Subject :vehicle technology/2	2/	:
Weekly Hours: Theoretical: 1	1:	:
Tutorial: 1	1	:
Experimental : 3	3:	
Units: 5	5	:

week	Contents	_____	_____
1.		المقدمة: انواع منظومات السيطرة	1.
2.		منظومة الوقود في محركات البنزين	2.
3.		المغذيات	3.
4.		المغذي البسيط	4.
5.		المغذيات الحديثة، المنظومات	5.
6.		منظومة سرعة التكاسل والسرعة المنخفضة	6.
7.		منظومة السرعة العالية	7.
8.		منظومة التعجيل ، القدرة، مانع التثليج	8.
9.		صمام الخنق الذاتي والكهربائي، تأثير الارتفاع على المغذي	9.
10.		المغذيات: سولكس، SU ، زينيث	10.
11.		حقن الوقود في المحركات	11.
12.		مضخة حقن وقود الديزل/ الاعطال	12.
13.		الدوائر الكهربائية في السيارة	13.
14.		البطارية، محرك بدء الحركة	14
15.		الملف، المكثف	15.
16.		موزع الشرر، قاطع الاتصال، شمعات القدح	16.
17.		دائرة الشحن، مولد التيار المتناوب والمستمر	17.
18.		=	18.
19.		قاطع التيار، منظم التيار، منظم الجهد، المقاومات	19.
20.		=	20.
21.		منظومة الاشتعال الميكانيكية، الترانسيستور الالكترونية	21.
22.		=	22.
23.		نظام الايقاف (الكبح) في السيارة	23.
24.		انظمة الايقاف الحديثة	24
25.		العجلات ، الاطارات، الموازنة	25.
26.		=	26.
27.		*التكييف في السيارة*	27.
28.		تشخيص ومعالجة الاعطال في اجهزة التكييف	28.
29.		التطورات الحديثة في السيارة	29.
30.		المنظومات الحديثة في السيارات(العقل والمتحسسات)	30.

مفردات المنهج التدريبي لطلبة الجامعة التكنولوجية في وحدة السيارات

المرحلة الرابعة للعام الدراسي 2008/2007 لقسم هندسة المكائن والمعدات / فرع السيارات

عدد ساعات التدريب	وقت التدريب	الموضوع	الأسبوع
2ساعة	10.30-12.30	فتح صندوق تروس أوتوماتيكي	ألأول
2ساعة	10.30-12.30	صيانة وفحص صندوق التروس الأوتوماتيكي	الثاني
2ساعة	10.30-12.30	فتح وتركيب صندوق التروس	الثالث
2ساعة	10.30-12.30	فتح وتركيب المحور الخلفي	الرابع
2ساعة	10.30-12.30	منظومة تكييف / صيانة	الخامس
2ساعة	10.30-12.30	فتح وتركيب مضخة حقن وقود الديزل	السادس
2ساعة	10.30-12.30	فتح وتركيب الرشاشات وفحصها	السابع
2ساعة	10.30-12.30	تنظيم مضخة حقن وقود الديزل	الثامن
2ساعة	10.30-12.30	فحص وصيانة المنظومة الكهربائية	التاسع
2ساعة	10.30-12.30	فحص وصيانة منظومة ألأشتعال الأعتيادية وألألكترونية	العاشر
2ساعة	10.30-12.30	فحص وصيانة منظومة تقديم وتأخير الشرارة	الحادي عشر
2ساعة	10.30-12.30	فحص وصيانة أجهزة القيادة	الثاني عشر
2ساعة	10.30-12.30	فحص وصيانة منظومة التعليق	الثالث عشر
2ساعة	10.30-12.30	فحص وصيانة الموقفات	الرابع عشر

9 Steering

9.1 Steering gearbo   fundamental design

9.1.1 Steering gearbox angular ratios

The steering gearbox has two main functions: it produces a gear reduction between the input steering wheel and the output drop arm (Pitman arm) and it redirects the input to output axis of rotation through a right angle.

Generally, the steering road wheel stub axles must be capable of twisting through a maximum steering angle of $40°$ either side of the straight ahead position. The overall angular gear ratio of a steering gearbox may be as direct as 12:1 for light small vehicles or as indirect as 28:1 for heavy vehicles. Therefore, lock to lock drop arm angular displacement amounts to $80°$ and with a 12:1 and 28:1 gear reduction the number of turns of the steering wheel would be derived as follows:

$$\text{Lock to lock steering wheel turns for 12:1} = \frac{80 \times 12}{360}$$
$$= 2\ 66 \text{ revolutions}$$

$$\text{Lock to lock steering wheel turns for 28:1 reduction} = \frac{80 \times 28}{360}$$
$$= 6\ 22 \text{ revolutions}$$

From these results plotted in Fig. 9.1 it can be seen that the 12:1 reduction needs the steering wheel to be rotated 1.33 turns from the straight ahead position. The 28:1 reduction will require more than twice this angular displacement, namely 3.11 turns. Thus with the 12:1 gear reduction, the steering may be heavy but can be turned from the straight ahead position to full lock and back again relatively quickly. However the 28:1 reduction will provide a light steering wheel but the vehicle will be compelled to corner much slower if the driver is to be able to complete the manoeuvre safely.

9.1.2 Screw and nut steering gear mechanism
(Fig. 9.2)

To introduce the principles of the steering gearbox, the screw and nut type mechanism will be examined as this is the foundation for all the other types of steering box gear reduction mechanisms.

A screw is made by cutting an external spiral groove around and along a cylindrical shaft, whereas a nut is produced by cutting a similar spiral groove on the internal surface of a hole made in a solid block.

The thread profile produced by the external and internal spiral grooves may take the form of a vee, trapezoidal, square or semicircle, depending upon the actual application.

A nut and screw combination (Fig. 9.2) is a mechanism which increases both the force and

Fig. 9.1 Relationship of overall angular gear ratio and steering wheel lock to lock revolutions

Fig. 9.2 Screw and nut friction steering gear mechanism

movement ratios. A small input effort applied to the end of a perpendicular lever fixed to the screw is capable of moving a much larger load axially along the screw provided that the nut is prevented from rotating.

If the screw is prevented from moving longitudinally and it revolves once within its nut, the nut advances or retracts a distance equal to the axial length of one complete spiral groove loop. This distance is known as the thread pitch or lead (p).

The inclination of the spiral thread to the perpendicular of the screw axis is known as the helix angle $\propto$. The smaller the helix angle the greater the load the nut is able to displace in an axial direction. This is contrasted by the reduced distance the nut moves forwards or backwards for one complete revolution of the screw.

The engaged or meshing external and internal spiral threads may be considered as a pair of infinitely long inclined planes (Fig. 9.3(a and b)). When the nut is prevented from turning and the screw is rotated, the inclined plane of the screw slides relative to that of the nut. Consequently, a continuous wedge action takes place between the two members in contact which compels the nut to move along the screw.

Because of the comparatively large surface areas in contact between the male and female threads and the difficulty of maintaining an adequate supply of lubricant between the rubbing faces, friction in this mechanism is relatively high with the result that mechanical efficiency is low and the rate of wear is very high.

A major improvement in reducing the friction force generated between the rubbing faces of the threads has been to introduce a series of balls (Fig. 9.4) which roll between the inclined planes as the screw is rotated relatively to the nut.

The overall gear ratio is achieved in a screw and nut steering gearbox in two stages. The first stage occurs by the nut moving a pitch length for every one complete revolution of the steering wheel. The second stage takes place by converting the linear movement of the nut back to an angular one via an integral rocker lever and shaft. Motion is imparted to the rocker lever and shaft by a stud attached to the end of the rocker lever. This stud acts as a pivot and engages the nut by means of a slot formed at right angles to the nut axis.

Fig. 9.4 Screw and nut recirculating ball low friction gear mechanism

Fig. 9.3 (a and b) Principle of screw and nut steering gear

orward and reverse efficiency The forward efficiency of a steering gearbox may be defined as the ratio of the output work produced at the drop arm to move a given load to that of the input work done at the steering wheel to achieve this movement.

i e   Forward efficiency =

$$\frac{\text{Output work at drop arm}}{\text{Input work at steering wheel}} \times 100$$

Conversely the reverse efficiency of a steering gearbox is defined as the ratio of the output work produced at the steering wheel rim causing it to rotate against a resisting force to that of the input work done on the drop arm to produce this movement.

i e   Reverse efficiency =

$$\frac{\text{Output work at steering wheel}}{\text{Input work at drop arm}} \times 100$$

A high forward efficiency means that very little energy is wasted within the steering gearbox in overcoming friction so that for a minimum input effort at the steering wheel rim a maximum output torque at the drop arm shaft will be obtained.

A small amount of irreversibility is advantageous in that it reduces the magnitude of any road wheel oscillations which are transmitted back to the steering mechanism. Therefore the vibrations which do get through to the steering wheel are severely damped.

However, a very low reverse efficiency is undesirable because it will prevent the self-righting action of the kingpin inclination and castor angle straightening out the front wheels after steering the vehicle round a bend.

Relationship between the forward and reverse efficiency and the heli angle (Figs 9.3, 9.4 and 9.5) The forward efficiency of a screw and nut mechanism may be best illustrated by considering the inclined plane (Fig. 9.3(a)). Here the inclined plane forms part of the thread spiral of the screw and the block represents the small portion of the nut. When the inclined plane (wedge) is rotated anticlockwise (moves downwards) the block (nut) is easily pushed against whatever load is imposed on it. When the screw moves the nut the condition is known as the *forward efficiency*.

In the second diagram (Fig. 9.3(b)) the block (nut) is being pressed towards the right which in turn forces the inclined plane to rotate clockwise (move upward), but this is difficult because the

helix angle (wedge angle) is much too small when the nut is made to move the screw. Thus when the mechanism is operated in the reverse direction the efficiency (reverse) is considerably lower than when the screw is moving the nut. Only if the inclined plane angle was to be increased beyond 40° would the nut be easily able to rotate the screw.

The efficiency of a screw and nut mechanism will vary with the helix angle (Fig. 9.5). It will be at a maximum in the region of 40–50° for both forward and reverse directions and fall to zero at the two extremes of 0 and 90° (helix angle). If both forward and reverse efficiency curves for a screw and nut device were plotted together they would both look similar but would appear to be out of phase by an amount known as the friction factor.

Selecting a helix angle that gives the maximum forward efficiency position (A) produces a very high reverse efficiency (A) and therefore would feed back to the driver every twitch of the road wheels caused by any irregularities on the road surface. Consequently it is better to choose a smaller helix angle which produces only a slight reduction in the forward efficiency (B) but a relatively much larger reduced reverse efficiency (B–B). As a result this will absorb and damp the majority of very small vibrations generated by the tyres rolling over the road contour as they are transmitted through the steering linkage to the steering gearbox.

A typical value for the helix angle is about 30° which produces forward and reverse efficiencies of about 55% and 30% without balls respectively. By incorporating recirculating balls between the screw and nut (Fig. 9.4) the forward and reverse efficiencies will rise to approximately 80% and 60% respectively.
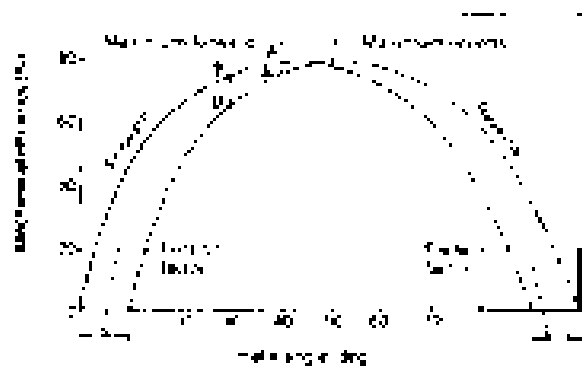

Fig. 9.5 Efficiency curves for a screw and nut recirculating ball steering gear

Summary and forward and reverse efficiency The efficiency of a screw and nut mechanism is relatively high in the forward direction since the input shaft screw thread inclined plane angle is small. Therefore a very large wedge action takes place in the forward direction. In the reverse direction, taking the input to be at the steering box drop arm end, the nut threads are made to push against the steering shaft screw threads, which in this sense makes the inclined plane angle very large, thus reducing the wedge advantage. Considerable axial force on the nut is necessary to rotate the steering shaft screw in the reverse direction, hence the reverse efficiency of the screw and nut is much lower than the forward efficiency.

9.1.3 Cam and peg steering gearbox (Fig. 9.6)

With this type of steering box mechanism the conventional screw is replaced by a cylindrical shaft supported between two angular contact ball bearings (Fig. 9.6). Generated onto its surface between the bearings is a deep spiral groove, usually with a variable pitch. The groove has a tapered side wall profile which narrows towards the bottom.

Positioned half-way along the cam is an integral rocker arm and shaft. Mounted at the free end of the rocker arm is a conical peg which engages the tapered sides of the groove. When the camshaft is rotated by the steering wheel and shaft, one side of the spiral groove will screw the peg axially forward or backward, this depending upon the direction the cam turns. As a result the rocker arm is forced to pivot about its shaft axis and transfers a similar angular motion to the drop arm which is attached to the shaft's outer end.

To increase the mechanical advantage of the cam and peg device when the steering is in the straight ahead position, the spiral pitch is generated with the minimum pitch in the mid-position. The pitch progressively increases towards either end of the cam to give more direct steering response at the expense of increased steering effort as the steering approaches full lock.

Preload adjustment of the ball races supporting the cam is provided by changing the thickness of shim between the end plate and housing. Spring loaded oil seals are situated at both the drop arm end of the rocker shaft and at the input end of the camshaft.

