdrey Brake. Introduced in 1936, the Bendix-Cowdrey braking system has since been applied to a number of well-known cars. Servo shoes are employed, with their ends coupled by a floating adjuster, but the expander is quite different from that of the Bendix brake,

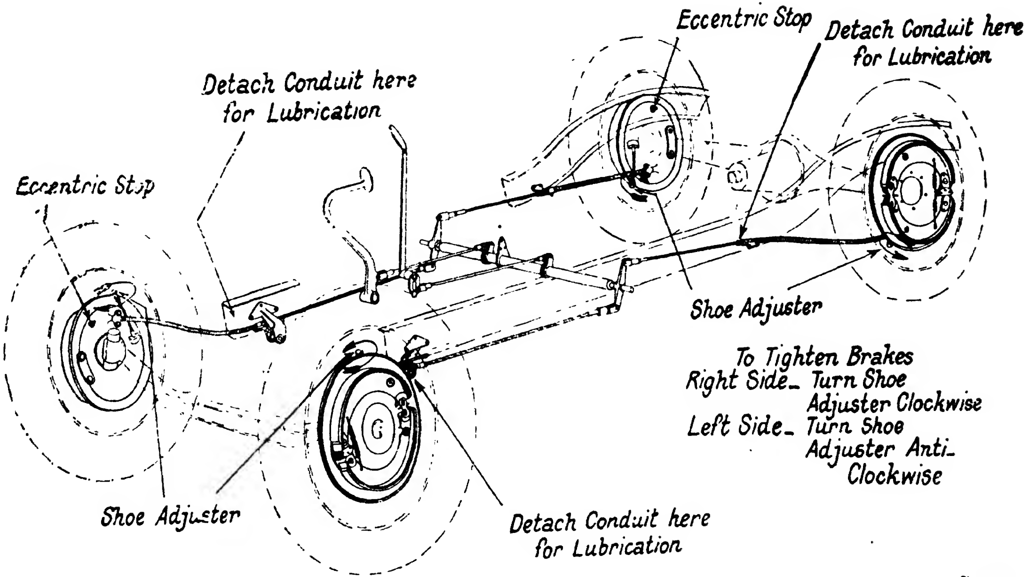

FIG. 41. A TYPICAL BENDIX OPERATING SYSTEM, SHOWING THE CROSS-SHAFT, RODS, LEVERS, AND CABLES EMPLOYED TO COUPLE THE BRAKING UNITS TO THE PEDAL AND THE HAND LEVER

as is also the operating mechanism used between the expanders and the brake pedal.

Each expander works in the following manner (see Figs. 42 and 43). A plunger, of hardened steel, fits in a housing at right angles to the shoes and is pulled towards the centre of the car when the brake pedal is applied. Hardened steel balls are thus forced to roll between inclined faces formed at the ends of opposed tappets which, when moved outwards by this wedging effect, expand the shoes into contact with the drum.

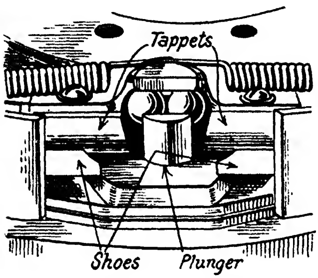

FIG. 42. THE SHOE EXPANDING DEVICE OF A BENDIX-COWDREY BRAKE CONSISTS OF BALLS AND A PLUNGER OF HARDENED STEEL

The primary shoe applies a load to the secondary shoe and, as in the Bendix brake, the necessary "carry forward" motion is permitted by the expanding device. Thus the drag of the rotating drum pulls the shoes circumferentially until the end of the secondary shoe comes into contact with the housing of the expander. Further movement of the plunger enables extra load to be placed upon the primary shoe because the balls, in addition to rolling inwards between the inclined faces of the tappets, are able to move laterally through the hole in the plunger in which they are located.

The balls are made from hardened steel as are also the plungers; thrust plates are fitted to the shoes to take the end of the tappets. Owing to the fact that the shoes are operated through a leverage set by the angle of the tappet faces, there is no change in this leverage as wear of the linings occurs.

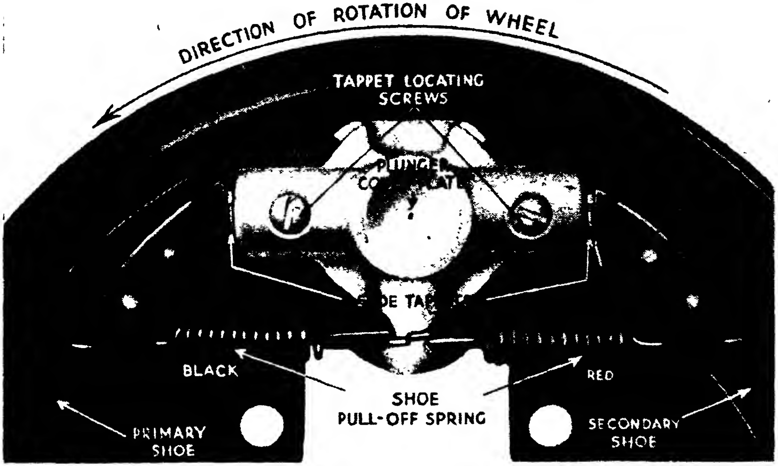

Fig. 43. Another View of the Bendix-Cowdrey Shoe Expander which shows the Tappets, Shoes, and Pull-off Spring

However, as in any other brake, when lining wear goes beyond a certain limit the pedal travel becomes excessive and adjustment is needed; this is effected at the brakes

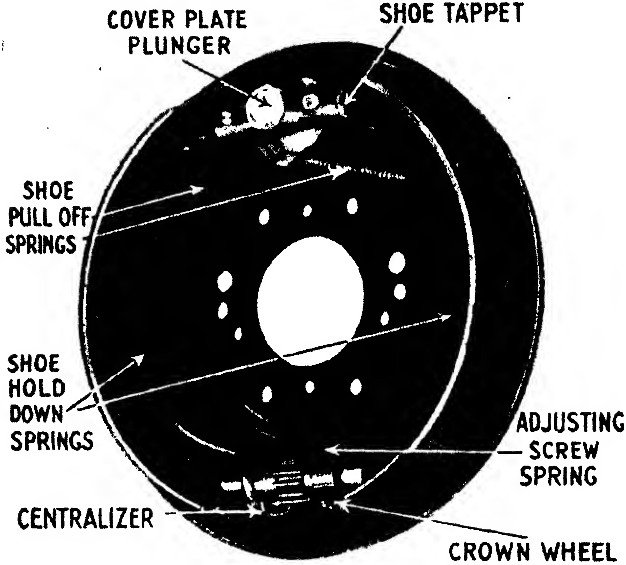

FIG. 44. THE ADJUSTER OF THE BENDIX-COWDREY BRAKE IS FITTED BETWEEN THE ENDS OF THE SHOES AND IS LOCATED BY A CENTRALIZING DEVICE

themselves and should on no account be attempted by shortening the brake rods. As shown in Fig. 44, the primary and secondary shoes are connected by a sleeve fitted to studs with left-hand and right-hand threads. A number of teeth formed on the outside of the sleeve engage with a crown wheel fitted to a spindle which passes through the back plate and terminates in a nut or other suitable device. By turning this nut the sleeve is caused to revolve and the shoes are moved apart.

A "clicking" mechanism prevents the adjuster from slacking back and also indicates the extent to which it is being turned. Thus, to adjust the brakes it is only necessary to turn the adjuster until the shoes are in firm contact with the drum, then reversing the adjustment by five "clicks" to obtain the right clearance. Before attempting this, the pedal and hand lever must, of course, be placed in their inoperative position. A detail worth noting is that the teeth on the sleeve can slide through those on the crown wheel when the brake is in action, so as to allow the slight circumferential movement which the shoes require for their servo effect.

In all Bendix-Cowdrey brakes "steady" springs are provided to locate the webs of the shoes against rests which project from the back plate; these are fitted as shown in Fig. 44. The usual pull-off springs are provided between each pair of shoes and are coloured to ensure that they shall be assembled in the correct locations. In all the different types, except the smallest (Light Range), the shoes are centralized in the drum by means of a double-acting spring bearing against each side of a collar fitted to the adjuster sleeve. The spring "gives" to shoe movement when the brake is operated but provides a centralizing effect when the shoes are released. The spring is located by a bolt passing through the back plate but needs adjustment for position only when the brake has been dismantled for fitting new shoes or for any other reason.

After fitting new shoes it may be found that the brake does not free itself properly although apparently in correct adjustment. This indicates that the expander housing needs re-locating in addition to the centralizer. The procedure is simply to slacken the nuts holding the housing

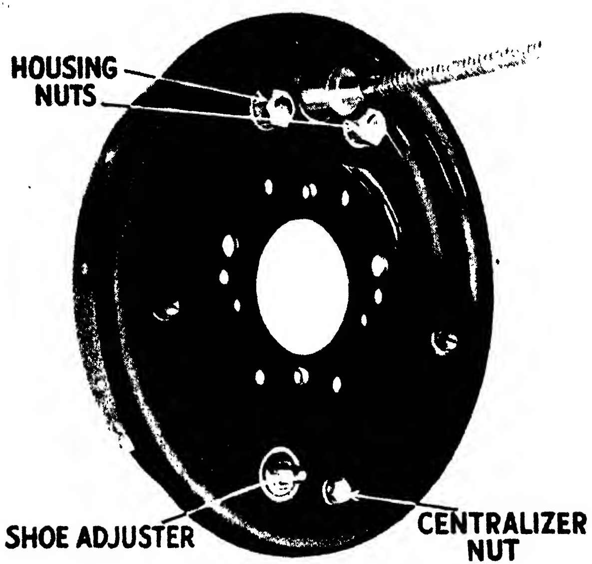

FIG. 45. VARIOUS PARTS OF THE BENDIX-COWDREY BRAKE WHICH MAY NEED ATTENTION AT INTERVALS, AS EXPLAINED IN THE TEXT

and the centralizing device (Fig. 45), then screwing up the adjuster until the shoes are forced against a drum. While this is being done the nuts should be tapped with a soft hammer to help the housing and centralizer to move slightly into their correct positions. Finally, the nuts must be fully tightened and the adjuster slacked back three-quarters of a turn (five or six "clicks").

Overhaul of Bendix-Cowdrey Brakes. The following procedure is recommended for the annual overhaul of these brakes or whenever replacement shoes are fitted. First,

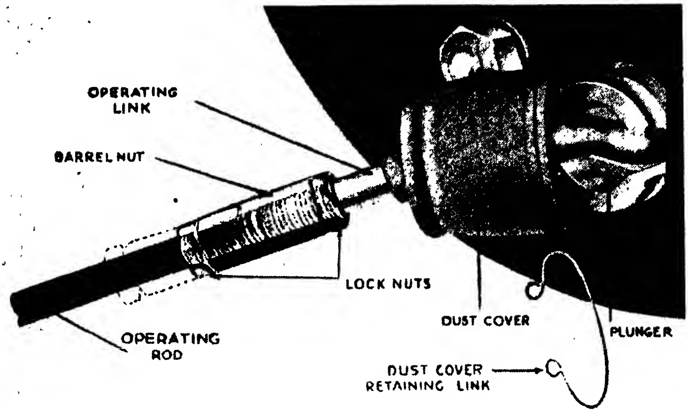

FIG. 46. WHEN A COMPLETE OVERHAUL IS NEEDED THE BENDIX-COWDREY OPERATING RODS, ONE OF WHICH IS SHOWN HERE, MUST BE DISCONNECTED BEHIND THE BACK PLATES

it is desirable to check the tightness of road spring U-bolts, back plate bolts, etc., after jacking up all four wheels. Next, wheels and brake drums must be removed and, on all brakes except the smallest size, the centralizer nuts should be loosened. The following parts can then be removed, in order—steady springs, pull-off springs, and brake shoes.

As a rule, locating screws will be found towards each end of the expander housing which must be removed before the tappets and balls can be taken out. The operating plunger can next be dismantled after disconnecting the link and the rod (or cable) behind the back plate (see Fig. 46). Where a rod is used the connexion is formed by a barrel nut (with lock-nuts) which can be screwed back along the rod until the parts come adrift.

Following this, all parts should be cleaned thoroughly and those which have work to do should be lubricated with a graphite grease of good quality. A little grease should also be applied to the steady rests and washers by which the shoes are located. Assembly of the brakes is simply carried out in the reverse order, taking care to replace the pull-off springs in their former positions as indicated by colours. In coupling the rods which operate the expander, care should be taken to ensure that the ends butt together inside the barrel nut and that the lock-nuts are fully tightened. Subsequently, the brakes are adjusted —and the shoes re-located—by the methods already described.

Bendix-Cowdrey Operating Gear. Although the details vary to some extent according to the make of car in question, the essential features are as follows (Fig. 47). Adjacent to the brake pedal there is a compensating device which distributes the effort to rods running forward and rearward. These rods in turn operate compensators (carried on the front and rear axles) from which additional rods run outwards to the four brake expanders. The handbrake is hooked up to the rear brake operating gear through a relay lever.

Beyond seeing that all the joints and bearings are properly lubricated, so as to work freely, the service engineer will not as a rule be concerned with the setting of the operating mechanism. With the brakes in proper

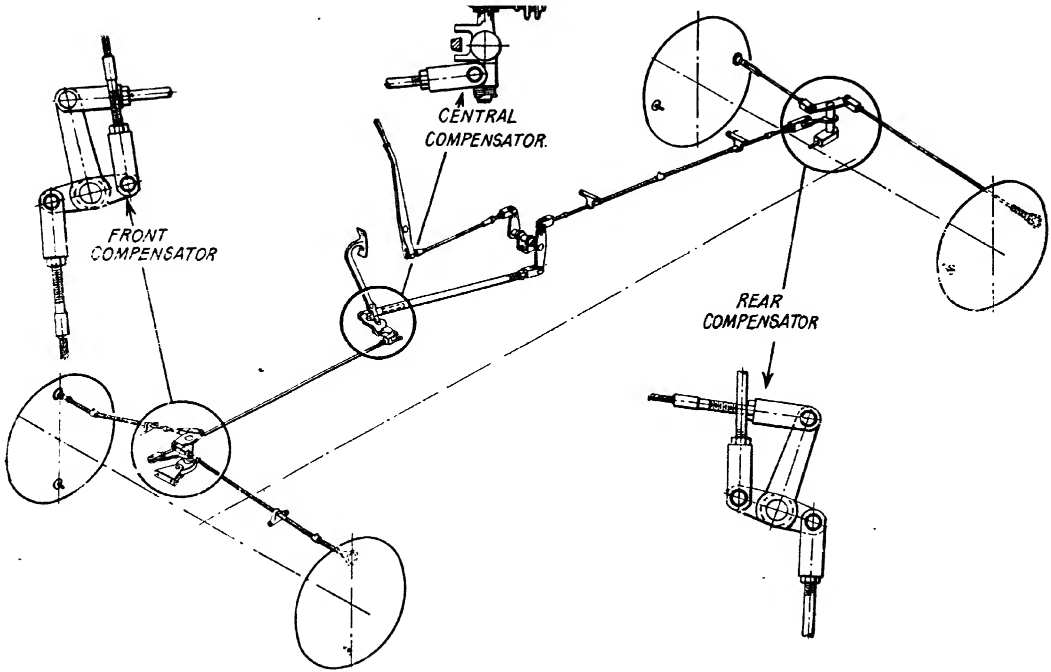

Fig. 47. A Typical Bendix-Cowdrey Operating Gear with the Three Compensating Devices (in Circles) Picked Out and Enlarged to Show the Mechanism Clearly

adjustment, the pedal should have a free travel of about $\frac{3}{4}$ in. from the "off" position to the point at which the shoes make contact. Similarly, the handbrake should have a free travel of about two notches on the ratchet. Full instructions are available from Bendix Ltd., for resetting the operating gear should this have to be done after a car has been completely dismantled; for example, following a crash or serious axle trouble.

The Girling Brake

Owing to its simple and effective design, the Girling brake has scored a remarkable success during the past few years. It was introduced by New-Hudson Ltd. in 1934, and by 1938 had become standardized by a large number of car manufacturers. Fundamental features are the provision of most of the leverage adjacent to the brake shoes, the arrangement of the operating gear to reduce friction and "springiness" to a minimum, and the provision of compensators which ensure the correct distribution of braking effort to the four wheels. Adjustment is simply carried out by a mechanism on each brake which extends the distance separating the shoe pivots; on no account should the operating rods be altered in a misguided attempt at adjustment.

A typical Girling brake assembly is shown in Fig. 48 from which it will be seen that the brake shoes are separated by an expander at one end and by an adjuster at the opposite end; these units are mounted on the back plate. The shoes are usually pressings, made from a solid drawn steel of T-section, although in some earlier designs shoes of aluminium alloy were employed.

An expander unit is shown in detail in Fig. 49 and comprises a housing, a central "cone" of hardened steel

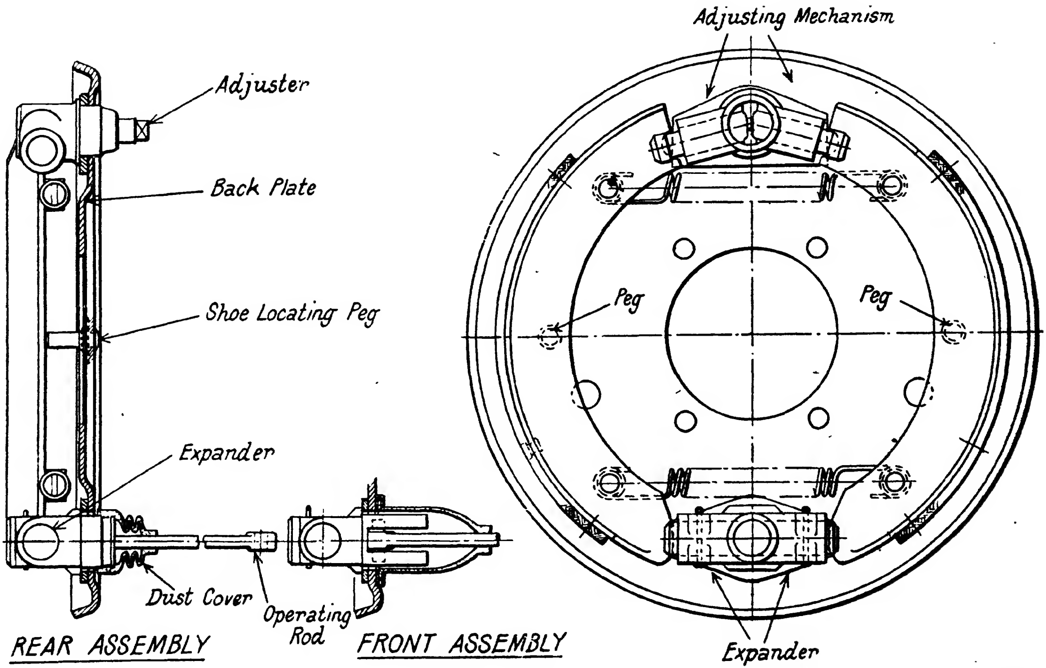

FIG. 48. A TYPICAL GIRLING BRAKE ASSEMBLY, SHOWN AS A SECTION THROUGH THE BACK PLATE (*left*) AND AS A FRONT VIEW (*right*)