Early low efficiency cam and peg steering boxes had the peg pressed directly into a hole drilled in the rocker arm, but to improve efficiency it is usual
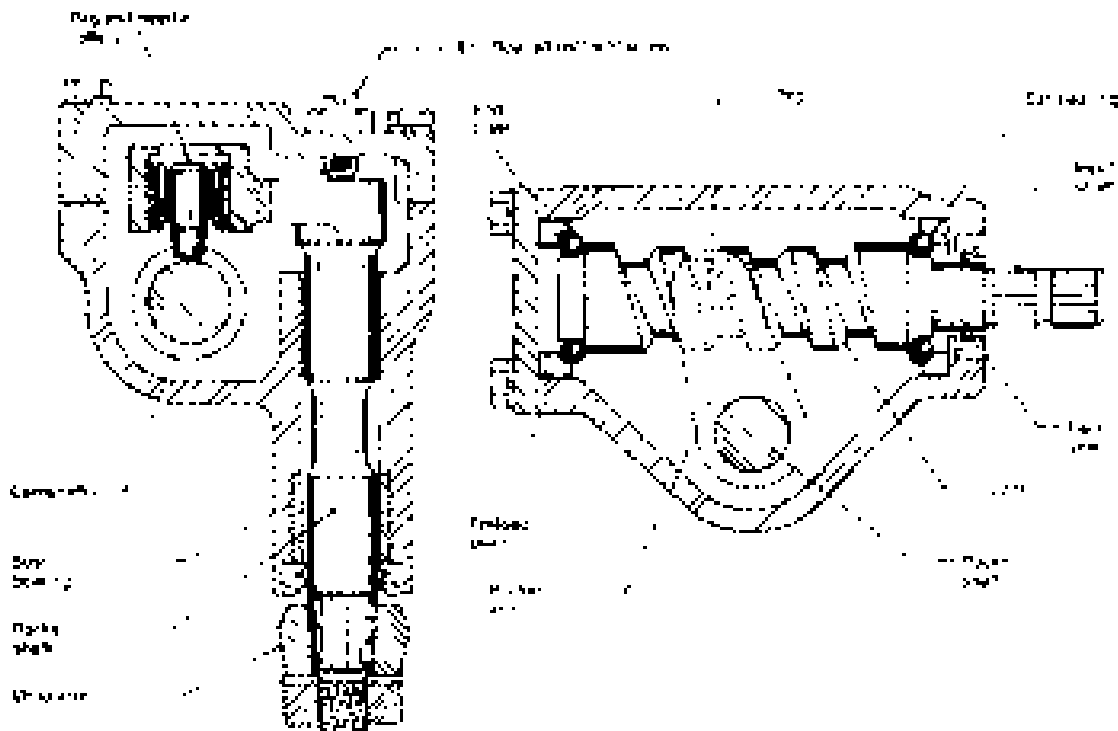
Fig. 9.6 Cam and peg steering type gearbox

314

to support the peg with needle rollers assembled inside an enlarged bore machined through the rocker arm. For heavy duty applications, and where size permits, the peg can be mounted in a parallel roller race with a combined radial and thrust ball race positioned at the opposite end to the peg's tapered profile. An alternative high efficiency heavy duty arrangement for supporting the peg uses opposing taper roller bearings mounted directly onto the rocker arm, which is shaped to form the inner tracks of the bearings.

Cam and peg mechanisms have average forward and reverse efficiencies for pegs that are fixed in the rocker arm of 50% and 30% respectively, but needle mounted pegs raise the forward efficiency to 75% and the reverse to 50%.

To obtain the correct depth of peg to cam groove engagement, a rocker shaft end play adjustment screw is made to contact a ground portion of the rocker shaft upper face.

The rocker shaft rotates in a bronze plain bearing at the drop arm end and directly against the bearing bore at the cam end. If higher efficiency is required, the plain bush rocker shaft bearing can be replaced by needle bearings which can raise the efficiency roughly 3–5%.

9.1.4 orm and roller type steering gearbox
(Fig. 9.7)

This steering gear consists of an hourglass-shaped worm (sometimes known as the cam) mounted between opposing taper roller bearings, the outer race of which is located in the end plate flange and in a supporting sleeve at the input end of the worm shaft (Fig. 9.7). Shims are provided between the end plates and housing for adjusting the taper roller bearing preload and for centralizing the worm relative to the rocker shaft.

Engaging with the worm teeth is a roller follower which may have two or three teeth. The roller follower is carried on two sets of needle rollers supported on a short steel pin which is located between the fork arm forged integrally with the rocker shaft.

In some designs the needle rollers are replaced by ball races as these not only support radial loads but also end thrust, thereby substantially reducing frictional losses.

Fig. 9.7 Worm and roller type steering gearbox

315

The rocker shaft is supported on two plain bushes; one located in the steering box and the other in the top cover plate. End thrust in both directions on the rocker shaft is taken by a shouldered screw located in a machined mortise or 'T' slot at one end of the rocker shaft.

To adjust the depth of mesh of the worm and roller (Fig. 9.7), move the steering wheel to the mid-position (half the complete number of turns of the steering wheel from lock to lock), screw in the end thrust shouldered screw until all free movement is taken up and finally tighten the lock nut (offset distance being reduced).

Centralization of the cam in relation to the rocker shaft roller is obtained when there is an equal amount of backlash between the roller and worm at a point half a turn of the steering wheel at either side of the mid-position. Any adjustment necessary is effected by the transference from one end plate to the other of the same shims as those used for the taper bearing preload (i.e. the thickness of shim removed from one end is added to the existing shims at the other).

The forward and reverse efficiencies of the worm roller gear tend to be slightly lower than the cam and peg type of gear (forward 73% and reverse 48%) but these efficiencies depend upon the design to some extent. Higher efficiencies can be obtained by incorporating a needle or taper roller bearing between the rocker shaft and housing instead of the usual plain bush type of bearing.

9.1.5 Recirculating ball nut and rocker lever steering gearbox (Fig. 9.8)

Improvement in efficiency of the simple screw and nut gear reduction is achieved with this design by replacing the male and female screw thread by semicircular grooves machined spirally onto the input shaft and inside the bore of the half nut and then lodging a ring of steel balls between the internal and external grooves within the nut assembly (Fig. 9.8).

The portion of the shaft with the spiral groove is known as the *worm*. It has a single start left hand spiral for right hand drive steering and a right hand spiral for left hand drive vehicles.

Fig. 9.8 Recirculating ball nut and rocker lever steering type gearbox

The worm shaft is supported between two sets of ball races assembled at either end normally in an aluminium housing. Steel shims sandwiched between the detachable plate at the input end of the shaft provide adjustment of the bearing pre-load. Situated on the inside of the end plate is a spring loaded lip seal which contacts the smooth surface portion of the worm shaft.

Assembled to the worm is a half nut with a detachable semicircular transfer tube secured to the nut by a retainer and two bolts. The passage formed by the grooves and transfer tube is fitted with steel balls which are free to circulate when the worm shaft is rotated.

The half nut has an extended tower made up of a conical seat and a spigot pin. When assembled, the conical seat engages with the bevel forks of the rocker lever, whereas a roller on the nut spigot engages a guide slot machined parallel to the worm axis in the top cover plate. When the worm shaft is rotated, the spigot roller engaged in its elongated slot prevents the nut turning. Movement of the nut along the worm will result in a similar axial displacement for the spigot roller within its slot.

End float of the rocker lever shaft is controlled by a spring loaded plunger which presses the rocker lever bevel forks against the conical seat of the half nut.

The rocker lever shaft is supported directly in the bore of the housing material at the worm end but a bronze bush is incorporated in the housing at the drop arm end of the shaft to provide adequate support and to minimize wear. An oil seal is fitted just inside the bore entrance of the rocker shaft to retain the lubricant within the steering box housing.

The worm shaft has parallel serrations for the attachment of the steering shaft, whereas the rocker shaft to drop arm joint is attached by a serrated taper shank as this provides a more secure attachment.

Forward and reverse efficiencies for this type of recirculating ball and rocker lever gear is approximately 80% and 60% respectively.

9.1.6 Recirculating ball rack and sector steering gearbox (Fig. 9.9)

To reduce friction the conventional screw and nut threads are replaced by semicircular spiral grooves (Fig. 9.9). These grooves are machined externally around and along the cylindrically shaped shaft which is known as the worm and a similar groove is machined internally through the bore of the nut.
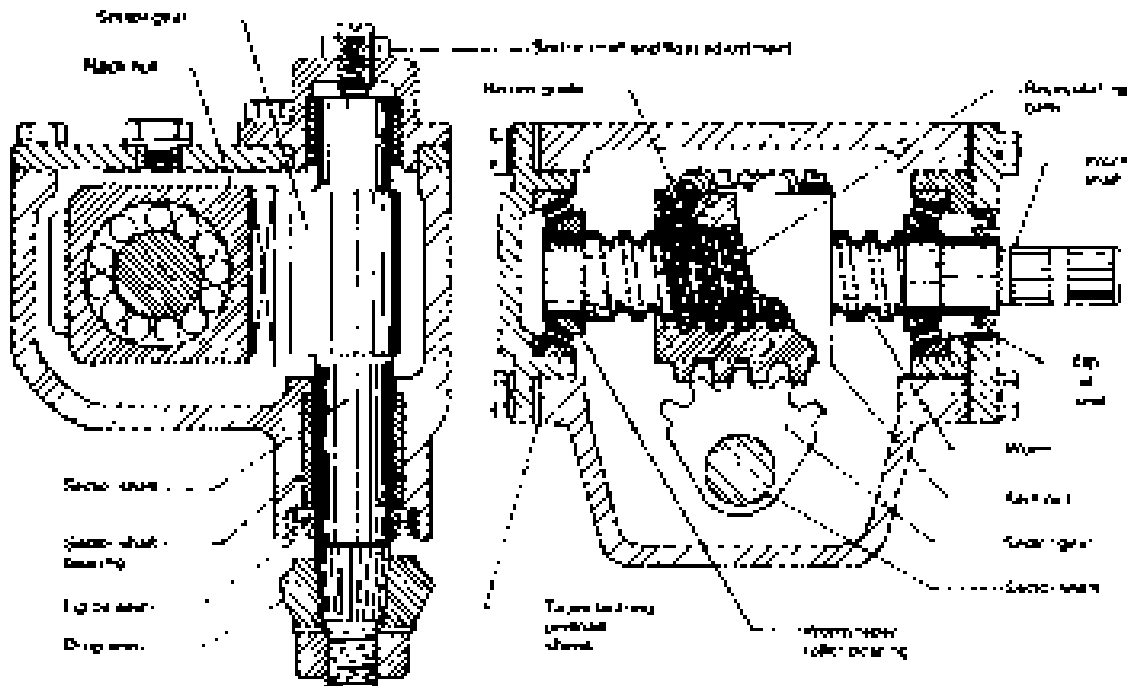
Fig. 9.9 Recirculating ball rack and sector steering gearbox

317

Engagement of the worm and nut is obtained by lodging a series of steel balls between the two sets of matching semicircular spiral grooves.

There are two separate ball circuits within the ball nut, and when the steering wheel and worm rotates, the balls roll in the grooves against the nut. This causes the nut to move along the worm. Each ball rotates one complete loop around the worm after which it enters a ball return guide. The guide deflects the balls away from the grooved passages so that they move diagonally across the back of the nut. They are then redirected again into the grooved passages on the other side of the nut.

One outer face of the rectangular nut is machined in the shape of teeth forming a gear rack. Motion from the nut is transferred to the drop arm via a toothed sector shaft which meshes with the rack teeth, so that the linear movement of the nut is converted back to a rotary motion by the sector and shaft.

An advantage of this type of steering gear is that the rack and sector provides the drop arm with a larger angular movement than most other types of mechanisms which may be an essential feature for some vehicle applications. Because of the additional rack and sector second stage gear reduction, the overall forward and reverse efficiencies are slightly lower than other recirculating ball mechanisms. Typical values for forward and reverse efficiencies would be 70% and 45% respectively.

9.2   he need for power assisted steering
(Figs 9.10 and 9.11)
With manual steering a reduction in input effort on the steering wheel rim is achieved by lowering the
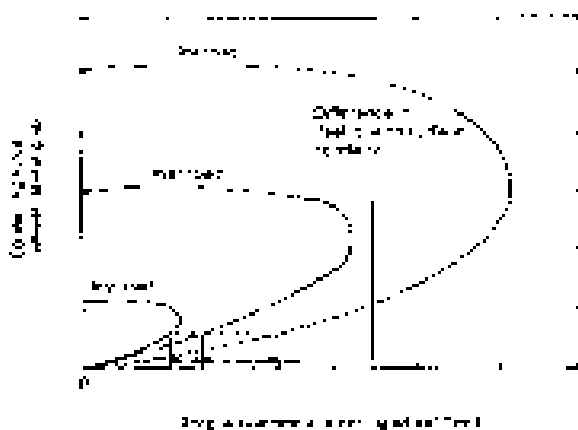
Fig. 9.10 Typical relationship of tyre grip on various road surfaces and the torque reaction on the driver's steering wheel
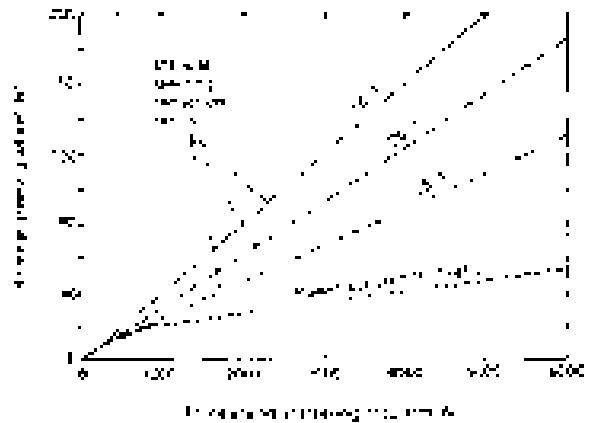
Fig. 9.11 Comparison of manual steering with different reduction gear ratio and power assisted steering

steering box gear ratio, but this has the side effect of increasing the number of steering wheel turns from lock so that manoeuvring of the steering will take longer, and accordingly the vehicle's safe cornering speed has to be reduced.

With the tendency for more weight to be put on the front steering wheels of front wheel drive cars and the utilization of radial ply tyres with greater tyre width, larger static turning torques are required. The driver's expectancy for faster driving and cornering makes power assisted steering desirable and in some cases essential if the driver's ability to handle the vehicle is to match its performance.

Power assistance when incorporated on passenger cars reduces the driver's input to something like 25–30% of the total work needed to manoeuvre it. With heavy trucks the hydraulic power (servo) assistance amounts to about 80–85% of the total steering effort. Consequently, a more direct steering box gear reduction can be used to provide a more precise steering response. The steering wheel movement from lock to lock will then be reduced approximately from 3   to 4 turns down to about 2   to 3 turns for manual and power assistance steering arrangements respectively.

The amount of power assistance supplied to the steering linkage to the effort put in by the driver is normally restricted so that the driver experiences the tyres' interaction with the ground under the varying driving conditions (Fig. 9.10). As a result there is sufficient resistance transmitted back to the driver's steering wheel from the road wheels to enable the driver to sense or feel the steering input requirements needed effectively to steer the vehicle.

The effects of reducing the driver's input effort at the steering wheel with different steering gear overall gear ratios to overcome an output opposing resistance at the steering box drop arm is shown in Fig. 9.11. Also plotted with these manual steering gear ratios is a typical power assisted steering input effort curve operating over a similar working load output range. This power assisted effort curve shows that for very low road wheel resistance roughly up to 1000 N at the drop arm, the input effort of 10 to 20 N is practically all manual. It is this initial manual effort at the steering wheel which gives the driver his sense of feel or awareness of changes in resistance to steering under different road surface conditions, such as whether the ground is slippery or not.