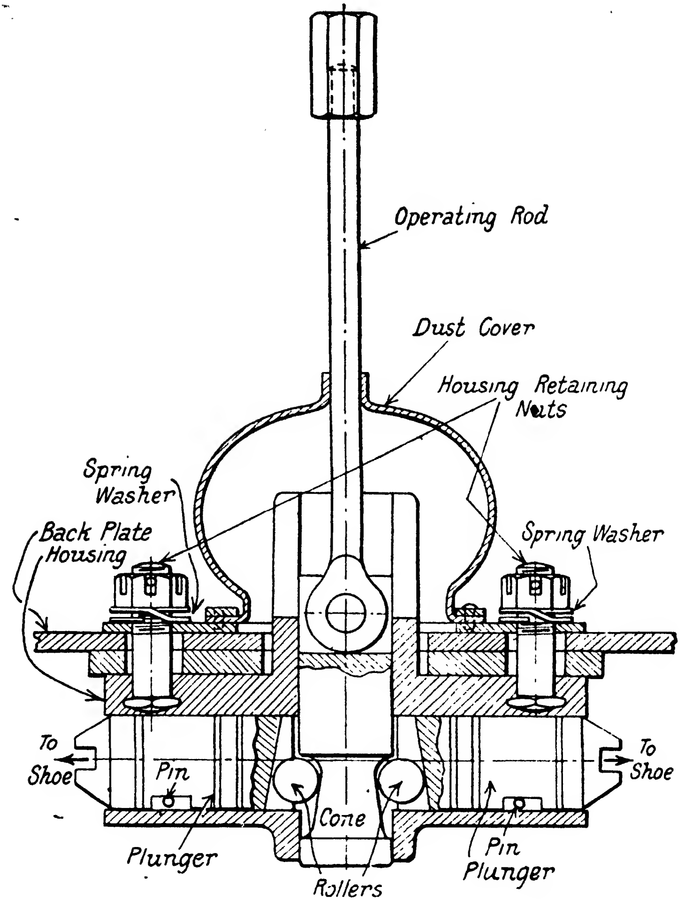

FIG. 49. A DETAILED VIEW, IN SECTION, OF THE GIRLING EXPANDER, WHICH CONSISTS OF A CONE, ROLLERS AND PLUNGERS OF HARDENED STEEL

The cone is operated by a pull-rod

(operated by a pull-rod or cable), a pair of rollers, and two plungers. We use the word "cone" because it is employed by the brake makers, but the part in question is actually a sliding rod in which a recess of tapering diameter is formed between collars. The rollers fit in this recess so that, when the rod is operated, the cone-shaped surface pushes them outwards and separates the plungers which, in turn, act upon the ends of the brake shoes. On assembly, the housing is packed with a special grease, and it requires no attention except when a complete overhaul is needed after prolonged use.

A feature of the mechanism is that the expander is able to slide to a slight extent on the back plate so as to ensure that the shoes can centralize themselves in the drum. To enable it to do so, the studs by which it is attached fit through slots and are loaded by flat spring washers under brass nuts; on no account must these nuts be fully tightened. If the brake is dismantled at any time, the bedding surface of the housing should be smeared with the special grease, provided by the makers, before assembly.

The plungers are retained in the housing by split pins which pass through slots to allow of the necessary movement. In earlier designs set-screws were employed, and it is important that these should not be very forcibly tightened; they are prevented from working loose by spring washers under the heads. Here it may also be mentioned that, in the earlier cars in which cast shoes were used, the expander took a slightly different form, but the main features were the same as those just described.

The expander unit is protected from the ingress of grit or moisture by a flexible cover fitted over the operating rod and riveted to a steel plate mounted on the studs by which the housing is retained; this plate serves to cover the slotted holes in which the studs work. If any sluggishness develops in the action of the expander, it should be stripped down, cleaned and lubricated with the zinc-base grease recommended by the makers; it is a Keenolized lubricant known as K.G. 20.

The Girling Brake Adjuster. The adjusting unit used between the brake shoes consists of a steel housing in which are fitted a screwed cone and two plungers (Fig. 50).

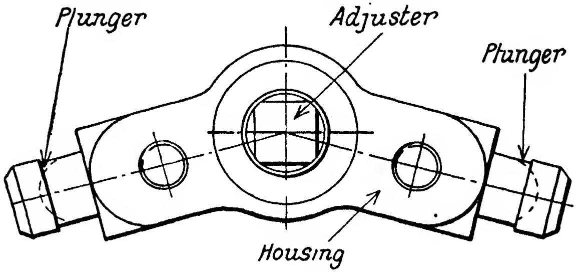

FIG. 50. THE GIRLING ADJUSTER; THE PLUNGERS CARRY THE ENDS OF THE SHOES IN SLOTS

(A cone with four "flats" fits between the plungers and is operated by a screw, passing through the back plate, with a square end projecting)

At one time the plungers were located in a straight line but in all recent types they are inclined as shown. The stem of the cone projects through the back plate and at the end it is machined to a square for operation by an ordinary spanner. Four flats are machined on the cone, where this part passes between the plungers, and the outer ends of the plungers carry slots to engage with the brake shoes. In recent designs the screwed part of the cone stem is always protected by the boss on the housing, whereas in earlier types it projected to some extent and was apt to need treatment with penetrating oil to remove mud or rust.

In order to adjust the brakes, the projecting end of the stem is turned right-hand (clockwise) until resistance is felt as the shoes make contact with drum. It must then be slacked back one full notch. This can easily be gauged because, after an initial "click" as the flats on the cone come into line with the plungers, a second click will indicate that the correct clearance has been reached. Having done this to all four brakes the pedal should be given a sudden and hefty application in order to centralize the shoes (by slightly shifting the expander housings) before any tests are made. It should hardly be necessary to add that the pedal and hand lever must be "off" before the adjusters are operated.

The shoes are fitted with studs which are connected in pairs by pull-off springs. In order to hold the shoes square against the slightly off-set loading of the springs the shoe webs are arranged to rest upon locating pegs fitted to the back plate (Fig. 48). In earlier designs these pegs were fitted with graphite inserts and various types of cap in order to prevent squeaking, but at present plain pegs are used which are greased on assembly and show no wear liable to cause misalinement.

Replacing Girling Brake Shoes. When the linings are worn out, replacement shoes for Girling brakes can be obtained from Ferodo stockists complete with linings accurately ground and ready for use. After removing the drum, the old shoes are dismantled as follows. A screwdriver is used as a lever between the web of the shoe and the locating peg on the back plate, so as to prise one end of the shoe out of the slot in the expander plunger. This releases the spring tension and enables both the shoes to be taken off. The expander and adjuster units should be left in place and care should be taken to avoid overstretching the pull-off springs.

After cleaning the mechanism, and greasing if necessary, the adjuster should be slacked back as far as possible. Next, the shoes and springs should be put on as an assembly, first fitting the semicircular slots in the shoe ends to the expander plungers and then guiding the opposite ends of the shoes into the expander plungers. The end of the second shoe can be put into place by using a screwdriver as a lever, as when dismantling.

Before attempting to adjust the new shoes the nuts which secure the adjuster housing to the back plate should be slacked back to the extent of one full turn. After this the adjuster should be operated until the shoes are tight in the drum. Then the adjuster housing nuts may be fully tightened before slackening the adjuster to get the right clearance.

Girling Brake Operating Gear. A typical lay-out is shown in Fig. 51. The rods which operate the four brake expanders (numbered 3) are coupled in pairs to balance levers, or compensators (4), mounted on the front and rear axles. These balancing devices are in turn connected by pull-rods (1) to a compensator below the pedal (2) and to a lever on a short cross-shaft. The compensator at the lower end of the pedal is connected to the forward pull-rod and is also coupled to a cross-shaft lever by a compression rod embodying a sliding spring-loaded joint. The spring maintains a slight degree of tension throughout the system when the brakes are free and so serves to prevent rattle in the various joints. Consequently, the designers are able to permit sufficient play in the joints to ensure free action without lubrication.

Care should be taken that the sliding joint in the compression rod works freely and that a clearance of about $\frac{1}{16}$ in. is maintained between the plunger part and the end

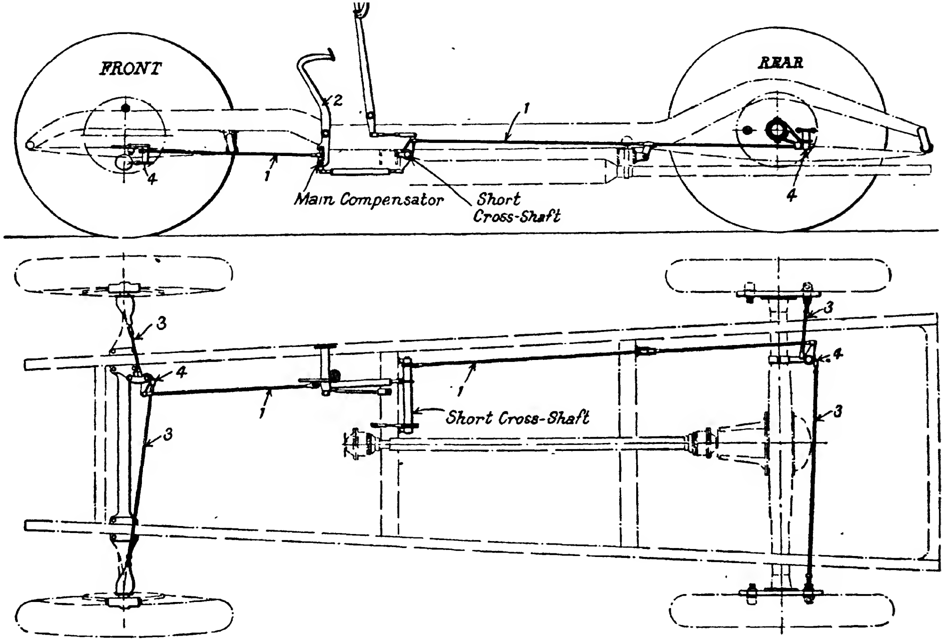

FIG. 51. TYPICAL CHASSIS LAY-OUT OF A GIRLING BRAKING SYSTEM, IN ELEVATION AND IN PLAN
There are three compensators and the brake expanders are operated by pull-rods
(Note the directness and simplicity of the design)

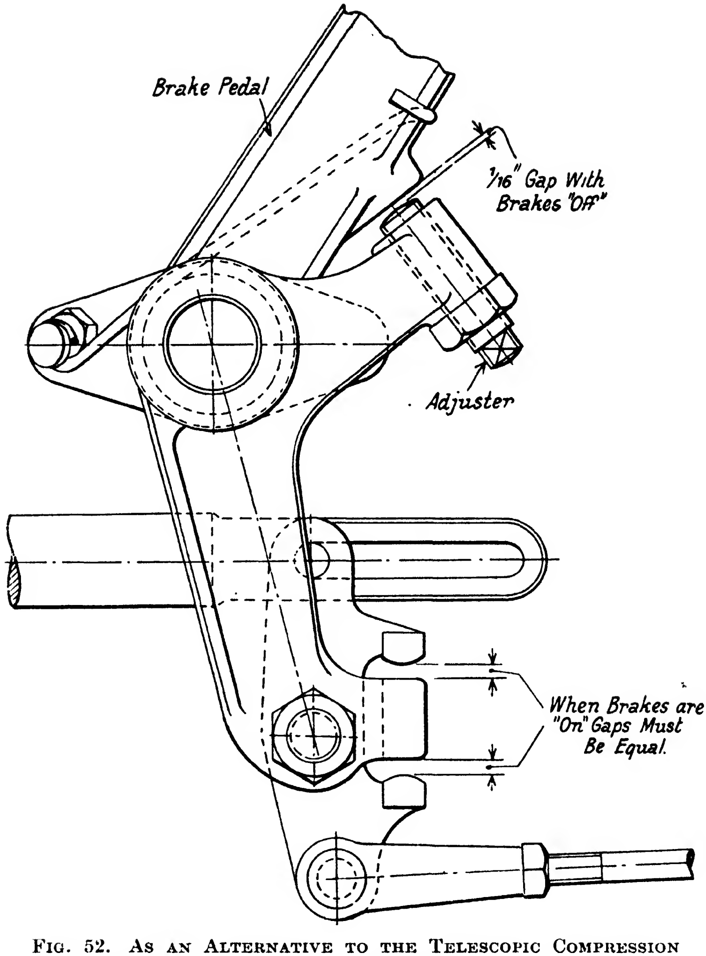

FIG. 52. AS AN ALTERNATIVE TO THE TELESCOPIC COMPRESSION ROD OPERATED BY THE PEDAL AS SHOWN IN FIG. 51, a SOLID ROD IS USED IN THE GIRLING SYSTEM AS FITTED TO THE LARGER AUSTIN MODELS, AND THE PEDAL IS ARTICULATED

of the tube in which it fits, when the pedal is "off." In an alternative design used on the larger Austin models a solid compression rod is employed and the brake pedal is articulated as shown in Fig. 52. The clearance is then necessary at the abutment through which the upper part of the pedal operates the lower part; it is adjusted by a set-screw.

Apart from individual differences in certain makes of car, easily understood by consulting instruction books, the general principle of the operating mechanism is as shown in Fig. 51. Thus, the balance lever at the bottom of the pedal ensures a correct distribution of effort, front/rear, and a safety stop is provided so that if the forward rod should break the rear rod will remain operative, and vice versa.

For the compensating levers on the axles two designs are widely used; these are shown in Figs. 53 and 54. One of them (Fig. 53) is mounted on a swinging link and is suitable for systems in which the rods which it operates run alongside the axle, behind the front axle and ahead of the rear axle. The alternative fulcrum pin type (Fig. 54) is employed for rods arranged ahead of a front axle or behind a rear axle. These compensators, or balance lever assemblies, are usually fitted with self-lubricating bushes, needing no attention, although in some cars bronze bushes are used and are fitted with grease gun nipples.

Although pull-rods are almost invariably employed in the Girling system there are certain chassis in which the expander units are operated by cables running through conduits. The ends of each conduit are anchored (1) to a recess in the bore of the expander housing; (2) to an adjustable abutment on the frame of the car.

The hand lever, as shown in Fig. 51, is simply connected to a lever on the short cross-shaft by a slotted link. When the pedal is depressed, a pin on the lever moves within the slot without affecting the handbrake. Conversely, when

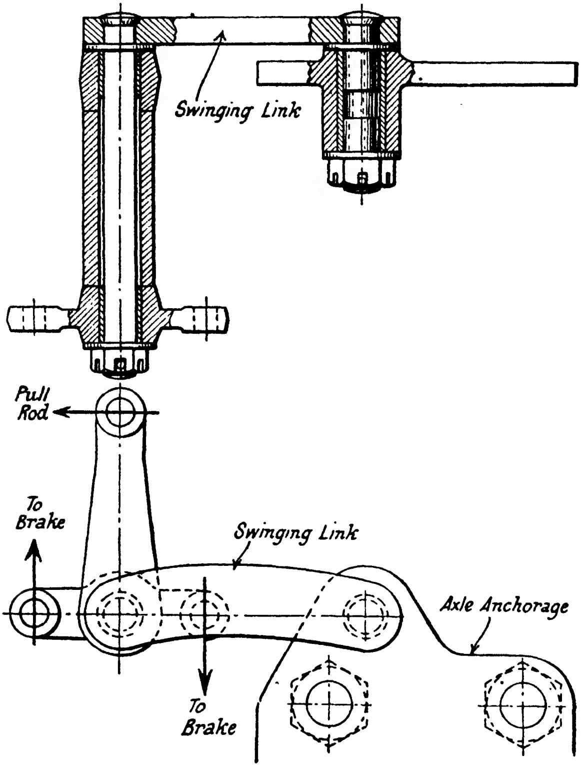

FIG. 53. ONE FORM OF AXLE-MOUNTED COMPENSATOR, USED IN GIRLING BRAKING SYSTEMS, WHICH IS CARRIED BY A SWINGING LINK

the hand lever is applies, the sliding joint (already described) takes up the movement without operating the pedal.

With this plan the handbrake takes effect upon the rear wheels only but there are alternative lay-outs in which it operates upon all four wheels.

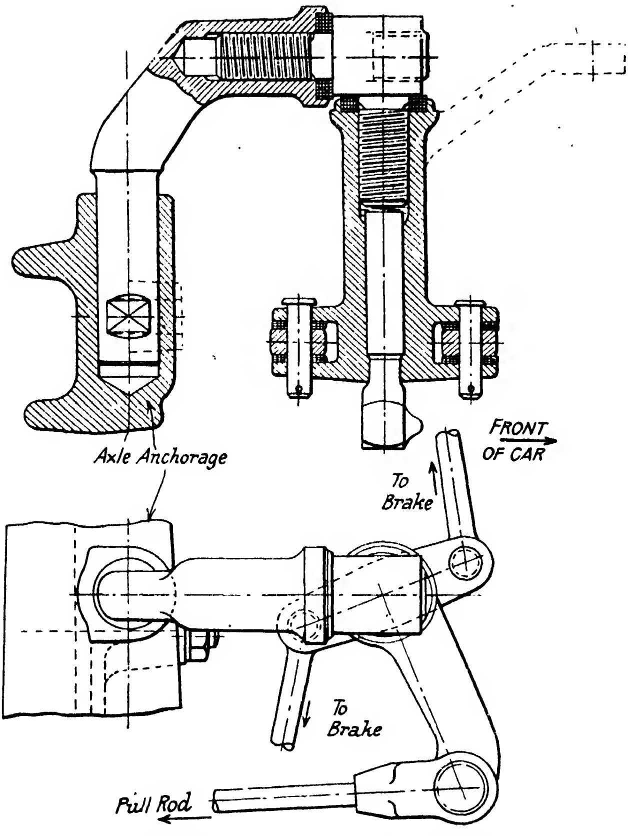

FIG. 54. AN ALTERNATIVE TYPE OF GIRLING COMPENSATOR WHICH IS SUPPORTED FROM THE AXLE ON A VERTICAL FULCRUM PIN

The Lockheed Hydraulic Brake

This excellent system has been manufactured in England since 1930 and is standardized for many very popular cars. The leading advantages of the hydraulic principle employed were described in the preceding chapter and, so far as maintenance is concerned, the need for occasional replenishment of the reservoir with fluid is offset by the fact that there is no operating gear to require lubrication or to give trouble by creating friction. The slight risk of brake failure through leakage has been eliminated so far as possible by extreme care in manufacture and by developing piping which has a big margin of safety beyond the limits of pressure ever likely to be reached in service.

A point which is so important that it cannot too often emphasized is that the system can very quickly be ruined by using fluids other than those supplied by the Lockheed concern. Mineral oils, including shock-absorber oils, have a destructive effect upon the rubber seals used in the system and it is almost equally harmful to employ petrol or paraffin (instead of brake fluid) for "washing out." Apart from oil, petrol, etc., there are substitute fluids which should mostly be regarded with great suspicion. They may cause deterioration of the rubbers, corrosion of the brake cylinder bores, gumming or vapour locks. So the best safeguard against leakage and other troubles is always to use the Lockheed fluid; this is not ust an advertising slogan, but a serious recommendation endorsed by the car makers.

The Lockheed Lay-out. A typical chassis lay-out is shown in Fig. 55 which should be studied in conjunction with the more detailed views of the brakes themselves which are given in Fig. 56. In certain makes of car, variations

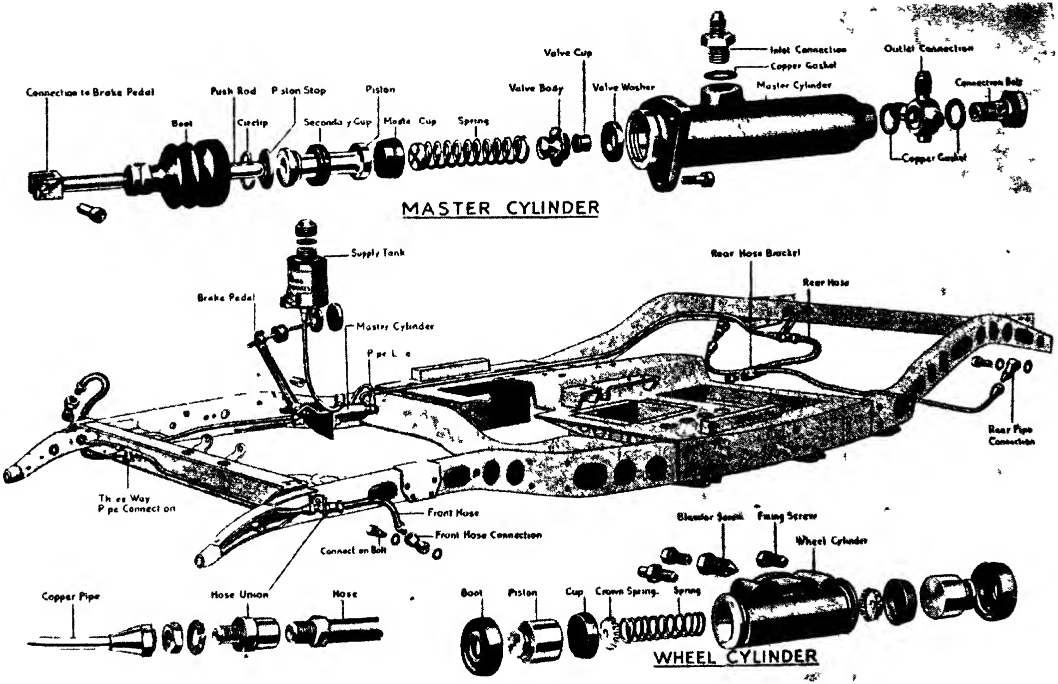

FIG. 55. TYPICAL CHASSIS LAY-OUT FOR A LOCKHEED HYDRAULIC BRAKING SYSTEM WITH THE MASTER CYLINDER PARTS (*top*) AND THE COMPONENTS OF A WHEEL CYLINDER (*bottom*) SHOWN IN DETAIL