9.2.1 External direct coupled power assisted steering power cylinder and control valve

escription of power assisted steering system (Figs 9.12, 9.13 and 9.14) This directly coupled power assisted system is hydraulic in operation. The power assisted steering layout (Fig. 9.14) consists of a moving power cylinder. Inside this cylinder is a double acting piston which is attached to a ramrod anchored to the chassis by either rubber bushes or a ball joint. One end of the power cylinder is joined to a spool control valve which is supported by the steering box drop arm and the other end of the power cylinder slides over the stationary ramrod. When the system is used on a commercial vehicle with a rigid front axle beam (Fig. 9.12), the steering drag link is coupled to the power cylinder and control valve by a ball joint. If a car or van independent front suspension layout is used (Fig. 9.13), the power cylinder forms a middle
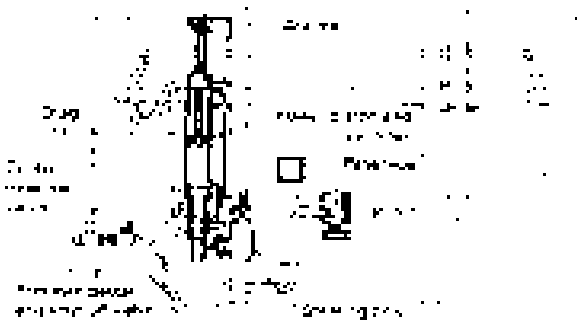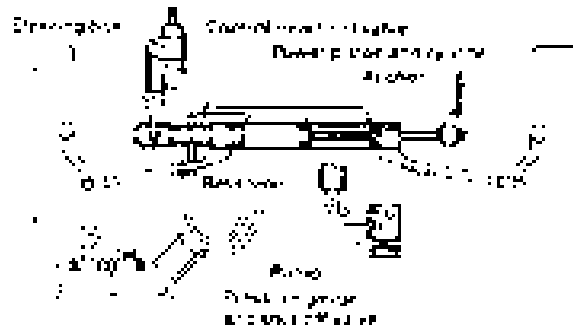
Fig. 9.13 Steering box with external directly coupled power assisted steering utilized with independent front suspension

moveable steering member with each end of the split track rods attached by ball joints at either end. The power source comes from a hydraulic pump mounted on the engine, and driven by it a pair of flexible hydraulic pipes connect the pump and a fluid reservoir to the spool control valve which is mounted at one end of the power cylinder housing. A conventional steering box is used in the system so that if the hydraulic power should fail the steering can be manually operated.

With the removal of any steering wheel effort a pre-compressed reaction spring built into the control valve (Fig. 9.14) holds the spool in the neutral position in addition to a hydraulic pressure which is directed onto reaction areas within the control valve unit. Provided the steering effort is less than that required to overcome the preload of the reaction spring, the spool remains central and the fluid is permitted to circulate from the pump through the valve and back to the reservoir. Under these conditions there will be no rise in hydraulic pressure and the steering will be manually operated. Consequently, the pump will be running light and therefore will consume very little power.

When the steering effort at the driver's wheel is greater than the preload stiffness of the reaction spring, the spool valve will move slightly to one side. This action partially traps fluid and prevents it returning to the reservoir so that it now pressurizes one side or the other of the double acting piston, thereby providing the power assistance necessary to move the steering linkage. The more the spool valve misaligns itself from the central position the greater the restriction will be for the fluid to return to the reservoir and the larger the pressure build up will be on one side or the other of the double acting piston to apply the extra steering thrust to turn the steering road wheels.

Fig. 9.12 Steering box with external directly coupled power assisted steering utilized with rigid axle front suspension

Operation of control valve and power piston (Fig. 9.14)

Neutral position (Fig. 9.14(a)) With the valve spool in the neutral position and no power assistance being used, fluid from the pump passes freely from the right hand supply port and annular groove in the valve housing, across the spool valve middle land to the return groove and port in the valve housing, finally returning to the reservoir. At the same time fluid passes from both the spool grooves to passages leading to the left and right hand power cylinder chambers which are sealed off from each other by the double acting piston. Thus whatever the position of the piston in the power cylinder when the spool is in the central or neutral position, there will be equal pressure on either side of the double acting piston. Therefore the piston will remain in the same relative position in the cylinder until steering corrections alter the position of the spool valve.

Right hand steering movement (Fig. 9.14(b)) If the drop arm pushes the ball pin to the right, the spool control edges 1 and 3 now overlap with the valve housing lands formed by the annular grooves. The fluid flows from the supply annular groove into the right hand spool groove where it then passes along passages to the right hand cylinder chamber where the pressure is built up to expand the chamber.

The tendency for the right hand cylinder chamber to expand forces fluid in the left hand contracting cylinder chamber to transfer through passages to the left hand spool groove. It then passes to the valve housing return annular groove and port back to the reservoir. Note that the ramrod itself remains stationary, whereas the power cylinder is the moving member which provides the steering correction.

eft hand steering movement (Fig. 9.14(c)) Movement of the drop arm to the left moves the spool with it so that control edges 2 and 4 now overlap with the adjacent valve housing lands formed by the annular grooves machined in the bore. Fluid flows from the supply annular groove in the valve housing to the axial passage in the spool and is then diverted radially to the valve body feed annular groove and the spool left hand groove. Fluid continues to flow along the passage leading to the left hand power cylinder chamber where it builds up pressure. As a result the left hand chamber

expands, the right hand chamber contracts, fluid is thus displaced from the reducing space back to the right hand spool groove, it then flows out to the valve housing return groove and port where finally it is returned to the reservoir.

rogressive power assistance (Fig. 9.14(a)) While the engine is running and therefore driving the hydraulic power pump, fluid enters the reaction chamber via the axial spool passage.

Before any spool movement can take place relative to the valve housing to activate the power assistance, an input effort of sufficient magnitude must be applied to the drop arm ball pin to compress the reaction spring and at the same time overcome the opposing hydraulic pressure built up in the reaction chamber. Both the reaction spring and the fluid pressure are utilized to introduce a measure of resistance at the steering wheel in proportion to the tyre to ground reaction resistance when the steered road wheels are turned and power assistance is used.

Progressive resistance at the steering wheel due to the hydraulic pressure in the reaction chamber can be explained in the following ways:

Right hand spool reaction (Fig. 9.14(b)) Consider the drop arm ball pin initially moved to the right. The reaction ring will also move over and slightly compress the reaction spring. At the same time the hydraulic pressure in the reaction chamber will oppose this movement. This is because the pressure acts between the area formed by the annular shoulder in the valve chamber housing taking the reaction spring thrust, and an equal projected area acting on the reaction ring at the opposite end of the chamber. The greater the hydraulic pressure the larger the input effort must be to turn the steering wheel so that the driver experiences a degree of feel at the steering wheel in proportion to the resisting forces generated between the tyre and road.

eft hand spool reaction (Fig. 9.14(c)) If the drop arm and ball pin is moved to the left, the reaction washer will move over in the same direction to compress the reaction spring. Opposing this movement is the hydraulic pressure which acts between the reaction washer shoulder area formed by the reduced diameter of the spool spindle and an equal projected area of the reaction ring situated at the opposite end. If the steering wheel effort is removed, the hydraulic pressure in the reaction

Fig. 9.14 External directly coupled power assisted steering

321

chamber will react between the reaction ring, the reaction washer housing and spool shoulders, and thereby attempt to move the spool back to its original central or neutral position.

Correction for the variation in cross section areas on opposite side of the power piston (Fig. 9.14(b and c))   To counteract the reduction in effective area on the ramrod side of the double acting piston, the annular shoulder area of the spool (Fig. 9.14(b)) is made slightly larger than the reaction ring annular shoulder area (Fig. 9.14(c)) machined in the reaction chamber. Consequently, a greater opposing hydraulic reaction will be created when turning the steering to the left to oppose the full cross-sectional area of the power piston compared to the situation when the steering is turned to the right and a reduced power piston cross-sectional area due to the ram is exposed to the hydraulic pressure in the power cylinder. In this way a balanced self-centralizing response is obtained in whatever position the road steering wheels may be positioned.

9.2.2 Rack and pinion power assisted steering gear power cylinder and control valve

Rack and pinion gear (Fig. 9.15) This power assisted steering system is comprised of a rack and pinion gear with double acting power (servo) piston mounted on the rack and a rotary valve coaxial with the extended pinion shaft (Fig. 9.15).

Helical teeth are cut on the 3% nickel, 1% chromium case hardened steel pinion shaft, they mesh with straight cut teeth on a 0.4% carbon manganese silicon steel rack which is induction hardened. To accommodate these gears, the axis of the pinion is inclined at 76° to that of the rack.
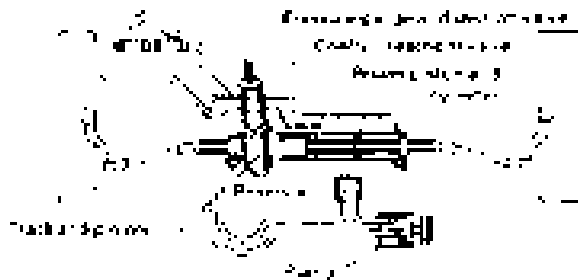
Fig. 9.15 Integral rack and pinion power assisted steering utilized with independent front suspension

escription of rotary control valve (Fig. 9.16(a, b and c))   The three major components of the rotary type control valve are the rotor shaft, the torsion bar and the valve sleeve (Fig. 9.16(a)).

Slots are milled longitudinally on the periphery of the rotor shaft (Fig. 9.16(b)) and on the inner surface of the valve sleeve of which the rotor is assembled and in which it is free to rotate. The sleeve is rotated by the pinion gear shaft. Limited rotation of the rotor shaft, relative to the pinion gear, occurs when the torsion bar that connects the rotor shaft to the pinion shaft is angularly deflected. Hence when steering effort is applied, the torsion bar twists and the slots on the rotor move relative to those in the sleeve to allow fluid to pass to one side of the double acting piston which operates inside the power cylinder. The direction of rotation of the rotor relative to the sleeve determines which side of the double acting piston the fluid will act.

The rotor shaft of the valve forms part of the 0.5% carbon steel rotor shaft and is connected by a torsion bar to the pinion shaft. The outer case-hardened 0.15% carbon steel sleeve which is coaxial with the rotor floats on the ground surface of the rotor shaft, there being a diametrical clearance of 0.004 to 0.012 mm between the rotor and sleeve. The sleeve is connected by a steel trim pin screw to the pinion shaft. This pin, the threaded end of which is in a tapped radial hole in the pinion shaft, has a spherical head that fits with no clearance in a radial hole drilled in the sleeve at the pinion end. The axis of the head of this pin is eccentric to that of the threaded shank. Hence, when the pinion shaft and rotor shaft have been locked in the central position of rotation, the correct angular position of the sleeve, relative to that of the rotor, can be set by rotating the pin. This permits the valve assembly to be trimmed, so that the division of fluid flow between rotor and sleeve slot edges is balanced. The amount of opening between the rotor and sleeve control slots is equal to the angular deflection of the torsion bar. Four square section Teflon seals are assembled into annular grooves in the periphery of the sleeve. Between these seals are three wider annular grooves, again on the periphery of the sleeve, from which fluid enters or leaves the valve assembly.

Rotor shaft to pinion shaft coupling (Fig. 9.16(a)) Splines at one end of the rotor shaft register in an internally splined recess in the pinion shaft. The width of the splines is such that the torsion bar can twist a total of seven degrees before the splines

contact one another. Manual steering effort is transferred from the rotor shaft through the 1% chromium-molybdenum-steel torsion bar, which has an approximate waist diameter of 5 mm, to the pinion shaft. The splined coupling between these two members ensures that, in the event of failure of the power assistance, the steering gear can be operated manually without overstressing the torsion bar.

Rotor and sleeve longitudinal and annular grooves (Fig. 9.16(b)) Six equally spaced longitudinal slots are milled on the circumference of the rotor. Three of the rotor slots are longer than the other three and these two lengths are disposed alternately around the rotor periphery. There are also six matching equally spaced groove slots with closed-off ends in the bore of the valve sleeve. The angular relationship of these two sets of slots controls the flow of fluid from the pump to the power cylinder and from the cylinder to the reservoir.

The positioning of the ports and annular grooves around the valve sleeve are now considered from the pinion end of the rotor shaft.

The first valve port feeds or returns fluid from the left hand side of the power cylinder, the second port delivers fluid from the pump, the third port feeds or returns fluid from the right hand side of the power cylinder and the fourth port acts as the return passage to the reservoir.

Each of the first three ports (counting from the left) communicates with corresponding annular grooves in the outer periphery of the sleeve, while the fourth port, through which fluid is returned to the reservoir, communicates with an annular space between the right hand end of the valve housing and the end of the sleeve.

Radial holes in the central sleeve groove connect the pump pressurized fluid to the short supply slots on the periphery of the rotor shaft. The sleeve grooves on either side are connected through small radial holes to the three longer slots on the rotor surface. These longer slots provide a return passage to the annular space on the right of the valve which then leads back to the reservoir.

Whenever the pump is operating, fluid passes through the delivery port in the sleeve and is then transferred to the three short rotor slots. Thereafter the fluid either passes directly to the longer slots, and hence to the return port, or flows through one feed and return port going to the power cylinder, while fluid from the other end of the cylinder is returned through the longer slots to the reservoir.

Torsion bar stiffness (Fig. 9.16(a)) When torque is applied at the steering wheel, the grip of the tyres on the road causes the torsion bar to twist. Relative movement between the rotor shaft and sleeve upsets the balanced flow of fluid and pressure at the rotary control valve going to both sides of the power piston. Therefore the deflection rate of the torsion bar is important in determining the torque required to steer the vehicle. To transmit the desired input torque from the steering wheel and rotor shaft to the pinion shaft, a standard 108 mm long torsion bar is used. Its diameter and stiffness are 5.2 mm and 1.273 N/deg and the actual diameter of the torsion bar can be changed to suit the handling requirements of the vehicle.

Rotor slot edge control (Fig. 9.16(b)) If the valve rotor and sleeve slot control edges were sharp as shown in the end view (Fig. 9.16(b)), the area through which the fluid flowed would vary directly with the deflection of the torsion bar, and hence the applied torque. Unfortunately, the relationship between pressure build-up and effective valve flow area is one in which the pressure varies inversely as the square of the valve opening area. This does not provide the driver with a sensitive feel at the steering wheel rim. The matching of the pressure build-up, and hence valve opening area, relative to the driver's input effort at the steering wheel rim has been modified from the simple sharp edge slots on the rotor, to contoured stepped edges (Fig. 9.16(b)) which provide a logarithmic change of area so that the increase of pressure with rim effort is linear. This linear relationship is retained up to 28 bar which is the limit for driving a large car under all road conditions. Above 28 bar the rise of pressure with applied steering wheel torque is considerably steepened to ensure that parking efforts are kept to a minimum. The relationship between the angular displacement of the rotary valve and the hydraulic pressure applied to one side of the power cylinder piston is shown in Fig. 9.17.

An effort of less than 2.5 N at the rim of the steering wheel is sufficient to initiate a hydraulic pressure differential across the double acting power piston. During manoeuvres at very low speeds when parking, a manual effort of about 16 N is required. When the car is stationary on a dry road, an effort of 22 N is sufficient to move the steering from the straight ahead position to full lock position.

Operation of the rotary control valve and power piston

Neutral position (Fig. 9.16(a)) With no steering effort being applied when driving along a straight

Fig. 9.16 (a–d) Rack and pinion power assisted steering with rotary control valve
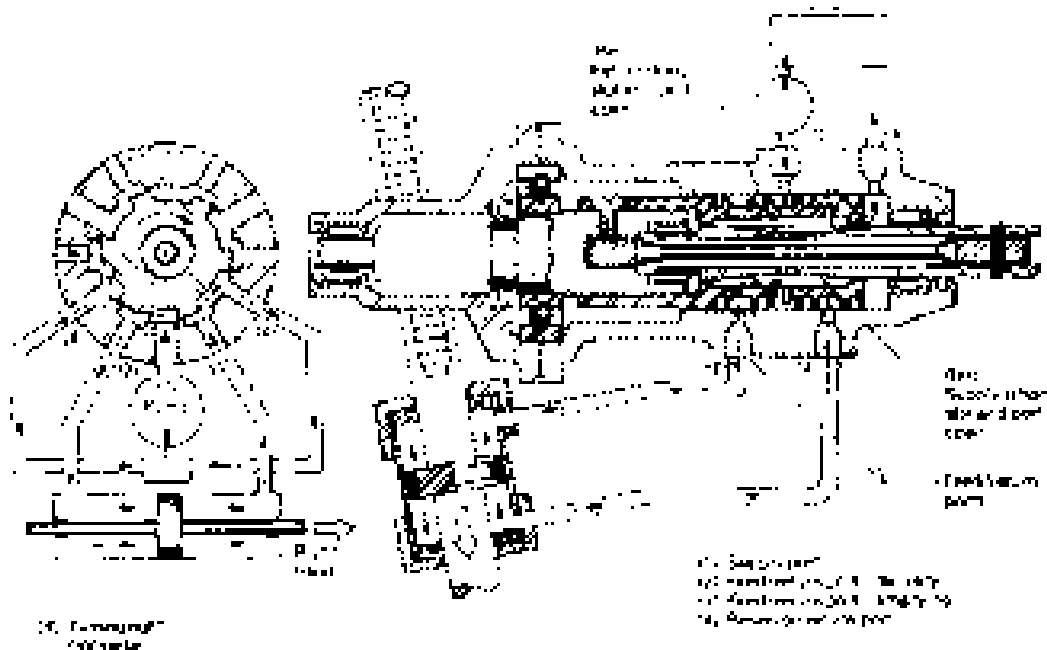
Fig. 9.16 *contd*

track, the longitudinal lands formed by the slots milled on the rotor periphery angularly align with the internal sleeve slots so that equal space exists between the edges of adjacent rotor and sleeve slots.

Fluid therefore flows from the pump to the delivery port into the short slots in the rotor. Some fluid then passes to the feed/return slots in the sleeve and out to the ports communicating with either side of the power cylinder. The majority of the fluid will pass between the edges of both the rotor and sleeve slots to the rotor long slots where it then flows out through the return port back to the reservoir. Because fluid is permitted to circulate from the pump to the reservoir via the control valve, there is no pressure build-up across the power piston, hence the steering remains in the neutral position.

Anticlockwise rotation of the steering wheel (Fig. 9.16(b)) Rotating the steering wheel anticlockwise twists the rotor shaft and torsion bar so that the leading edges of the rotor lands align with corresponding lands on the sleeve, thereby blocking off the original fluid exit passage. Fluid now flows from the pump delivery port to the short slots on the rotor. It then passes between the trailing rotor and sleeve edges, through to the sleeve slots and from there to the left hand side of the power
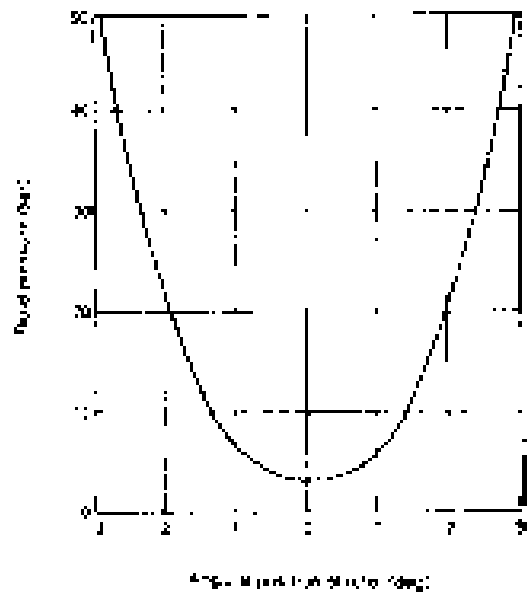
Fig. 9.17 Relationship of fluid pressure delivered to the power cylinder from the control valve and the angular deflection of the control valve and torsion bar

cylinder via the feed/return pipe so that the pressure on this side of the piston rises. At the same time fluid will be displaced from the right hand cylinder end passing through to the sleeve slots

325

via the right hand feed/return pipe and port. The flow of fluid continues, passing between the trailing land edges of the rotor and sleeve to the long rotor slots and out through the return pipe back to the reservoir. Thus a pressure difference is established across the double acting piston, to provide power assistance.

Clockwise rotation of the steering wheel (Fig. 9.16(c)) Rotating the steering wheel clockwise angularly deflects the rotor so that the leading edges of the rotor lands overlap with corresponding internal lands on the sleeve. Fluid now flows from the pump delivery port into the short rotor slots and out to the right hand feed/return port to the power cylinder via the gap created between the trailing edges of the rotor and sleeve lands. Pressurizing the right hand side of the power cylinder pushes fluid out from the left hand side of the cylinder, through the feed/return pipe and port into the sleeve slots, through the enlarged gap created between the trailing rotor and sleeve edges and into the long rotor slots. It is then discharged through the return port and pipe back to the reservoir.

rogressive power assistance (Fig. 9.16(b and c)) When the steering wheel is turned left or right, that is, anticlockwise or clockwise, the rotor shaft which is rigidly attached to the steering column shaft rotates a similar amount. A rotary movement is also imparted through the torsion bar to the pinion shaft and the valve sleeve as these members are locked together. However, due to the tyre to ground resistance, the torsion bar will twist slightly so that the rotation of the pinion and sleeve will be less than that of the rotor input shaft. The greater the road wheel resistance opposing the turning of the front wheel, the more the torsion bar will twist, and therefore the greater the misalignment of the rotor and sleeve slots will be. As a result, the gap between the leading edges of both sets of slots will become larger, with a corresponding increase in fluid pressure entering the active side of the power cylinder.

As the steering manoeuvres are completed, the initially smaller sleeve angular movement catches up with the rotor movement because either the road wheel resistance has been overcome or steering wheel turning effort has been reduced. Consequently, the reduced torque now acting on the steering column shaft enables the torsion bar to unwind (i.e. straighten out). This causes the power assistance to be reduced in accordance with the realignment or centralization of the rotor slots relative to the sleeve lands.

9.2.3 *ntegral power assisted steering gear power cylinder and control valve*

escription of steering gear and hydraulic control valve (Figs 9.18 and 9.19) The integral power assisted steering gearbox can be used for both rigid front axle and independent front suspension (Fig. 9.18) layouts.

The rack and sector recirculating ball steering gear, power cylinder and hydraulic control valves are combined and share a common housing (Fig. 9.19(a)). The power piston in this arrangement not only transforms hydraulic pressure into force to assist the manual input effort but it has two other functions:

1 it has a rack machined on one side which meshes with the sector,
2 it has a threaded axial bore which meshes via a series of recirculating balls with the input worm shaft.

The input end of the worm shaft, known as the worm head, houses two shuttle valve pistons which have their axes at right angles to the worm shaft. Since they are assembled within the worm head they rotate with it.

Drive is transferred from the hollow input shaft to the worm shaft through a torsion bar. Movement of the shuttle valves relative to the worm shaft which houses the valves is obtained by the hollow double pronged input shaft. Each prong engages with a transverse hole situated mid-way between the shuttle valve ends.
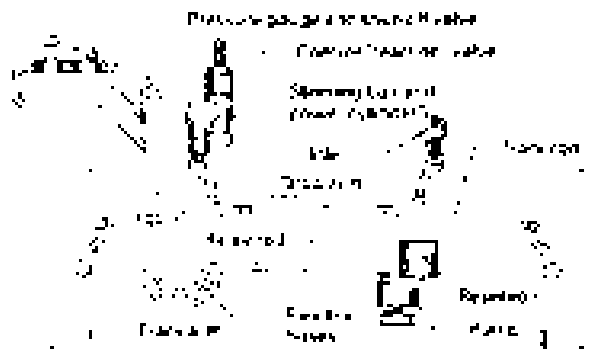
Fig. 9.18 Integral steering gearbox and power assisted steering utilized with independent front suspension
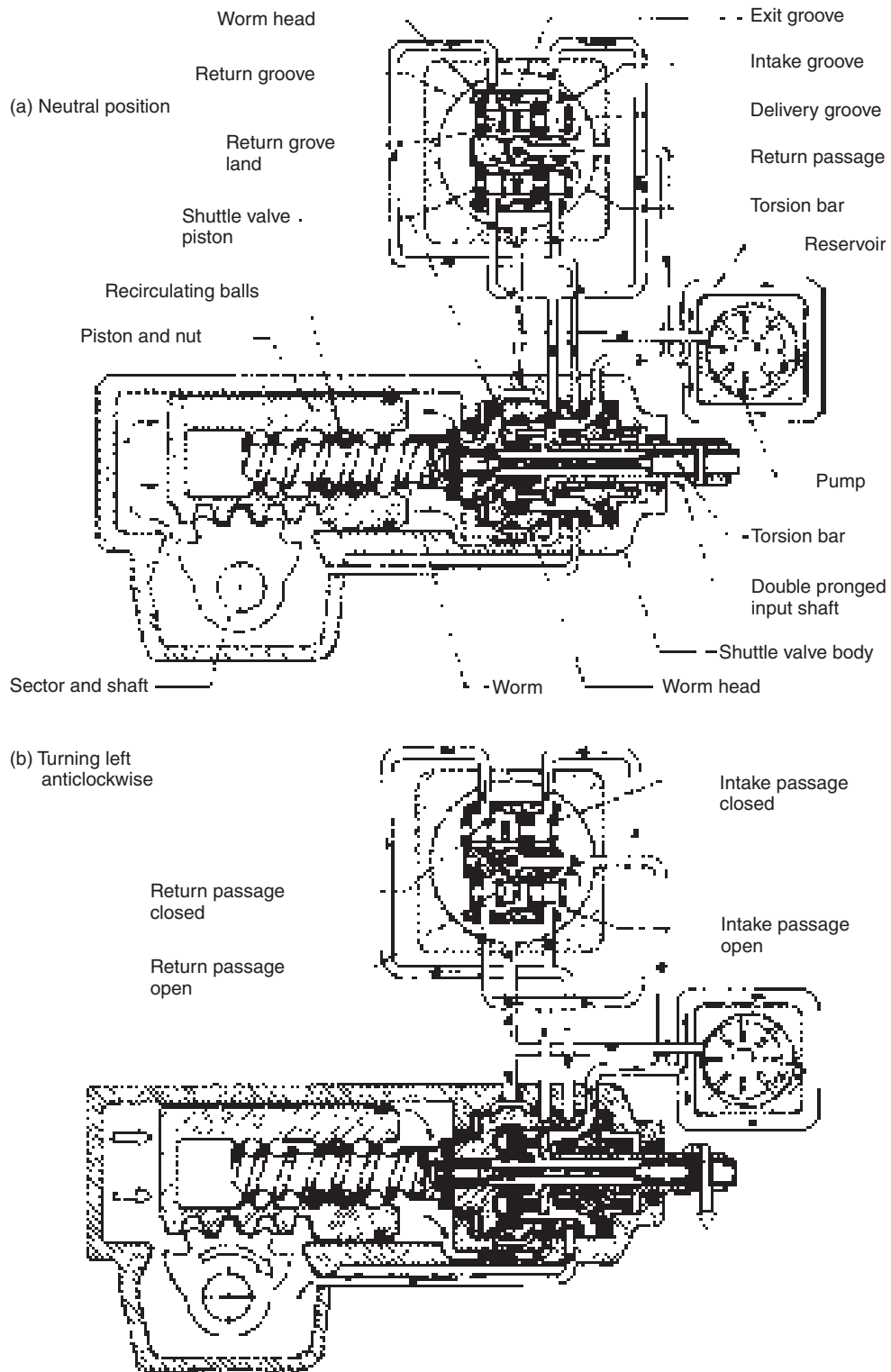
Fig. 9.19 (a–c) Integral power assisted steering gear power cylinder and control valve

327

(c) Turning right clockwise

Intake passage open

Return passage open

Return passage closed

Intake passage closed

Double pronged input shaft

Torsion bar

(One of two)

Shuttle valve piston
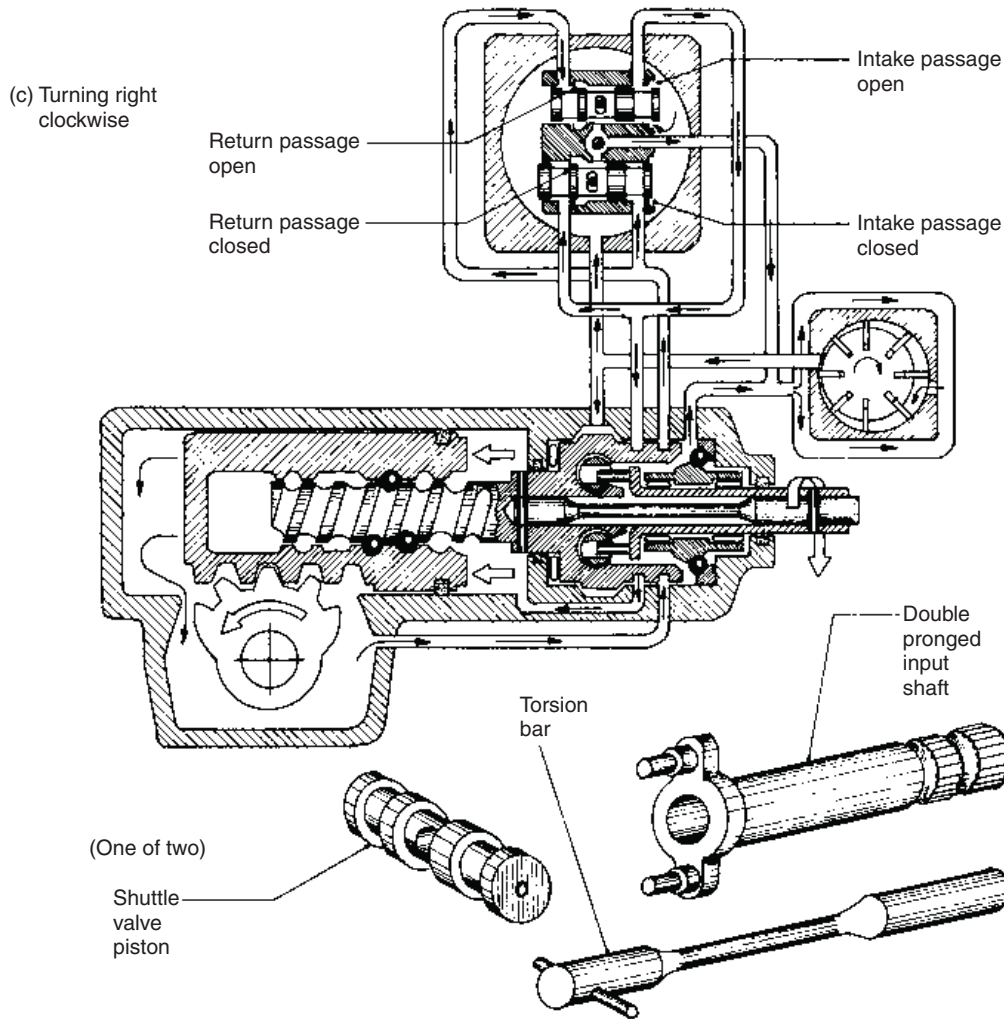
Fig. 9.19 *contd*

When the steering wheel is turned, the tyre to ground reaction on the front road wheels causes the torsion bar to twist according to the torque applied on the steering column shaft. Therefore the relative angular movement of the worm shaft to that of the input shaft increases in proportion to the input torque at the steering wheel, so that the shuttle valves will both be displaced an equal amount from the mid-neutral position. As soon as the steering wheel effort is released, the elastic torsion bar ensures that the two shuttle valves return to the neutral or mid position. The function of these shuttle valves is to transfer fluid under pressure, in accordance to the steering input torque, from the pump delivery port to one or other end of the integral power cylinder whilst fluid from the opposite end of the cylinder is released and returned to the reservoir.