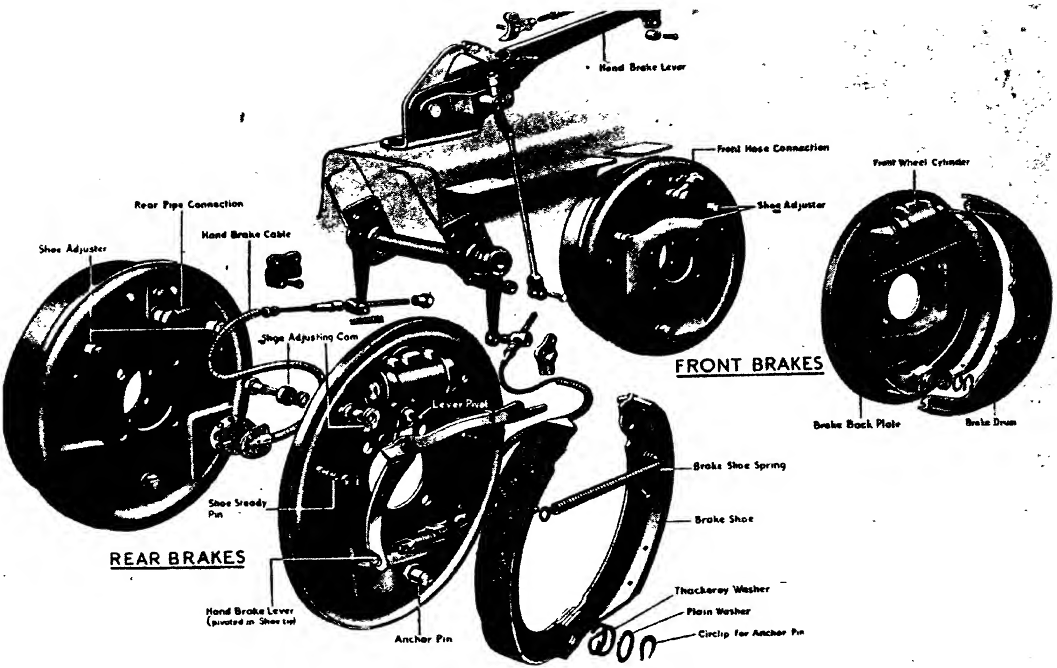

FIG. 56. LOCKHEED BRAKING UNITS

A self-explanatory illustration which also shows the mechanical linkage employed to operate the rear brake shoes from the hand lever from the design shown in these illustrations may be noticed, such as the use of a supply tank (or reservoir) built as a unit with the master cylinder (Fig. 57—instead of separated, as in Fig. 55) or the provision of finger-tip shoe adjusters, slotted shoe anchorages or a tandem master cylinder, details which are explained on later pages.

However, taking the basic system as usually employed, it will be seen that the pedal operates a piston, or plunger, in the master cylinder which is sealed by two cups of special rubber composition. The cylinder is connected to the supply tank by a pipe and a small feed hole; this hole is uncovered whenever the brakes are "off," so as to allow for expansion or contraction. The waist of the piston between the seals is also filled with fluid from the reservoir as an extra safeguard against the ingress of air.

Fluid expelled from the master cylinder when the pedal is depressed creates pressure which is conveyed through pipe lines to the four wheel cylinders. Copper piping is used where possible but, to allow for wheel movements, an extremely strong type of flexible hose is employed to make the connexions between the chassis and axles or, in independent springing systems, between chassis and wheels. If these parts have to be dismantled at any time, the ends of the hose should not be unscrewed. The correct way is to disconnect the copper tube from the hose union, then removing the nut which holds this union to the frame bracket. The axle, or wheel mounting, can now be taken off complete with brake and hose.

On private cars the wheel cylinders are contained within the brake drums and the pipe connexions emerge through the back plates. Each cylinder is fitted with two opposed pistons, sealed by cup washers, and these pistons operate the free ends of the shoes. At the opposite ends the shoes

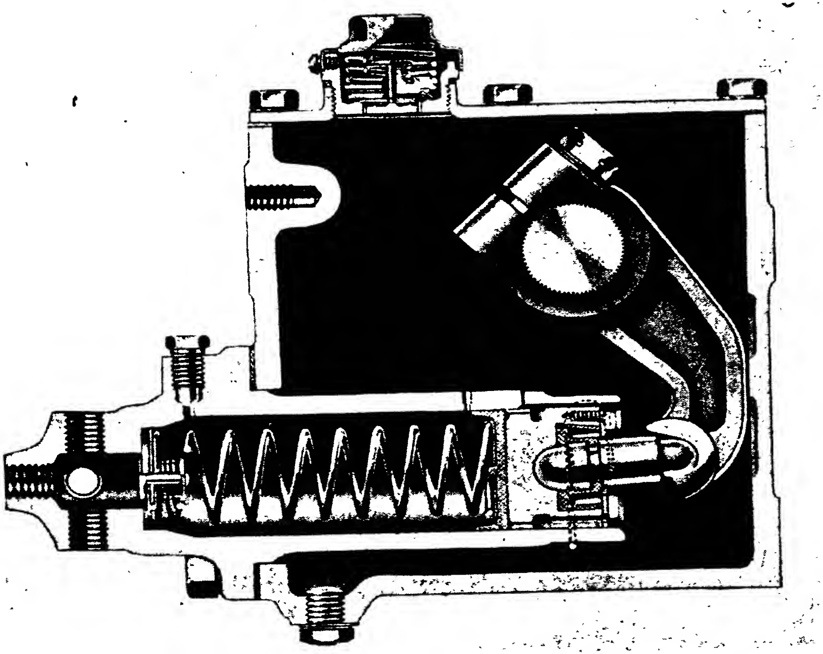

Fig. 57. In Place of a Separate Reservoir the Fluid Container of the Lockheed System can be Designed as a Unit with the Master Cylinder as shown here

hinge upon an anchor pin which is fitted to the back plate in the usual way.

In order to meet the legal requirement of an alternative means for applying the brakes, a mechanical linkage is always arranged between the hand lever and the rear brake shoes. A simple way of carrying this out is shown in Fig. 56. The hand lever is coupled to a cross-shaft from which cables, guided by conduits, are taken to levers located within the brake drums; the cables pass through slots in the back plates. In addition to the shoe clearance adjustment described below, an adjusting nut is provided on the pull-rod operated by the hand lever.

Lockheed Brake Adjustment and Maintenance. In the first place it is essential that the brake pedal should be correctly set in order that the feed hole in the master cylinder shall be uncovered as already described: otherwise, pressure will be built up in the system which may eventually cause the shoes to drag against the drums although the pedal is released. To avoid this condition the pedal should have about $\frac{1}{2}$ in. of free movement from the "off" position to the point at which resistance can be felt. Secondly, the reservoir should be examined at intervals and the fluid maintained to within 1 in. of the top. Rapid loss of fluid indicates leakage and the whole system should be checked over to find the cause.

As wear of the linings takes place the clearance between shoes and drums increases to a point necessitating excessive pedal travel; *all* the brake shoes should then be adjusted. This is done by turning the adjusting nuts outwards (away from the axle) until the shoes make contact, then backing them off until the wheel spins freely when jacked off the ground. The adjustment is self-locking and the nuts require only a fraction of a complete turn; they rotate cams against which the shoes return, under the action of pull-off springs, when the pedal is released. The nuts and cams can be seen in Fig. 56. There is also an alternative type of adjuster (not shown) which consists of a screwed connexion between each shoe and the piston by which it is operated.

The hydraulic principle ensures that the loading on each pair of brake shoes shall be identical, so that if the car pulls to the left or right it cannot be corrected by brake adjustment. Other causes must be sought such as differences in the linings used in the various brakes, grease or oil on the linings, scored drums, or protruding rivets.

Air in the system will prevent effective braking because of its compressibility; the pedal action will feel "spongy" or the brakes may fail to act. As a rule, this trouble will only occur if any part of the system has been disconnected, for which reason "bleeding" should always follow work of this kind.

Just above the hose connexion in each back plate is the head of a bleeder screw which enters the wheel cylinder. Taking each brake in turn, unscrew the plug from the end of the bleeder screw and screw in a draining pipe which is part of the service equipment: connected to it is a rubber tube which can be allowed to fall into a clean glass container.

After unscrewing the bleeder screw one turn, depress the pedal quickly and remove the foot to allow it to snap back. This pumps fluid out of the system and should be repeated until the flow into the container is free from bubbles of air. Ten strokes will probably be required for each brake and the supply tank should be kept at least half full throughout the operation. Fluid bled from the system should be allowed to stand for some time before it is used again because of its aerated condition. In cars fitted with a tandem master cylinder, one front and one rear wheel cylinder should be bled simultaneously.

Service depots now employ a pressure bleeding outfit, supplied by the Lockheed Co., which permits a great saving in time as compared with the pedal-pumping process. It consists of a container for fluid in which pneumatic pressure can be built up by connexion with an air line or tyre pump; 25 lb. per sq. in. is recommended. By using a suitable adapter in place of the filler cap, the container can be connected by a pipe to the reservoir on the car, the latter being first filled with fluid. Then, after slackening the four bleeder screws, it is only necessary to turn a tap on the piped connexion when the compressed air will force fluid through the system.

Specialized Lockheed Components. Although not used on a very large number of cars at the time of writing, the following components (developed by the Lockheed Co.) are likely to increase in popularity and are therefore worth attention from anyone interested in brakes.

First, there is the tandem master cylinder; as the name implies, the cylinder is divided into halves by two pistons (Figs. 58, 59). One half is connected to the front brakes and the other to the rear brakes, the advantage being that a leakage in one part of the system does not affect the other part. Consequently, serious loss of fluid (due to damage or other causes) results in the "loss" of two brakes instead of four brakes.