Operation of control valve and power piston

Neutral position (Fig. 9.19(a)) Fluid from the pump flows into and around an annular chamber surrounding the worm head in a plane similar to that of the shuttle valves where it acts on the exposed end faces of the shuttle valve pistons.

With the shuttle valves in the neutral position, fluid moves through the intake passages on the right hand end of the shuttle valve pistons, to the two annular grooves on the periphery of the worm head. Fluid then passes from the worm head annular grooves to the left hand side of the power piston

328

via the horizontal long passage and sector chamber, and to the right hand piston face directly by way of the short passage. From the worm head grooves fluid will also flow into the shuttle valve return grooves, over each return groove land which is aligned with the exit groove, to the middle waisted region of the shuttle valve and into the torsion bar and input shaft chamber. Finally fluid moves out from the return pipe back to the pump reservoir.

Turning left (anticlockwise rotation) (Fig. 9.19(b))
An anticlockwise rotation of the steering wheel against the front wheel to ground opposing resistance distorts the torsion bar as input torque is transferred to the worm shaft via the torsion bar. The twisting of the torsion bar means that the worm shaft also rotates anticlockwise, but its angular movement will be less than the input shaft displacement. As a result, the prongs of the input shaft shift the upper and lower shuttle valves to the left and right respectively. Accordingly this movement closes both the intake and return passages of the upper shuttle valve and at the same time opens both the intake and return passages of the lower shuttle valve.

Fluid can now flow from the pump into the worm head annular space made in the outer housing. It then passes from the lower shuttle valve intake to the right hand worm head annular groove. The transfer of fluid is complete when it enters the left hand power cylinder via the sector shaft. The amount of power assistance is a function of the pressure build-up against the left side of the piston, which corresponds to the extent of the shuttle valve intake passage opening caused by the relative angular movements of both the input shaft and worm shaft.

Movement of the power piston to the right displaces fluid from the right hand side of the power cylinder, where it flows via the worm head annular groove to the lower shuttle valve return passage to the central torsion bar and input shaft chamber. It then flows back to the reservoir via the flexible return pipe.

Turning right (clockwise rotation) (Fig. 9.19(c))
Rotating the steering wheel in a clockwise direction applies a torque via the torsion bar to the worm in proportion to the tyre to ground reaction and the input effort. Due to the applied torque, the torsion bar twists so that the angular movement of the worm shaft lags behind the input shaft displacement. Therefore the pronged input will rotate clockwise to the worm head.

With a clockwise movement of the input shaft relative to the worm head, the upper shuttle valve piston moves to the right and the lower shuttle valve piston moves to the left. Consequently, the upper shuttle valve opens both the intake and return passages but the lower shuttle valve closes both the intake and return passages.

Under these conditions fluid flows from the pump to the annular space around the worm head in the plane of the shuttle valves. It then enters the upper valve intake, fills the annular valve space and passes around the left hand worm head groove. Finally, fluid flows through the short horizontal passage into the right hand side of the power cylinder where, in proportion to the pressure build-up, it forces the piston to the left. Accordingly the meshing rack and sector teeth compel the sector shaft to rotate anticlockwise.

At the same time as the fluid expands the right hand side of the power cylinder, the left hand side of the power cylinder will contract so that fluid will be displaced through the long horizontal passage to the worm head right hand annular groove. Fluid then flows back to the reservoir via the upper shuttle valve return groove and land, through to the torsion bar and input shaft chamber and finally back to the reservoir.

9.2.4 Power assisted steering lock limiters
(Fig. 9.20(a and b))
Steering lock limiters are provided on power assisted steering employed on heavy duty vehicles to prevent excessive strain being imposed on the steering linkage, the front axle beam and stub axles and the supporting springs when steering full lock is approached. It also protects the hydraulic components such as the pump and the power cylinder assembly from very high peak pressures which could cause damage to piston and valve seals.

Power assisted steering long stem conical valve lock limiter The lock limiters consist of a pair of conical valves with extended probe stems located in the sector shaft end cover (Fig. 9.20(a and b)). Each valve is made to operate when the angular movement of the sector shaft approaches either steering lock, at which point a cam profile machined on the end of the sector shaft pushes open one or other of the limiting valves. Opening one of the limiter valves releases the hydraulic pressure in the power cylinder end which is supplying the assistance; the

Reservoir
ands
pump

Right hand
valve closed

Pressure
relief
valve

Left hand
valve open

Flow
control
valve

Piston and nut

Input
hollow
shaft

Torsion
bar

(a) Turning left
anticlockwise

Conical
valve
and
profile
stem

Sector
gear

Cam
profile

Sector
shaft

Left hand
closed

Right hand
valve open

(b) Turning right
clockwise
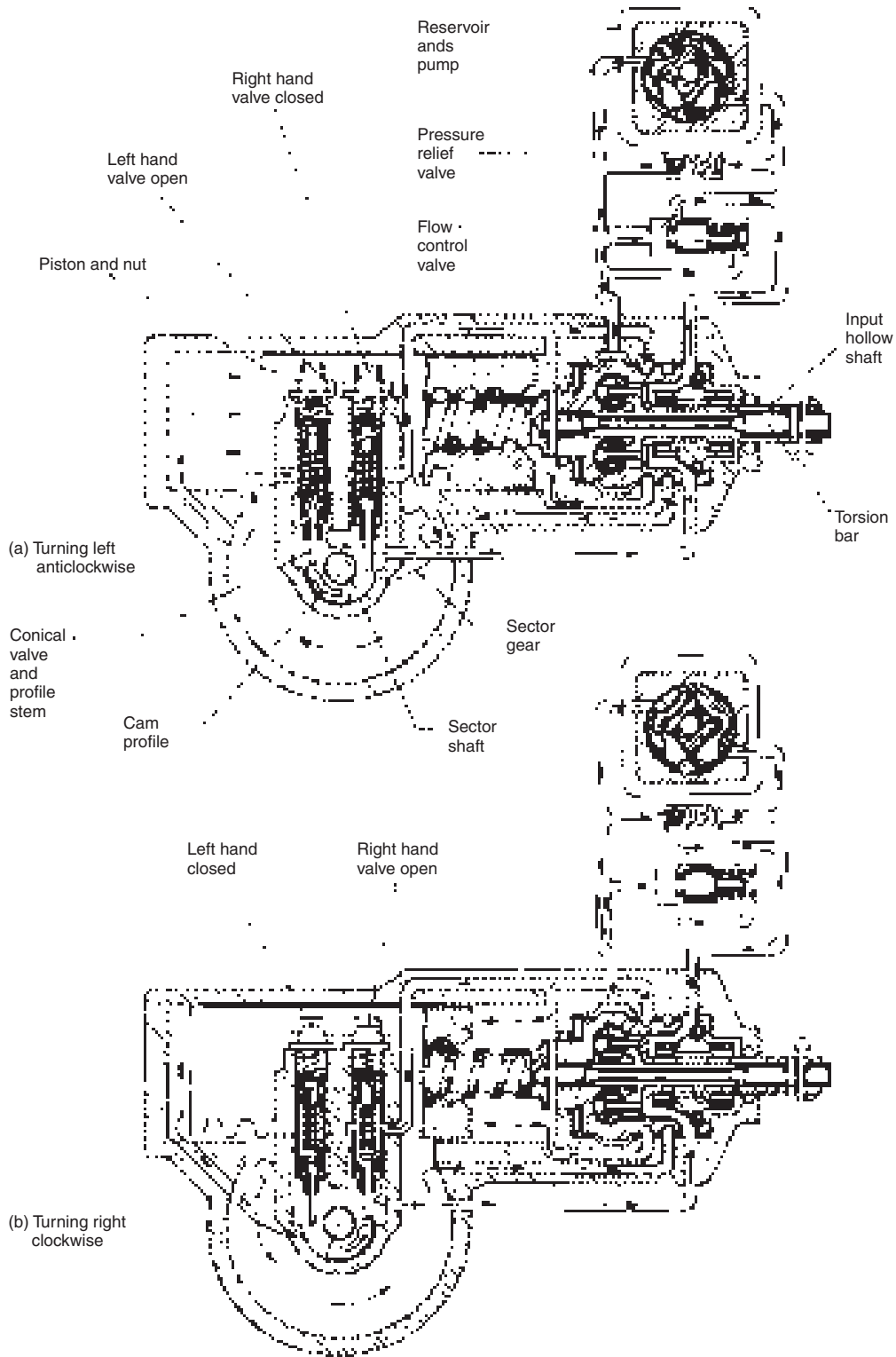
Fig. 9.20 (a and b) Power assisted steering long stem conical valve lock limiter

excess fluid is then permitted to flow back to the reservoir via the control housing.

Turning left (anticlockwise steering rotation) (Fig. 9.20(a))   Rotation of the input shaft anti-clockwise applies both manual and hydraulic effort onto the combined power piston and nut of the steering box so that it moves to the right within the cylinder. ust before the steering reaches full lock, one of the sector cam faces contacts the corresponding valve stem and pushes the conical valve off its seat. Pressurized fluid will immediately escape past the open valve through to the return chamber in the control valve housing, where it flows back to the reservoir. Therefore, any further rotation of the sector shaft will be entirely achieved by a considerable rise in manual effort at the steering wheel, this being a warning to the driver that the steering has reached maximum lock.

Turning right (clockwise steering rotation) (Fig. 9.20(b))   Rotation of the steering box input shaft clockwise screws the worm out from the piston and nut and actuates the control valve so that hydraulic pressure builds up on the right hand end of the piston. As the sector shaft rotation approaches maximum lock, the sector cam meets the valve stem, presses open the valve against the valve return spring tension and causes the hydraulic pressure in the right hand cylinder chamber to drop. The excess fluid will now flow back to the reservoir via the right hand end annular chamber in the control valve housing. The driver will immediately experience a considerable increase in manual effort at the steering wheel, indicating that the road wheels have been rotated to near enough maximum lock.

Power assisted steering double ball valve lock limiter   This lock limiter consists of a simple double ball valve located in the blank end of the integral piston and nut. To control the stroke of the piston an adjustable stop pin is mounted in the enclosed end of the power cylinder housing, while the right hand piston movement is limited by the stop pin mounted in the end of the worm shaft.

Turning left (anticlockwise steering rotation) (Fig. 9.21(a))   Rotation of the steering input shaft anticlockwise causes both manual and hydraulic effort to act on the combined power piston and nut, moving it towards the right. As

the steering lock movement is increased, the piston approaches the end of its stroke until the right hand ball valve contacts the worm shaft stop pin, thereby forcing the ball off its seat. The hydraulic pressure existing on the left side of the piston, which has already opened the left hand side ball valve, is immediately permitted to escape through the clearance formed between the internal bore of the nut and the worm shaft. Fluid will now flow along the return passage leading to the control reaction valve and from there it will be returned to the reservoir. The release of the fluid pressure on the right side of the piston therefore prevents any further hydraulic power assistance and any further steering wheel rotation will be entirely manual.

Turning right (clockwise steering rotation) (Fig. 9.21(b))   Rotation of the steering box input shaft clockwise screws the worm out from the piston and nut. This shifts the shuttle valve pistons so that the hydraulic pressure rises on the right hand end of the piston. Towards the end of the left hand stroke of the piston, the ball valve facing the blind end of the cylinder contacts the adjustable stop pin. Hydraulic pressure will now force the fluid from the high pressure end chamber to pass between the worm and the bore of the nut to open the right hand ball valve and to escape through the left hand ball valve into the sector gear chamber. The fluid then continues to flow along the return passage going to the control reaction valve and from there it is returned to the reservoir. The circulation of fluid from the pump through the piston and back to the reservoir prevents further pressure build-up so that the steering gearbox will only operate in the manual mode. Hence the driver is made aware that the road wheels have been turned to their safe full lock limit.

9.2.5 *Roller type hydraulic pump*
(Fig. 9.22(a and b))
The components of this pump (Fig. 9.22 (a and b)) consist of the stationary casing, cam ring and the flow and pressure control valve. The moving parts comprise of a rotor carrier mounted on the drive shaft and six rollers which lodge between taper slots machined around the rotor blank. The drive shaft itself is supported in two lead-bronze bushes, one of which is held in the body and the other in the end cover. A ball bearing at the drive end of the shaft takes the load if it is belt and pulley driven.

The rotor carrier is made from silicon manganese steel which is heat treated to a moderate hardness.

The rotor slots which guide the rollers taper in width towards their base, but their axes instead of being radial have an appreciable trailing angle so as to provide better control over the radial movement of the rollers. The hollow rollers made of case-hardened steel are roughly 10 mm in diameter and there are three standard roller lengths of 13, 18 and 23 mm to accommodate three different capacity pumps.

The cam ring is subjected to a combined rolling and sliding action of the rollers under the generated pressure. To minimize wear it is made from heat treated nickel-chromium cast iron. The internal profile of the cam ring is not truly cylindrical, but is made up from a number of arcs which are shaped to maximize the induction of delivery of the fluid as it circulates through the pump.

To improve the fluid intake and discharge flow there are two elongated intake ports and two similar discharge ports at different radii from the shaft axes. The inner ports fill or discharge the space between the rollers and the bottoms of their slots and the outer ports feed or deliver fluid in the space

formed between the internal cam ring face and the lobes of the rotor carrier. The inner elongated intake port has a narrow parallel trailing (transition) groove at one end and a tapered leading (timing) groove at the other end. The inner discharge port has only a tapered trailing (timing) groove at one end. These secondary circumferential groove extensions to the main inner ports provide a progressive fluid intake and discharge action as they are either sealed or exposed by the rotor carrier lobes and thereby reduce shock and noise which would result if these ports were suddenly opened or closed, particularly if air has become trapped in the rotor carrier slots.

Operating cycle of roller pump (Fig. 9.22(a and b)) Rotation of the drive shaft immediately causes the centrifugal force acting on the rollers to move them outwards into contact with the internal face of the cam ring. The functioning of the pump can be considered by the various phases of operation as

Fig. 9.21 (a and b)   Power assisted steering double ball valve lock limit

332

Fig. 9.22 (a and b) Power assisted roller type pump and control valve unit

an individual roller moves around the internal cam face through positions A, B, C, D, E and F.

Filling phase (Fig. 9.22(a))   As the roller in position A moves to position B and then to position C, the space between the eccentric mounted rotor carrier lobe and cam face increases. Therefore the volume created between adjacent rollers will also become greater. The maximum chamber volume occurs between positions C and D. As a result,

the pressure in these chambers will drop and thus induce fluid from the intake passages to enter by way of the outer chamber formed by the rotor lobe and the cam face and by the inner port into the tapered roller slot region. Filling the two regions of the chamber separately considerably speeds up the fluid intake process.

ressurization phase (Fig. 9.22(a))   With further rotation of the rotor carrier, the leading edge of the

333

rotor slot just beyond position C is just on the point of closing the intake ports, and the space formed between adjacent rollers at positions C and D starts to decrease. The squeezing action pressurizes the fluid.

ischarge phase (Fig. 9.22(a)) ust beyond roller position D the inner discharge port is uncovered by the trailing edge of the rotor carrier slot. This immediately enables fluid to be pushed out through the inner discharge port. As the rotor continues to rotate, the roller moves from position D to E with a further decrease in radial chamber space so that there is a further rise in fluid pressure. Eventually the roller moves from position E to F. This uncovers the outer discharge port so that an increased amount of fluid is discharged into the outlet passage.

Transition phase (Fig. 9.22(a)) The roller will have completed one revolution as it moves from position F to the starting position at A. During the early part of this movement the leading edge of the rotor slot position F closes both of the discharge ports and at about the same time the trailing edge of the rotor slot position A uncovers the transition groove in readiness for the next filling phase. The radial space between the rotor lobe and internal cam face in this phase will be at a minimum.

low and pressure control valves

escription of the flow and pressure control valve unit (Fig. 9.22(a and b)) The quantity of fluid discharged from the roller type pump and the build-up in fluid pressure both increase almost directly with rising pump rotor speed. These characteristics do not meet the power assisted steering requirements when manoeuvring at low speed since under these conditions the fluid circulation is restricted and a rise in fluid pressure is demanded to operate the power cylinders double acting piston. At high engine and vehicle speed when driving straight ahead, very little power assistance is needed and it would be wasteful for the pump to generate high fluid pressures and to circulate large amounts of fluid throughout the hydraulic system. To overcome the power assisted steering mismatch of fluid flow rate and pressure build-up, a combined flow control and pressure relief valve unit is incorporated within the cast iron pump housing. The flow control valve consists of a spring loaded plunger type valve and within the plunger body is a ball and spring pressure relief valve. Both ends of

the plunger valve are supplied with pressurized fluid from the pump. Situated in the passage which joins the two end chambers of the plunger is a calibrated flow orifice. The end chamber which houses the plunger return spring is downstream of the flow orifice.

Fluid from the pump discharge ports moves along a passage leading into the reduced diameter portion of the flow control plunger (Fig. 9.22(a)). This fluid circulates the annular space surrounding the lower part of the plunger and then passes along a right angled passage through a calibrated flow orifice. Here some of the fluid is diverted to the flow control plunger spring chamber, but the majority of the fluid continues to flow to the outlet port of the pump unit, where it then goes through a flexible pipe to the control valve built into the steering box (pinion) assembly. When the engine is running, fluid will be pumped from the discharge ports to the flow control valve through the calibrated flow orifice to the steering box control valve. It is returned to the reservoir and then finally passed on again to the pump's intake ports.

rinciple of the flow orifice (Fig. 9.22(a and b)) With low engine speed (Fig. 9.22(a)), the calibrated orifice does not cause any restriction or apparent resistance to the flow of fluid. Therefore the fluid pressure on both sides of the orifice will be similar, that is $_1$.

As the pump speed is raised (Fig. 9.22(b)), the quantity of fluid discharged from the pump in a given time also rises, this being sensed by the flow orifice which cannot now cope with the increased amount of fluid passing through. Thus the orifice becomes a restriction to fluid flow, with the result that a slight rise in pressure occurs on the intake side of the orifice and a corresponding reduction in pressure takes place on the outlet side. The net outcome will be a pressure drop of $_1 - {}_2$, which will now exist across the orifice. This pressure differential will become greater as the rate of fluid circulation increases and is therefore a measure of the quantity of fluid moving through the system in unit time.

Operation of the flow control valve (Fig. 9.22 (a and b)) When the pump is running slowly the pressure drop across the flow orifice is very small so that the plunger control spring stiffness is sufficient to fully push the plunger down onto the valve cap stop (Fig. 9.22(a)). However, with rising pump

speed the flow rate (velocity) of the fluid increases and so does the pressure difference between both sides of the orifice. The lower pressure p_2 on the output side of the orifice will be applied against the plunger crown in the control spring chamber, whereas the higher fluid pressure p_1 will act underneath the plunger against the annular shoulder area and on the blanked off stem area of the plunger. Eventually, as the flow rate rises and the pressure difference becomes more pronounced, the hydraulic pressure acting on the lower part of the plunger p_1 will produce an upthrust which equals the downthrust of the control spring and the fluid pressure p_2. Consequently any further increase in both fluid velocity and pressure difference will cause the flow control plunger to move back progressively against the control spring until the shouldered edge of the plunger uncovers the bypass port (Fig. 9.22(b)). Fluid will now easily return to the intake side of the pump instead of having to work its tortuous way around the complete hydraulic system. Thus the greater the potential output of the pump due to its speed of operation the further back the plunger will move and more fluid will be bypassed and returned to the intake side of the pump. This means in effect that the flow output of the pump will be controlled and limited irrespective of the pump speed (Fig. 9.23). The maximum output characteristics of the pump are therefore controlled by two factors; the control spring stiffness and the flow orifice size.

Operation of the pressure relief valve (Fig. 9.22 (a and b)) The pressure relief valve is a small ball and spring valve housed at one end and inside the plunger type flow control valve at the control spring chamber end (Fig. 9.22(a)). An annular groove
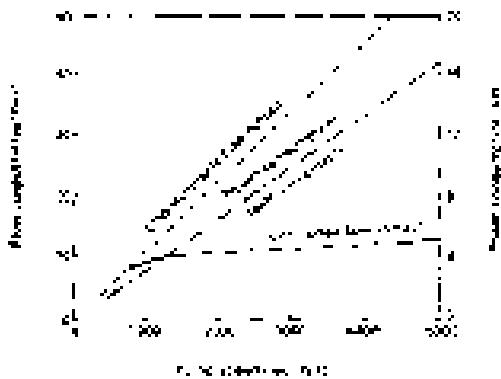
Fig. 9.23 Typical roller pump flow output and power consumption characteristics

is machined on the large diameter portion of the plunger just above the shoulder. A radial relief hole connects this groove to the central spring housing.

With this arrangement the ball relief valve is subjected to the pump output pressure on the downstream (output) side of the flow orifice.

If the fluid output pressure exceeds some predetermined maximum, the ball will be dislodged from its seat, permitting fluid to escape from the control spring chamber, through the centre of the plunger and then out by way of the radial hole and annular groove in the plunger body. This fluid is then returned to the intake side of the pump via the bypass port.

Immediately this happens, the pressure p_2 in the control spring chamber drops, so that the increased pressure difference between both ends of the flow control plunger pushes back the plunger. As a result the bypass port will be uncovered, irrespective of the existing flow control conditions, so that a rapid pressure relief by way of the flow control plunger shoulder edge is obtained. It is the ball valve which senses any peak pressure fluctuation but it is the flow control valve which actually provides the relief passage for the excess of fluid. Once the ball valve closes, the pressure difference across the flow orifice for a given flow rate is again established so that the flow control valve will revert back to its normal flow limiting function.

9.2.6 Fault diagnosis procedure

Pump output check (Figs 9.12, 9.13, 9.15 and 9.18)

1 Disconnect the inlet hose which supplies fluid pressure from the pump to the control (reaction) valve, preferably at the control valve end.
2 Connect the inlet hose to the pressure gauge end of the combined pressure gauge and shut-off valve tester and then complete the hydraulic circuit by joining the shut-off valve hose to the control valve.
3 Top up the reservoir if necessary.
4 Read the maximum pressure indicated on type rating plate of pump or manufacturer's data.
5 Start the engine and allow it to idle with the shut-off valve in the open position.
6 Close the shut-off valve and observe the maximum pressure reached within a maximum time span of 10 seconds. Do not exceed 10 seconds, otherwise the internal components of the pump will be overworked and will heat up excessively with the result that the pump will be damaged.

7 The permissible deviation from the rated pressure may be   10 . If the pump output is low, the pump is at fault whereas if the difference is higher, check the functioning of the flow and pressure control valves.

An average maximum pressure figure cannot be given as this will depend upon the type and application of the power assistant steering. A typical value for maximum pressure may range from 45 bar for a ram type power unit to anything up to 120 bar or even more with an integral power unit and steering box used on a heavy commercial vehicle.

Power cylinder performance check (Figs 9.12, 9.13, 9.15 and 9.18)

1 Connect the combined pressure gauge and shut-off valve tester between the pump and control valve as under pump output check.
2 Open shut-off valve, start and idle the engine and turn the steering from lock to lock to bleed out any trapped air.
3 Turn the steering onto left hand full lock. Hold the steering on full lock and check pressure reading which should be within 10% of the pump output pressure.
4 Turn the steering onto the opposite lock and again check the pump output pressure.
5 If the pressure difference between the pump output and the power cylinder on both locks is greater than 10% then the power cylinder is at fault and should be removed for inspection.
6 If the pressure is low on one lock only, this indicates that the reaction control valve is not fully closing in one direction.

A possible cause of uneven pressure is that the control valve is not centralizing or that there is an internal fault in the valve assembly.

Binding check A sticking or binding steering action when the steering is moved through a portion of a lock could be due to the following:

a) Binding of steering joint ball joints or control valve ball joint due to lack of lubrication. Inspect all steering joints for seizure and replace where necessary.
b) Binding of spool or rotary type control valve. Remove and inspect for burrs wear and damage.

cessive free-play in the steering If when turning the driving steering wheel, the play before the steering road wheels taking up the response is excessive check the following;

1 worn steering track rod and drag link ball joints if fitted,
2 worn reaction control valve ball pin and cups,
3 loose reaction control valve location sleeve.

Heavy steering Heavy steering is experienced over the whole steering from lock to lock, whereas binding is normally only experienced over a portion of the front wheel steering movement. If the steering is heavy, inspect the following items:

1 External inspection   Check reservoir level and hose connections for leakage. Check for fan belt slippage or sheared pulley woodruff key and adjust or renew if necessary.
2 Pump output   Check pump output for low pressure. If pressure is below recommended maximum inspect pressure and flow control valves and their respective springs. If valve's assembly appears to be in good condition dismantle pump, examine and renew parts as necessary.
3 Control valve   If pump output is up to the manufacturer's specification dismantle the control valve. Examine the control valve spool or rotor and their respective bore. Deep scoring or scratches will allow internal leaks and cause heavy steering. Worn or damaged seals will also cause internal leakage.
4 Power cylinder   If the control valve assembly appears to be in good condition, the trouble is possibly due to excessive leakage in the power cylinder. If there is excessive internal power cylinder leakage, the inner tube and power piston ring may have to be renewed.

oisy operation To identify source of noise, check the following:

1 Reservoir fluid level   Check the fluid level as a low level will permit air to be drawn into the system which then will cause the control valve and power cylinder to become noisy while operating.
2 Power unit   Worn pump components will cause noisy operation. Therefore dismantle and examine internal parts for wear or damage.
3 If the reservoir and pump are separately located, check the hose supply from the reservoir to pump for a blockage as this condition will cause air to be drawn into the system.

Steering chatter If the steering vibrates or chatters check the following:

1 power piston rod anchorage may be worn or requires adjustment,
2 power cylinder mounting may be loose or incorrectly attached.

9.3 Steering linkage ball and socket oints

All steering linkage layouts are comprised of rods and arms joined together by ball joints. The ball joints enable track rods, drag-link rods and relay rods to swivel in both the horizontal and vertical planes relative to the steering arms to which they are attached. Most ball joints are designed to tilt from the perpendicular through an inclined angle of up to 20° for the axle beam type front suspension, and as much as 30° in certain independent front suspension steering systems.

9.3.1 Description of ball joint (Fig. 9.24(a–f))

The basic ball joint is comprised of a ball mounted in a socket housing. The ball pin profile can be divided into three sections; at one end the pin is parallel and threaded, the middle section is tapered and the opposite end section is spherically shaped. The tapered middle section of the pin fits into a similarly shaped hole made at one end of the steering arm so that when the pin is drawn into the hole by the threaded nut the pin becomes wedged.

The spherical end of the ball is sandwiched between two half hemispherical socket sets which may be positioned at right angles to the pin's axis (Fig. 9.24(a and b)). Alternatively, a more popular arrangement is to have the two half sockets located axially to the ball pin's axis, that is, one above the other (Fig. 9.24(c–f)).

The ball pins are made from steel which when heat treated provide an exceptionally strong tough core with a glass hard surface finish. These properties are achieved for normal manual steering applications from forged case-hardened carbon (0.15%) manganese (0.8%) steel, or for heavy duty power steering durability from forged induction hardened 3% nickel 1% chromium steel. For the socket housing which might also form one of the half socket seats, forged induction hardened steels such as a 0.35% carbon manganese 1.5% steel can be used. A 1.2% nickel 0.5% chromium steel can be used for medium and heavy heavy duty applications.