One of the pistons is directly operated from the brake pedal and the other floats between this piston and a stop at the end of the cylinder. Connexions are taken from the centre and the end of the cylinder to the two pairs of brakes, and feed holes are provided which lead into two reservoirs replenished from a common filler. When the pedal is depressed, the primary piston is moved to the right (see Fig. 58), operating one pair of brakes, and the fluid pressure simultaneously moves the secondary (floating) piston which operates the other pair of brakes: the

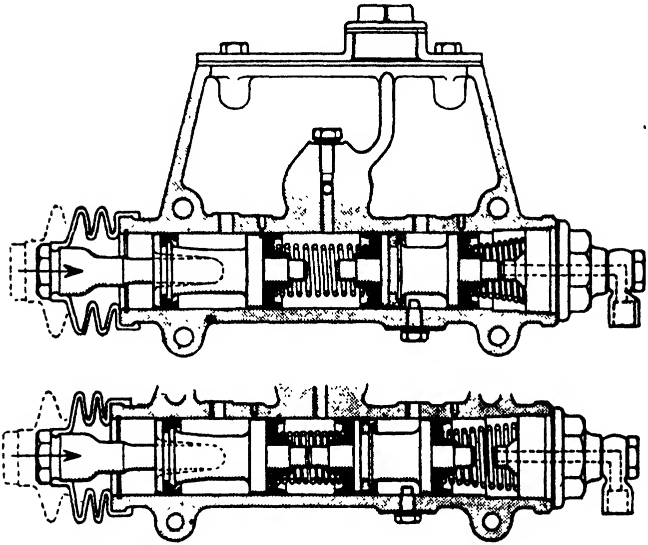

FIG. 58. DIAGRAMS WHICH SHOW THE PRINCIPLE OF THE LOCKHEED TANDEM MASTER CYLINDER, EXPLAINING WHY THE FAILURE OF ONE PAIR OF BRAKES DOES NOT AFFECT THE OPERATION OF THE OTHER PAIR

loading is automatically balanced between the two pistons. If leakage occurs in the front brake system, the primary piston moves farther to the right and operates the secondary piston directly, when projections come into contact. Similarly, if accidental leakage should occur in the rear brake system, the secondary piston will move up to the stop, after which pressure will be built up (by the primary piston) to operate the front brakes. Springs are used to ensure a correct return action under normal conditions.

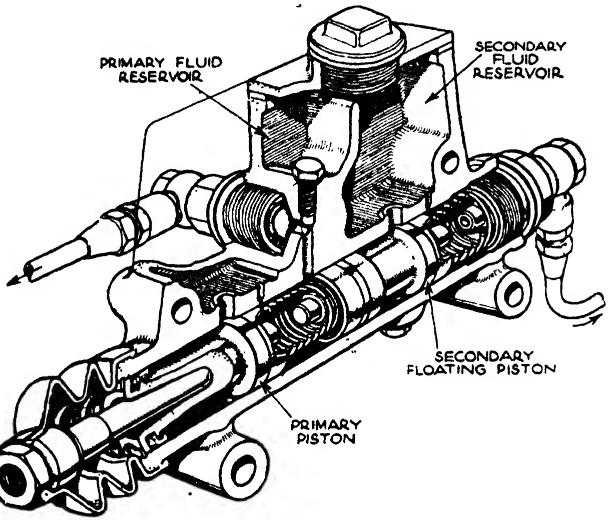

FIG 59. A PERSPECTIVE VIEW OF THE TANDEM MASTER CYLINDER, SHOWN IN SECTION

(This sketch, and the diagrams in Fig. 58, are reproduced by courtesy of "The Autocar")

Another refinement found in some Lockheed systems is a "finger-tip" adjuster which takes the form of a cam acting upon the shoe pivots (Fig. 60). It is turned by a capstan-type adjuster, located behind the back plate, until the shoes meet the drum; thereafter it is only necessary to release the capstan, whereupon it springs back through the angle required to afford the right shoe clearance. This is done by incorporating a cross-pin in the "drive" between the adjuster and the cam, working in a slot of set width

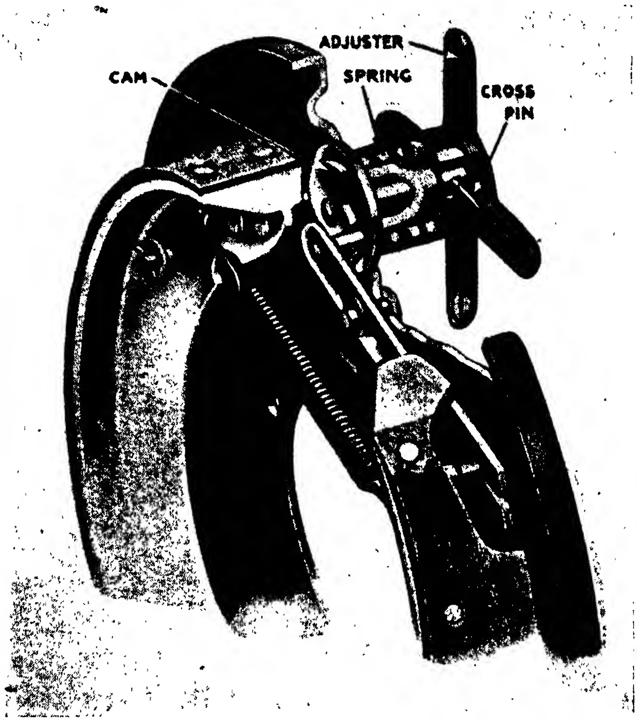

Fig. 60. A "Finger-tip" Adjuster Developed for Lockheed Brakes which Automatically Springs Back to Give the Correct Shoe Clearance

and loaded by a spring. The adjustment can be effected without jacking up the wheel and, as the body of the adjuster is mounted on the slotted back plate with a friction grip, the shoes become automatically centred in the drum.

A comparatively recent innovation is known as the Lockheed slotted shoe. This is designed to permit the shoe

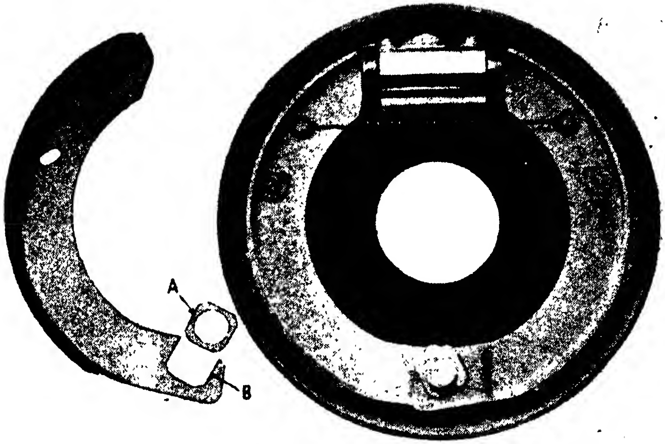

FIG. 61. THE SLOTTED SHOE WITH TRUNNION MOUNTINGS
A Lockheed device which provides an alternative to the ordinary pivot with various advantages (see text)

a certain degree of "float" on its pivot so that it can develop a servo effect without risk of grabbing; a 30 per cent increase in brake torque is claimed as being permitted by this design.

As shown in Fig. 61, the shoe pivot pin carries two square trunnion blocks (*A*) which fit into square recesses (*B*) cut in the shoe webs at a slight angle. When the shoe is applied to the drum (by the usual hydraulic method) it can ride on the trunnion block as it experiences the drag of the drum, obtaining full contact over the whole length of the lining.

An alternative way of operating the shoes has more

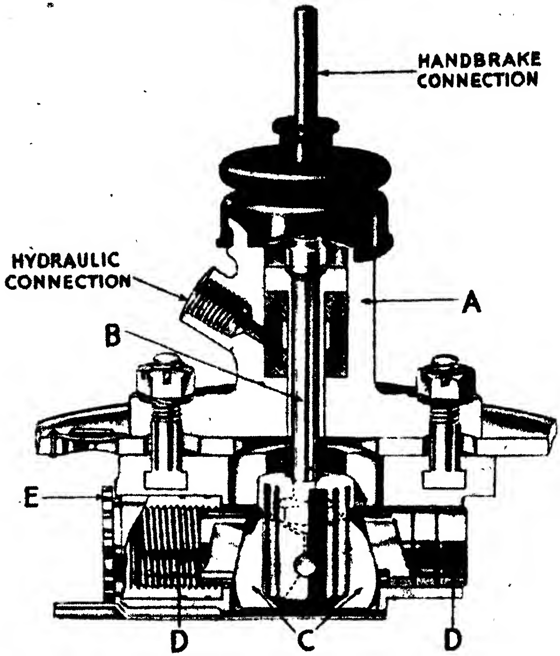

FIG. 62. THE LOCKHEED "BISECTOR EXPANDER"
The cylinder is externally mounted and a pull-rod provides a handbrake connexion to the shoes

recently been introduced by the Lockheed Co., and is shown in Fig. 62. The hydraulic cylinder (*A*) is mounted outside the back plate and works a transverse spindle (*B*) which separates the shoes by means of a mechanism called the "Bisector Expander" (*C*) and plungers (*D*). A single adjuster (*E*) is provided, operated by a screwdriver through a slot, which takes up the clearance in both shoes because the assembly is arranged to "float." One of the advantages of this mechanism is the cooler position of the cylinder; another is the simple mechanical handbrake hook-up which can be arranged to operate the spindle.

CHAPTER VI

OTHER TYPES OF BRAKES, SERVO MOTORS, DRUMS, AND LININGS

THE purpose of this chapter is to provide information regarding a number of points concerned with brakes which could not logically be included in the subject-matter selected for the preceding sections of this book.

First, there is the question of how best to deal with various brakes, which have been quite widely used by individual car manufacturers, other than the proprietary mechanisms already described. Their number has decreased towards vanishing point on the popular types of car so that it really seems unnecessary to include more than two types. These are (*a*) the systems which the Austin Co. employed on the 7 h.p. and 10 h.p. models prior to August, 1936 (when the Girling system was adopted), and (*b*) the mechanical brake which has been fitted to the popular 12 h.p. and 14 h.p. Vauxhall models since their inception in 1933.