9.3.2 Ball joint sockets (Fig. 9.24(c–f))

Modern medium and heavy duty ball and socket joints may use the ball housing itself as the half socket formed around the neck of the ball pin. The other half socket which bears against the ball end of the ball pin is generally made from oil impregnated sintered iron (Fig. 9.24(c)); another type designed for automatic chassis lubrication, an induction hardened pressed steel half socket, is employed (Fig. 9.24(d)). Both cases are spring loaded to ensure positive contact with the ball at all times. A helical (slot) groove machined across the shoulder of the ball ensures that the housing half socket and ball top face is always adequately lubricated and at the same time provides a bypass passage to prevent pressurization within the joint.

Ball and socket oints for light and medium duty To reduce the risk of binding or seizure and to improve the smooth movement of the ball when it swivels, particularly if the dust cover is damaged and the joint becomes dry, non-metallic sockets are preferable. These may be made from moulded nylon and for some applications the nylon may be impregnated with molybdenum disulphide. Polyurethane and Teflon have also been utilized as a socket material to some extent. With the nylon sockets (Fig. 9.24(e)) the ball pin throat half socket and the retainer cap is a press fit in the bore of the housing end float. The coil spring accommodates initial settling of the nylon and subsequent wear and the retainer cap is held in position by spinning over a lip on the housing. To prevent the spring loaded half socket from rotating with the ball, two shallow tongues on the insert half socket engage with slots in the floating half socket. These ball joints are suitable for light and medium duty and for normal road working conditions have an exceptionally longer service life.

For a more precise adjustment of the ball and socket joint, the end half socket may be positioned by a threaded retainer cap (Fig. 9.24(f)) which is screwed against the ball until all the play has been taken up. The cap is then locked in position by crimping the entrance of the ball bore. A Belleville spring is positioned between the half socket and the screw retainer cap to preload the joint and compress the nylon.

9.3.3 Ball joint dust cover (Fig. 9.24(c–f))

An important feature for a ball type joint is its dust cover, often referred to as the *boot* or *rubber gaiter*, but usually made from either polyurethane or nitrile rubber mouldings, since both these materials have a high resistance to attack by ozone and do not tend to crack or to become hard and brittle at low temperature. The purpose of the dust cover is

Fig. 9.24 (a–) Steering ball unit

to exclude road dirt moisture and water, which if permitted to enter the joint would embed itself between the ball and socket rubbing surfaces. The consequence of moisture entering the working section of the joint is that when the air temperature drops the moisture condenses and floods the upper part of the joint. If salt products and grit are sprayed up from the road, corrosion and a mild grinding action might result which could quickly erode the glass finish of the ball and socket surfaces. This is then followed by the pitting of the spherical surfaces and a wear rate which will rapidly increase as the clearance between the rubbing faces becomes larger.

Slackness within the ball joint will cause wheel oscillation (shimmy), lack of steering response, excessive tyre wear and harsh or notchy steering feel.

Alternatively, the combination of grease, grit, water and salts may produce a solid compound which is liable to seize or at least stiffen the relative angular movement of the ball and socket joint, resulting in steering wander.

The dust boot must give complete protection against exposure from the road but not so good that air and the old grease cannot be expelled when the joint is recharged, particularly if the grease is pumped into the joint at high pressure, otherwise the boot will burst or it may be forced off its seat so that the ball and socket will become exposed to the surroundings.

The angular rotation of the ball joint, which might amount to 40° or even more, must be accommodated. Therefore, to permit relative rotation to take place between the ball pin and the dust cover, the boot makes a loose fit over the ball pin and is restrained from moving axially by the steering arm and ball pin shoulder while a steel ring is moulded into the dust cover to prevent the mouth of the boot around the pin spreading out (Fig. 9.24(c–f)). In contrast, the dust cover makes a tight fit over the large diameter socket housing by a steel band which tightly grips the boot.

9.3.4 Ball joint lubrication

Before dust covers were fitted, ball joints needed to be greased at least every 1600 kilometres (1000 miles). The advent of dust covers to protect the joint against dirt and water enabled the grease recharging intervals to be extended to 160 000 kilometres (10 000 miles). With further improvements in socket materials, ball joint design and the choice of lubricant the intervals between greasing can be extended up to 50 000 kilometres (30 000 miles) under normal road working conditions. With the

demand for more positive and reliable steering, joint lubrication and the inconvenience of periodic off the road time, automatic chassis lubrication systems via plastic pipes have become very popular for heavy commercial vehicles so that a slow but steady displacement of grease through the ball joint system takes place. The introduction to split socket mouldings made from non-metallic materials has enabled a range of light and medium duty ball and socket joints to be developed so that they are grease packed for life. They therefore require no further lubrication provided that the boot cover is a good fit over the socket housing and it does not become damaged in any way.

9.4 Steering geometry and wheel alignment

9.4.1 *heel track alignment using Dunlop optical measurement equipment   calibration of alignment gauges*

1 Fit contact prods onto vertical arms at approximately centre hub height.
2 Place each gauge against the wheel and adjust prods to contact the wheel rim on either side of the centre hub.
3 Place both mirror and view box gauges on a level floor (Fig. 9.25(b)) opposite each other so that corresponding contact prods align and touch each other. If necessary adjust the horizontal distance between prods so that opposing prods are in alignment.
4 Adjust both the mirror and target plate on the viewbox to the vertical position until the reflection of the target plate in the mirror is visible through the periscope tube.
5 Look into the periscope and swing the indicator pointer until the view box hairline is positioned in the centre of the triangle between the two thick vertical lines on the target plate.
6 If the toe-in or -out scale hairline does not align with the zero reading on the scale, slacken off the two holding down screws and adjust indicator pointer until the hairline has been centred. Finally retighten screws.

oe-in or -out check (Fig. 9.25(a, b and c))

1 Ensure that tyre pressures are correct and that wheel bearings and track rod ends are in good condition.
2 Drive or push the vehicle in the forward direction on a level surface and stop. Only take

readings with the vehicle rolled forward and never backwards as the latter will give a false toe angle reading.

3 With a piece of chalk mark one tyre at ground level.

4 Place the mirror gauge against the left hand wheel and the view box gauge against the right hand wheel (Fig. 9.25(b)).

5 Push each gauge firmly against the wheels so that the prods contact the wheel on the smooth surface of the rim behind the flanged turnover since the edge of the latter may be slightly distorted due to the wheel scraping the kerb when the vehicle has been parked. Sometimes gauges may be held against the wheel rim with the aid of rubber bands which are hooked over the tyres.

6 Observe through the periscope tube the target image. Swing the indicator pointer to and fro over the scale until the hairline in the view box coincides with the centre triangle located between the thick vertical lines on the target plate which is reflected in the mirror.

7 Read off the toe-in or -out angle scale in degrees and minutes where the hairline aligns with the scale.

8 Check the toe-in or -out in two more positions by pushing the vehicle forward in stages of a third of a wheel revolution observed by the chalk mark on the wheel. Repeat steps 4 to 7 in each case and record the average of the three toe angle readings.

9 Set the pointer on the dial calculator to the wheel rim diameter and read off the toe-in
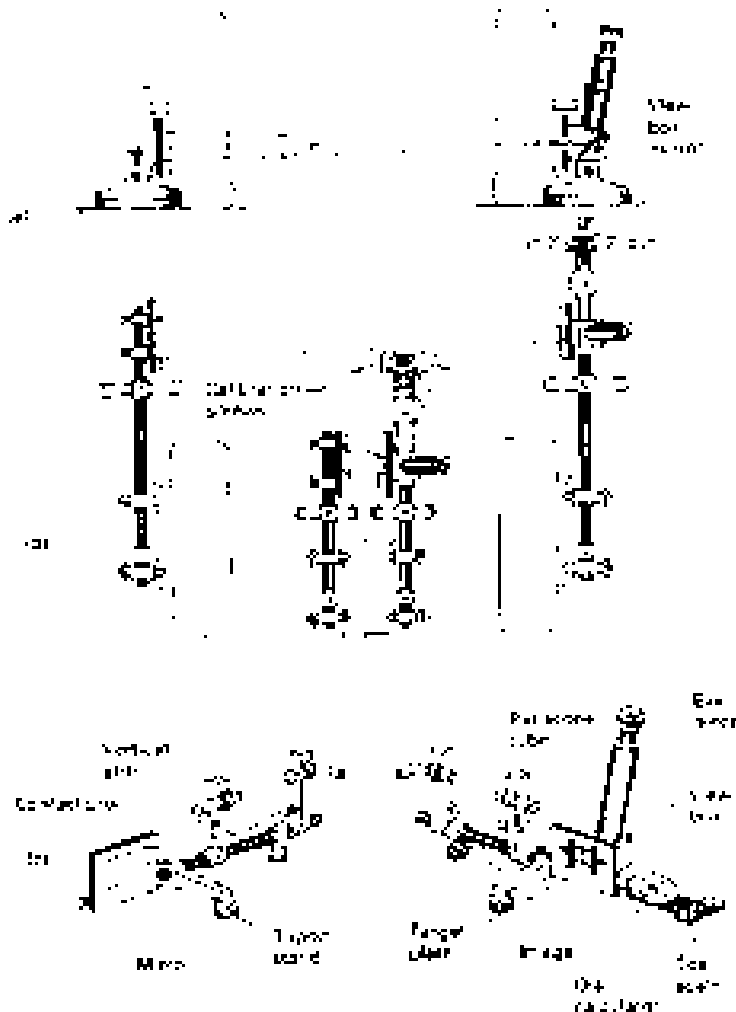
Fig. 9.25 (a–c) Wheel track alignment using the Dunlop equipment

or -out in millimetres opposite the toe angle reading obtained on the toe-in or -out scale. Alternatively, use Table 9.1 to convert the toe-in or -out angle to millimetres.

10 If the track alignment is outside the manufacturer's recommendation, slacken the track rod locking bolts or nuts and screw the track rods in or out until the correct wheel alignment is achieved. Recheck the track toe angle when the track rod locking devices have been tightened.

9.4.2 *heel track alignment using Churchill line cord measurement equipment*

Calibration of alignment gauges
(Fig. 9.26(a))

1 Clamp the centre of the calibration bar in a vice.
2 Attach an alignment gauge onto each end of the calibration bar.
3 Using the spirit bubble gauge, level both of the measuring gauges and tighten the clamping thumbscrews.
4 Attach the elastic (rubber) calibration cord between adjacent uncoloured holes formed in each rotor.
5 Adjust measuring scale by slackening the two wing nuts positioned beneath each measuring scale, then move the scale until the zero line aligns exactly with the red hairline on the pointer lens. Carefully retighten the wing nuts so as not to move the scale.
6 Detach the calibration cord from the rotors and remove the measuring gauges from calibration bar.

oe-in or -out check (front or rear wheels)
(Fig. 9.26(a))

1 Position a wheel clamp against one of the front wheels so that two of the threaded contact studs mounted on the lower clamp arm rest inside the rim flange in the lower half of the wheel. For aluminium wheels change screw studs for claw studs provided in the kit.
2 Rotate the tee handle on the centre adjustment screw until the top screw studs mounted on the upper clamp arm contact the inside rim flange in the upper half of the wheel. Fully tighten centre adjustment screw tee handle to secure clamp to wheel.
3 Repeat steps 1 and 2 for opposite side front wheel.

4 Push a measuring gauge over each wheel clamp stub shaft and tighten thumbscrews. This should not prevent the measuring gauge rotating independently to the wheel clamp.
5 Attach the elastic cord between the uncoloured hole in the rotor of each measuring gauge.
6 Wheel lateral run-out is compensated by the following procedure of steps 7–10.
7 Lift the front of the vehicle until the wheels clear the ground and place a block underneath one of the wheels (in the case of front wheel drive vehicles) to prevent it from rotating.
8 Position both measuring gauges horizontally and hold the measuring gauge opposite the blocked wheel. Slowly rotate the wheel one complete revolution and observe the measuring gauge reading which will move to and fro and record the extreme of the pointer movement on the scale. Make sure that the elastic cord does not touch any part of the vehicle or jack.
9 Further rotate wheel in the same direction until the mid-position of the wheel rim lateral run-out is obtained, then chalk the tyre at the 12 o'clock position.