The Austin Ten brake was of straighforward mechanical design with pedal and hand lever operating a common cross-shaft and with shoes hinged in pairs to single anchor pins and expanded by cams. A central adjustment was provided in the shape of a winged nut on the pull-rod between the pedal and the cross-shaft; for the handbrake, an adjustable pin was fitted just below the ratchet (Fig. 63). In addition, an adjusting nut (*A*, Fig. 64) was located at the end of each front-brake operating cable where this part passed through the end of the lever on the cam spindle.

Rear brake wear could be taken up independently by adjusting nuts at the forward end of each pull-rod, close to the cross-shaft (Fig. 63).

A somewhat different system was used for the Austin Seven. The main adjustment was placed at the rear end

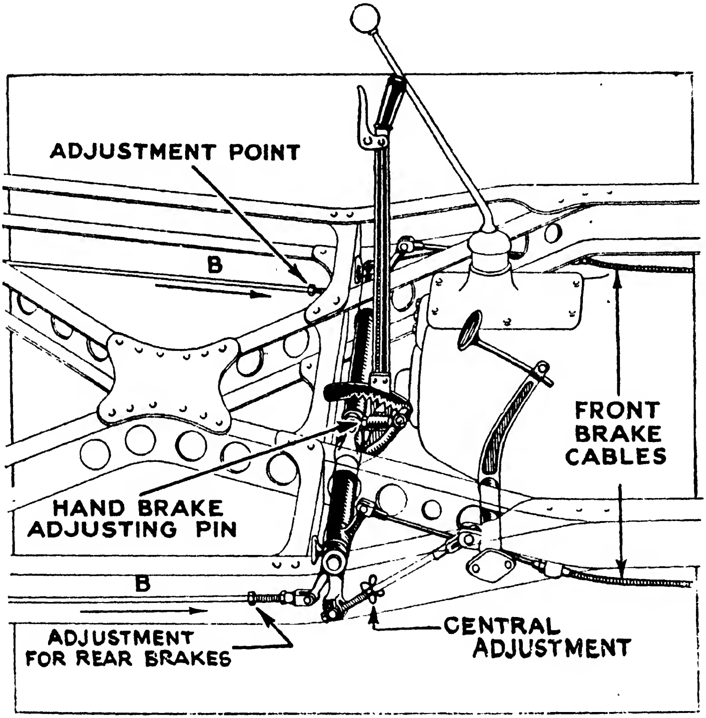

Fig. 63. The Braking System used for the Austin Ten Prior to August, 1936, showing the Central Adjustment, etc.

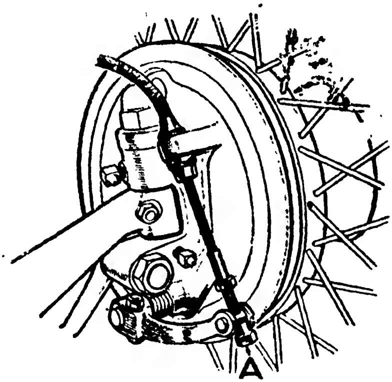

FIG. 64. ANOTHER FEATURE OF THE FORMER AUSTIN TEN BRAKES WAS THE PROVISION OF ADJUSTING NUTS (*A*) ON THE FRONT OPERATING CABLES

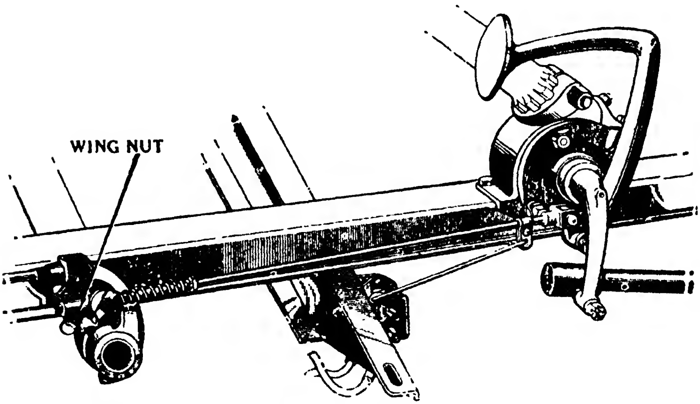

FIG. 65. PRIOR TO AUGUST, 1936, THE MAIN ADJUSTMENT FOR AUSTIN SEVEN BRAKES WAS A WING-NUT ON THE PULL-ROD OPERATED BY THE PEDAL

of the pull-rod, which extended from the pedal to the cross-shaft, and could be reached by lifting a small plate set in the floor just in front of the driver's seat (Fig. 65). Independent adjusters for the rear brakes took the form of screwed forks at the forward ends of the operating cables. After removing a fork from the lever to which it was pinned, the fork could be screwed down to a fresh position, taking care to avoid twisting the cable. A separate wing-nut adjuster was provided just behind the handbrake lever.

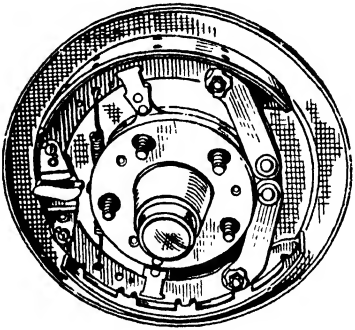

FIG. 66. FRONT BRAKE OF THE 14 H.P. VAUXHALL (1938 MODEL) SHOWING HOW EACH SHOE IS CARRIED BY A RADIUS LINK AND OPERATED BY A CAM

The adjuster (not shown) is fitted to the external cam spindle lever

The Vauxhall system comprises a simple operating gear, with pedal and hand lever interconnected to a single cross-shaft, and four pairs of expanding brake shoes operated by cams. Each shoe is indirectly connected to an anchor pin by means of a link which allows a certain degree of self-adjusting "float" to occur (Fig. 66). The connexion between the link and the shoe is spring loaded, to give frictional control, and the shoes themselves are firmly located laterally by guides fitted to the webs.

Each cam spindle carries an adjustable external lever operated by the corresponding cable from the cross-shaft. After jacking up the wheels and releasing the handbrake, the adjusting screws on the levers are turned until the shoes press firmly against the drums. Then a centralizer bolt on the back plate, close to each lever, should be tapped with a soft hammer before slacking the adjusters, to give just sufficient clearance to allow the wheels to turn freely. Unless the car has been completely dismantled the rest of the system should not be altered; full service instructions are available from the makers if this extra work becomes necessary. These remarks apply only to the 12 h.p. and 14 h.p. models (1938 and earlier), as Lockheed brakes are used for later types of Vauxhall chassis.

Servo Motors

Although a large number of mechanisms designed to assist the driver in applying the brakes has been used in the past, only one type of independent servo motor is at all widely employed nowadays; namely, the Clayton-Dewandre. As already explained, various brakes utilize the self-applying characteristics of the leading shoe, but here we are concerned with the alternative plan represented by a servo mechanism separate from the shoes and acting upon the operating gear.

It is essential that any such servo motor should (1) work progressively in proportion to the extent to which the driver applies the brake pedal; (2) enable the driver to "feel" the extent of effort exerted; (3) permit the brakes to be worked by pedal directly should the servo action fail from any cause. These requirements are all met by the ingenious design of the Dewandre device, and its action can easily be followed from the diagram in Fig. 67 by reading the notes, numbered 1 to 19, in sequence.

The pedal pull-rod is in direct connexion with the rod which operates the cross-shaft and brakes, but an intermediate lever enables the driver's effort to be supplemented by atmospheric pressure, acting upon a large piston (C),

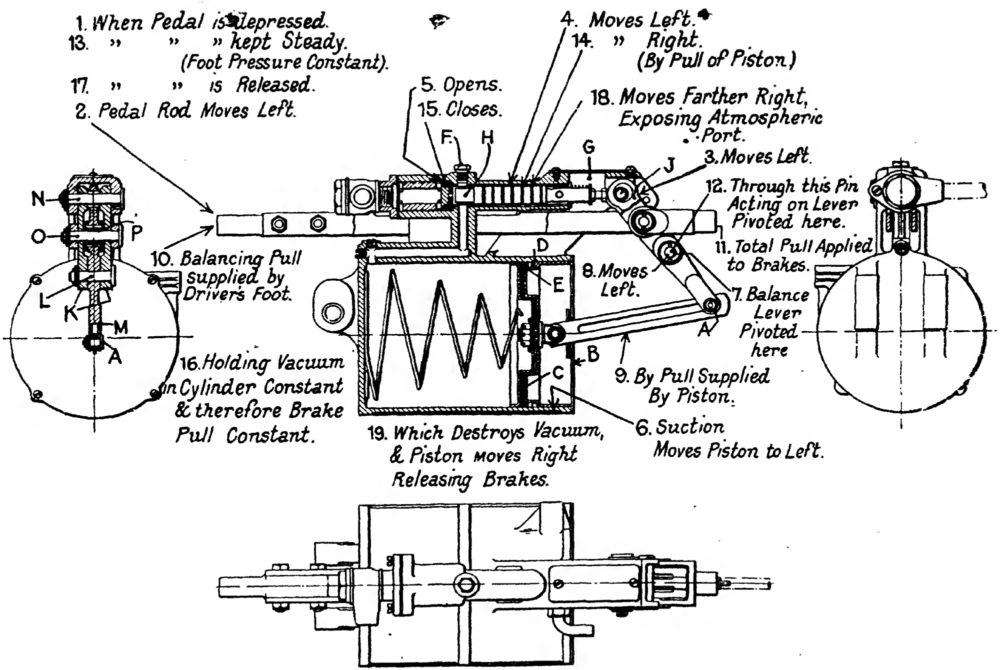

FIG. 67. DIAGRAMS WHICH ENABLE THE ACTION OF THE CLAYTON-DEWANDRE VACUUM SERVO MOTOR TO BE UNDERSTOOD IF THE DESCRIPTIVE NOTES ARE READ IN NUMERICAL SEQUENCE

when the opening of a value (H) allows the engine to exhaust air from a cylinder. This is simply done by connecting the blind end of the cylinder to the inlet pipe of the engine. The valve (H) between the two is operated through a loose pin joint (N, J) which allows a definite amount of play. In brief, the piston "follows up" as the pedal is depressed, loading the pull-rod to a greater and still greater extent and so aiding the driver. Should this servo action fail, the brakes can still be applied directly, although greater physical effort is then needed. In some cars a reservoir is included in the vacuum circuit to improve the operation of the servo device.

Beyond the obvious need for cleaning and oiling the various external joints, maintenance work requires only the occasional addition of a few drops of oil through the plug (F) fitted to the casing above the valve. An adjustment is provided on the valve rod and the clearance at the end of the valve (H) should be half the clearance at the loose pin joint (J) in the mechanism shown in Fig. 67. On no account should the length of the rod between the servo and the brake pedal be altered once it has been set correctly; all brake adjustments should be made behind the servo. After long use the piston leather (D) may cease to be pliable and need renewal. The expanding ring (E) beneath it may have to be slightly opened out to ensure proper contact between the leather and the cylinder.

Before concluding this section mention may be made of the "friction clutch" type of servo motor, driven from the gearbox, which has for many years been used on Rolls Royce cars and is also employed on the 4¼-litre and 3½-litre Bentley chassis. Its adjustment is not, however, likely to be undertaken by anyone other than the maker's representatives.

Brake Drums

In view of the close clearances desirable between linings and drum, and the need for smoothness in braking, it is extremely important that the internal drum surface should run truly and should be circular. Any ovality is also to be deprecated because of the way in which it tends to make the brake "grab" at low speeds. The working surface of the drum should be smooth because otherwise it will soon damage the lining. Scoring of the brake drum is most likely to occur when the material is a low carbon steel, chosen for its facility in press work rather than for its suitability as a brake component. However, this criticism is not so severe when a non-metallic lining is used in place of the materials in which brass wire is woven.

Cast-iron drums have become rather popular and give very satisfactory results provided that the interior surface is really smooth; "finish" is more important than in the case of steel. Modern iron alloys are free from the brittleness usually associated with cast iron and this form of drum, by suitable external ribbing, can be made much more rigid than a pressed steel drum. If the working surface becomes damaged, it is essential that it should be given a ground finish after machining, whereas with a steel drum a machined finish is satisfactory.

Rigidity in the drums is important because any "yielding" increases the shoe travel required and also causes a departure from true circularity of the working surface. It may also result in the drum becoming "bell-mouthed"; these forms of distortion are shown by diagrams in Fig. 68. It is not usually advisable to take more than a very small amount of metal out of a pressed steel drum, when turning it to remove scoring, because the thin section will not be rigid enough to hold its shape.

Even when great care has been used to make sure that a brake drum is "trued up" accurately on the machine, it may still be found to run out when mounted on the car, particularly after fitting the wheel. Such cases are usually

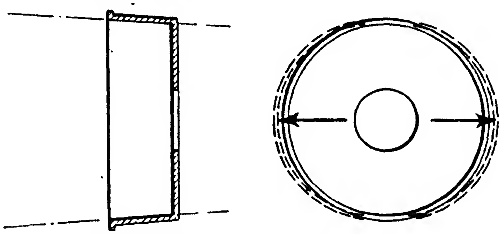

FIG. 68. DIAGRAMS WHICH SHOW TWO KINDS OF BRAKE DRUM DISTORTION; OVALITY (*right*) AND "BELL-MOUTHING" (*left*)
Arrows indicate the expanding forces applied by the shoes to the drum

traceable to a wheel centre which is not flat so that the effect of tightening the wheel nuts is to distort the brake drum.

CHOICE OF BRAKE LININGS

Great progress has been made in the development of lining materials which meet a wide variety of working conditions, provide a good resistance to wear, and do not tend to damage the working surface of the drum. Another important property of a good modern lining is that the coefficient of friction will remain uniform over a wide range of temperatures. Just what this coefficient may be depends mainly upon the properties of the lining; the material of the drum has little effect. Most of the materials commercially available give coefficients between 0·3 and 0·45, and it is important to use the grade recommended by the car or brake manufacturer. Particularly is this the case when servo shoes are relined, because too high a coefficient will at once lead to grabbing trouble by increasing the servo effect (see page 64).

In order to facilitate replacement, many concerns supply linings as "spare parts," cut and drilled ready for use. The old linings must be carefully removed from the shoes, in order to avoid damaging the metal, after which the new linings are firmly secured by countersunk rivets. Clamps can be had which hold the lining firmly for riveting (Fig.

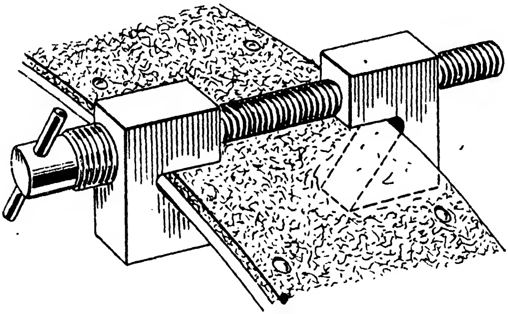

FIG. 69. A CONVENIENT CLAMP FOR SECURING LININGS TO SHOES BEFORE DRILLING AND RIVETING, AVAILABLE FROM MESSRS. FERODO

69). Many garages employ machines on which relined brake shoes can be mounted so that the new working surface can be trued either by grinding or milling. This practice obviates most of the "bedding in" which is otherwise necessary after the shoes have been fitted to the car. On no account should grease or oil be allowed to reach the linings; even handling them with greasy fingers will prevent the best results from being attained.

A plan of replacement has been developed by brake makers whereby shoes with worn linings can be exchanged for shoes with new linings, ready fitted and machined, which are available at moderate prices in all the popular sizes.

The composition of all brake linings includes asbestos, which is a fibrous substance with a high resistance to heat. It is bonded with various synthetic materials, and the skill and care with which this process is undertaken have as great an effect upon the life and performance of the lining as have the materials chosen for the bonding medium. For these reasons it is a false economy to use linings of inferior quality.

At one time brass wire of fine gauge was widely used as a reinforcement for the lining and was often blamed for scoring the softer kinds of steel drum. Other kinds of lining were developed which could do without the help of wire, but latterly softer metals, such as aluminium, have again popularized metallic reinforcement. "Die-pressed" linings are made from materials which have the property of becoming rigid when moulded under high pressure and are sold as segments which can be fitted to the shoes without bending. Anyone interested in a closer study of the subject will find an excellent account of the properties of modern linings in a paper presented before the Institution of Automobile Engineers in December, 1936.

Brake Squeak

This trouble, at one time very prevalent, has fortunately become less evident in modern cars. It can arise from any one of a very large variety of causes so that there is no simple cure applicable to all cases. The noise is undoubtedly produced by a high-frequency vibration which may be set up by a leading shoe if the coefficient of friction rises temporarily and increases the servo effect. For this reason squeak will often occur after a car has been standing in a damp atmosphere, which rusts the polished surfaces of the drum, but disappears as soon as the film of rust is dispersed by braking.

These facts also explain why a squeak can often be cured by tapering off the ends of the linings which, as explained earlier, exert the maximum effect upon self-applying qualities in the shoe. Exposed rivet heads can set up vibration and squeak as can also a lining which gives a higher coefficient than is desirable for the design of brake in question. Yet another possibility is the presence of abraded material and road grit in the drum; back plates which effectively exclude water provide no means of escape for the particles which collect in the drum.

Squeak is certainly magnified by thin drums, for which reason it can often be cured by clamping or shrinking a steel band around the external surface. Shoe vibration is another fault leading to squeak so that this trouble is sometimes curable by loading the shoe webs sideways with spring washers fitted to pins passing through slots and riveted to the back plate. In many cars the brake shoes are designed to be located in some such manner so that squeak may indicate that the locating device should be tightened.

Brakes which Fade

On a long descent or in heavy traffic the effectiveness of the brakes often tends to fall off. requiring an increase in pedal pressure; this is a temperature effect known as "fading." Prolonged or repeated braking heats the drums and their expansion increases the shoe clearances in a way resembling the effect of lining wear.

Two factors tend to aggravate fading troubles. These are (1) a poor lining material, the coefficient of friction of which decreases rapidly at high temperatures; (2) thin drums of insufficient rigidity which, by expanding in "bell-mouthed" fashion, reduce the effective area of lining contact. Rear brakes are more often subject to fading than front brakes because they are not so open to cooling by the flow of air past the car.

Chassis Faults that Affect Brake Performance

It would not do to conclude this book without a short reference to the fact that good braking cannot be obtained

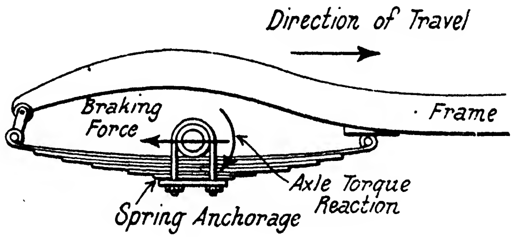

Fig. 70. Braking Force and Torque Reaction Stress the Anchorage Securing the Spring to the Axle and also Impose Loads upon the Spring and its Attachments to the Frame

if there are certain defects in the chassis. The reason for this is that the brakes can only retard the *wheels*, and therefore their effect upon the car as a whole must to some extent depend upon the mechanism which connects the wheels to the car.

In a conventional design the wheels are carried by axles which, in turn, are joined to the chassis frame by semi-elliptic springs. Braking forces tend to drag the axle back along the springs and also tend to pull the springs off the car. Furthermore the back plates carrying the shoes, together with the axle, would turn with the drums except for the restraint exercised by the springs.

For these reasons powerful brakes impose quite severe stresses upon the anchorages used to secure the springs to the axle and upon the attachments between the springs and the frame (Fig. 70). In the case of front brakes the steering heads are also stressed. Slackness in any of these parts of the mechanism will allow play to develop, which is a common cause of brake "judder" under emergency stopping conditions. It is scarcely necessary to add that the condition and inflation pressure of the tyres also affect brake performance.

In independent springing systems the connexion between wheels and car takes the more rigid form of radius arms, but here again good results are dependent upon absence of play in the pivots and joints, of which there is often a considerable number.

No matter what kind of suspension is used, correct wheel alinement is important. Another point is that the operating gear has to convey the pedal effort from the car to the brakes with due allowance for springing movements. In some mechanical systems this allowance is upset by play in the connexions between wheels and chassis and/or by incorrect alinement. These facts are emphasized because many a service engineer has struggled to overcome faults in brake performance, by means of shoe adjustment, without success, for the simple but not always obvious reason that defects in the chassis were the real cause of the trouble.

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