ab e 9.1 Conversion of degrees to millimetres

Degree	Rim size					
	0 mm	2 mm	3 mm	4 mm	5 mm	mm
5	0.40	0.48	0.53	0.57	0. 0	0. 4
0	0.80	0.9	.0	. 3	.2	.28
5	.20	.44	.59	.70	.8	.92
20	. 0	.92	2. 2	2.27	2.42	2.5
25	2.00	2.40	2. 5	2.84	3.02	3.20
30	2.40	2.88	3. 9	3.40	3. 2	3.84
35	2.80	3.3	3.72	3.97	4.22	4.48
40	3.20	3.84	4.25	4.54	4.83	5. 2
45	3. 0	4.32	4.78	5.	5.43	5.7
50	4.00	4.80	5.3	5. 7	.03	.40
55	4.40	5.28	5.84	.24	. 4	7.04
.00	4.80	5.7	.37	.8	7.24	7. 8
.05	5.20	.24	.90	7.38	7.85	8.32
. 0	5. 0	.72	7.43	7.95	8.45	8.9
. 5	.00	7.20	7.9	8.5	9.0	9. 0
.20	.40	7. 8	8.49	9.07	9.	0.24
.25	.80	8.	9.03	9. 4	0.25	0.88
.30	7.20	8. 4	9.5	0.2	0.8	.52
.35	7. 0	9. 2	0.09	0.78	.47	2.
.40	8.00	9. 0	0. 2	.35	2.08	2.80
.45	8.40	0.08	. 5	.9	2. 8	3.44
.50	8.80	0.5	. 8	2.48	3.28	4.08
.55	9.20	.04	2.2	3.05	3.89	4.72
2.00	9. 0	.52	2.75	3. 2	4.49	5.3
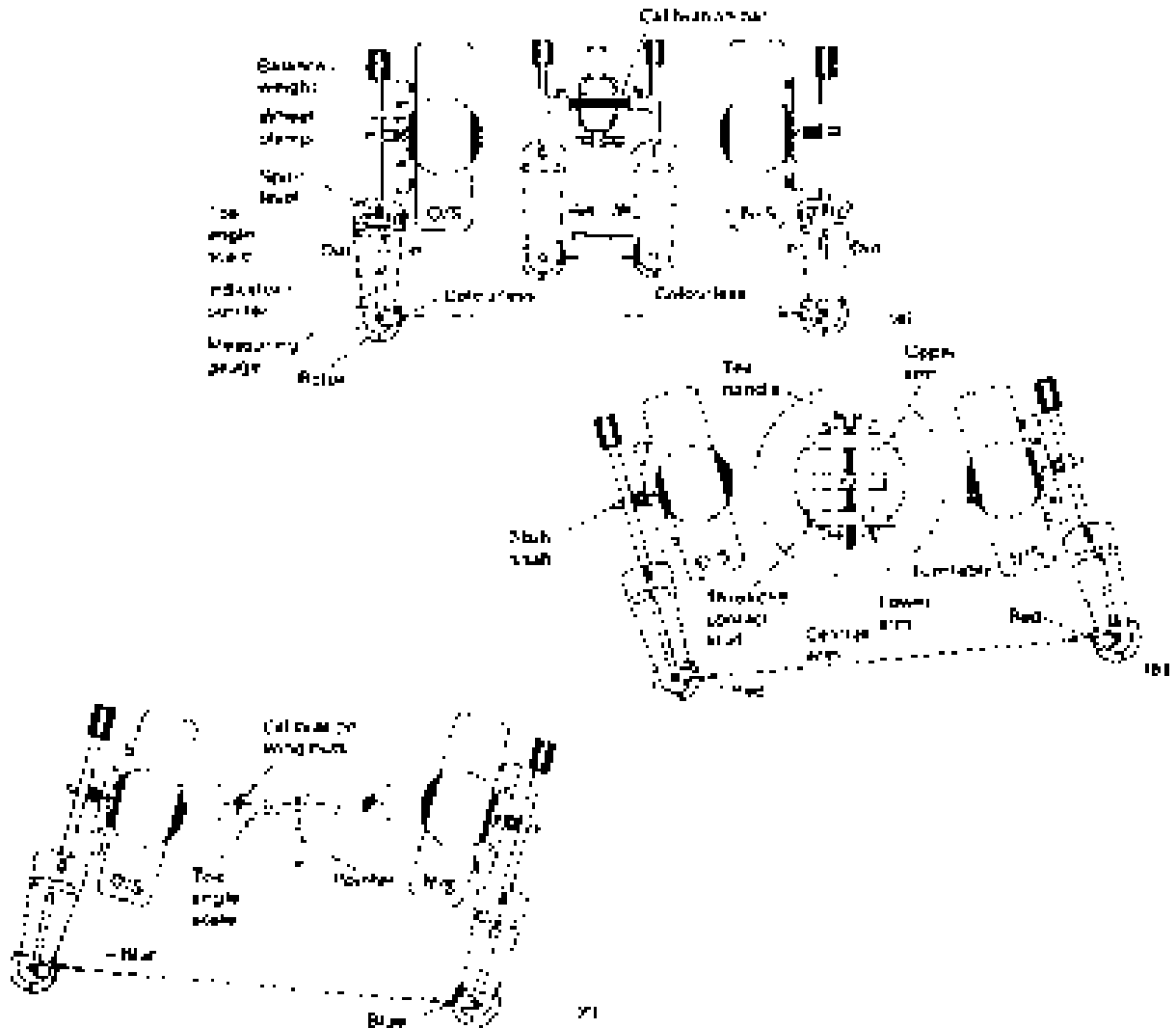
Fig. 9.26 (a–c) Wheel track alignment using the Churchill equipment

10 Repeat steps 7 to 9 for the opposite side front wheel.

11 Position each front wheel with the chalk mark at 12 o'clock.

12 Utilize the brake pedal depressor to prevent the wheels from rotating.

13 Slide a turntable underneath each front wheel, remove the locking pins and then lower the front wheels onto both turntables.

14 Bounce the front of the vehicle so that the suspension quickly settles down to its normal height.

15 Tilt each measuring gauge to the horizontal position by observing when the spirit level bubble is in the mid-position. Lock the measuring gauge in the horizontal position with the second thumbscrew.

16(a) Observe the left and right toe angle reading and add them together to give the combined toe angle of the front wheels.

(b) Alternatively, turn one road wheel until its measuring gauge pointer reads zero, then read the combined toe angle on the opposite side measuring gauge (front wheels only).

342

17 Using Table 9.1 provided, convert the toe angle into track toe-in or -out in millimetres and compare with the manufacturer's recommendations.

oe-out on turns check (Fig. 9.26(b and c))

1 After completing the toe-in or -out check, keep the wheel clamp and measuring gauge assembly attached to each front wheel. Maintain the mid-wheel lateral run-out position with the tyre at the 12 o'clock position and ensure the brake pedal depressor is still applied.
2 Transfer the position of the elastic cord hook ends attached to the measuring gauge rotors from the uncoloured holes to the red holes which are pitched 15° relative to the uncoloured holes.
3 Rotate the right hand (offside) wheel in the direction the arrow points on the measuring gauge facing the red hole in which the cord is hooked until the scale reads zero. At this point the right hand wheel (which becomes the outer wheel on the vehicle turning circle) will have been pivoted 15°. Make sure that the cord does not touch any obstruction.
4 Observe the reading on the opposite left hand (near side) measuring gauge scale, which is the toe-out turns angle for the left hand (near side) wheel (the inner wheel on the vehicle's turning circle).
5 Change the cord to the blue holes in each measuring gauge rotor.
6 Rotate the left hand (near side) wheel in the direction the arrow points on the measuring gauge facing the blue hole until the hairline pointer on the left hand measuring gauge reads zero.
7 Read the opposite right hand (offside) wheel measuring gauge scale which gives the toe-out on turns for the right hand (offside) wheel (the inner wheel on the vehicle's turning circle).
8 Compare the left and right hand toe-out turns readings which should be within one degree of one another.

9.4.3 Front to rear wheel misalignment
(Fig. 9.27(a))
An imaginary line projected longitudinally between the centre of the front and rear wheel tracks is known as the vehicle's *centre line* or the *axis of symmetry* (Fig. 9.27(a)). If the vehicle's body and suspension alignment is correct, the vehicle will travel in the same direction as the axis of symmetry. When the wheel axles at the front and rear are misaligned, the vehicle will move forward in a skewed line relative to the axis of symmetry. This second directional line is known as the *thrust axis* or *driving axis*. The angle between the axis of symmetry and the thrust axis is referred to as the *thrust axis deviation* which will cause the front and rear wheels to be laterally offset to each other when the vehicle moves in the straight ahead direction.

If the vehicle has been constructed and assembled correctly the thrust axis will coincide with the axis of symmetry, but variation in the rear wheel toe angles relative to the axis of symmetry will cause the vehicle to be steered by the rear wheels. As a result, the vehicle will tend to move in a forward direction and partially in a sideway direction. The vehicle will therefore tend to pull or steer to one side and when driving round a bend the steering will oversteer on one lock and understeer on the opposite lock. In the case of Fig. 9.27(a), with a right hand lateral offset the vehicle will understeer on left hand bends and oversteer on right hand turns.

The self-steer effect of the rear wheels due to track or axle misalignment will conflict with the suspension geometry such as the kingpin inclination and castor which will therefore attempt to direct the vehicle along the axis of symmetry. Consequently, the tyres will be subjected to excessive scrub.

Thrust axis deviation may be produced by body damage displacing the rear suspension mounting points, rear suspension worn bushes, poorly located leaf springs and distorted or incorrectly assembled suspension members.

ront to rear alignment check using Churchill line cord measurement equipment (Fig. 9.28(a and b))

1 Check rear wheel toe angle by using the procedure adopted for front wheel toe angle measurement (Fig. 9.26(a)). Use the convention that toe-in is positive and toe-out is negative.
2 eep the wheel clamp and measuring gauge assembly on both rear wheels.
3 Attach a second pair of wheel clamps to both front wheels.
4 Remove the rear wheel toe elastic cord from the two measuring gauges.
5 Hook a front to rear alignment elastic cord between the stub shaft deep outer groove of the front wheel clamps and the single hole in the measuring gauge rotors set at 90° from the middle hole of the three closely spaced holes (Fig. 9.28(a and b)).

Fig. 9.27 (a and b)   Front to rear wheel alignment procedure

344

6 Apply a slight tension to the front to rear alignment cord using the metal plate adjusters.

7 With all four wheels pointing in the straight ahead direction, read and record the left and right hand measuring gauge scales (Fig. 9.27 (a and b)). To determine the thrust axis deviation (TAD) angle subtract the left reading from the right reading and divide the difference of the reading by two.

i.e.

$$\text{Thrust axis deviation (TAD) angle} = \frac{R-}{2}$$

where R = Right hand measuring gauge reading

= Left hand measuring gauge reading

8 The lateral offset can be approximately determined from the formula

$$\text{Lateral offset} \quad \text{Wheel base} \times \tan\left(\frac{R-}{2}\right)$$

however, the makers of the equipment supply Table 9.2 to simplify the conversion from thrust axis deviation angle to lateral offset.

xample (Fig. 9.27(b)) Determine the rear wheel toe-in or -out and the front to rear lateral offset for a 3000 mm wheelbase vehicle having a rigid rear axle and 13 inch diameter wheels from the following information:

Left rear wheel toe angle reading	0
Right rear wheel toe angle reading	0
Left side front to rear measurement reading out (out negative)	−30
Right side front to rear measurement reading in (in positive)	+30

a) Toe-in or -out:
   Rear wheel combined toe angle $= 0 + 0 = 0$
   Thus wheels are *parallel*.
b) Lateral offset:

$$\text{Thrust axis deviation} = \frac{R-}{2}$$

$$= \frac{+30 - (-30)}{2}$$

$$= \frac{+30 + 30}{2}$$

$$= \frac{60}{2} = 30$$

Note A minus minus makes a plus; $-(-) = +$

From lateral offset Table 9.2, a thrust axis deviation of 30 for a wheel base of 3000 mm is equivalent to a lateral offset to the right of 22 mm when the vehicle moves in a forward direction.

xample (Fig. 9.27(b)) Determine the rear wheel toe-in or -out and the front to rear lateral offset of a vehicle having independent rear suspension from the following data:

Wheelbase	3400 mm
Wheel diameter	13 inches
Left rear wheel toe angle reading out (out negative)	−40
Right rear wheel toe angle reading in (in positive)	+15
Left side front to rear measuring gauge reading out	−55
Right side front to rear measuring gauge reading in	+25

a) Toe-in or -out:
   Rear wheel combined toe angle $= (-40) + (+15) = 25$
   From toe conversion table a toe angle of −25 for a 13 inch diameter wheel is equivalent to a toe-out of 2.65 mm.

b) Lateral offset:

$$\text{Thrust axis deviation (TAD) angle} = \frac{R-}{2}$$

$$= \frac{+25 - (-55)}{2}$$

$$= \frac{+25 + 55}{2}$$

$$= \frac{80}{2} = 40$$

From lateral offset Table 9.2, a thrust axis deviation of 40 for a wheelbase of 3400 mm is equivalent to a lateral offset to the right of 33.5 mm when the vehicle is moving in the forward direction.

9.4.4 Six wheel vehicle with tandem rear axle steering geometry (Fig. 9.28)

For any number of road wheels on a vehicle to achieve true rolling when cornering, all projected lines drawn through each wheel axis must intersect at one common point on the inside track, this being the instantaneous centre about which the vehicle travels. In the case of a tandem rear axle arrangement in which the axles are situated parallel to each

Lateral offset of front wheels in relation to rear wheels (Measurements in millimeters)						
Wheelbase mm	0°	20°	30°	40°	50°	0°
800	4.0	7.5	.5	5.0	9.0	23.0
2000	4.5	8.5	3.0	7.5	22.0	2 .0
2200	5.0	0.0	5.0	20.0	24.5	30.0
2400	5.5	.0	.5	22.0	27.5	33.5
2 00	.0	2.0	8.5	24.5	30.5	37.0
2800	.5	3.5	20.0	2 .5	33.5	40.5
3000	7.0	4.5	22.0	29.0	3 .5	44.0
3200	8.0	5.5	23.5	3 .5	39.0	47.5
3400	8.5	7.0	25.0	33.5	42.0	5 .0
3 00	9.0	8.0	27.0	3 .0	45.0	54.5
3800	9.5	9.0	28.5	38.5	48.0	58.0

(Measurements in inches)						
Wheelbase ft in	0°	20°	30°	40°	50°	0°
0	0.2	0.3	0.5	0.	0.8	0.9
	0.2	0.3	0.5	0.7	0.8	.0
7 0	0.2	0.4	0.	0.7	0.9	.
7	0.2	0.4	0.	0.8	.0	.2
8 0	0.2	0.4	0.7	0.9	.	.3
8	0.2	0.5	0.7	.0	.2	.4
9 0	0.3	0.5	0.8	.0	.3	.5
9	0.3	0.5	0.8	.	.4	.
0 0	0.3	0.	0.9	.2	.5	.7
0	0.3	0.	0.9	.2	.5	.9
0	0.3	0.7	.0	.3	.	2.0
	0.3	0.7	.0	.4	.7	2.

A negative (−) value indicates front wheels offset to left
A positive (+) value indicates front wheels offset to right

other, lines projected through the axles would never meet and in theory true rolling cannot exist. However, an approximate instantaneous centre for the steered vehicle can be found by projecting a line mid-way and parallel between both rear axles, this being assumed to be the common axis of rotation (Fig. 9.30). Extended lines passing through both front wheel stub axles, if made to intersect at one point somewhere along the common projected single rear axle line, will then produce very near true rolling condition as predicted by the Ackermann principle.

Improvements in rear axle suspension design have introduced some degree of roll steer which minimizes tyre scrub on the tandem axle wheels.

This is achieved by the camber of the leaf springs supporting the rear axles changing as the body rolls so that both rear axles tend to skew in the plan view
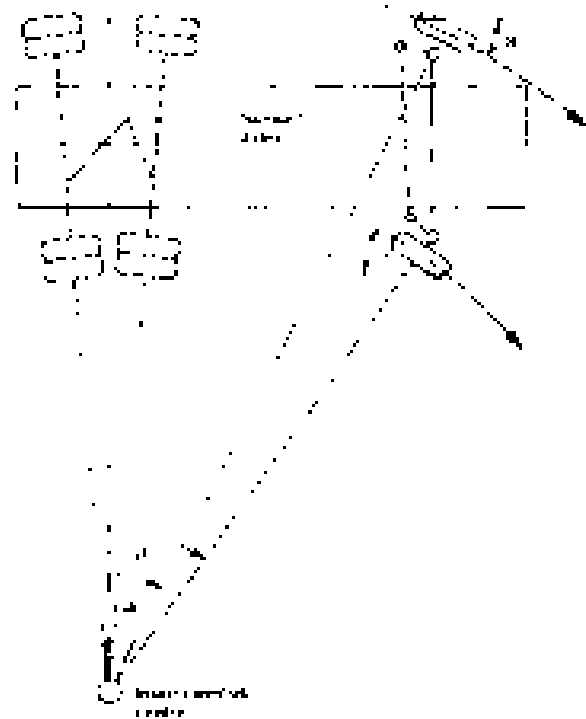
Fig. 9.28 Six wheel tandem rear axle vehicle steering geometry

so that the imaginary extended lines drawn through both rear axles would eventually meet. Unfortunately lines drawn through the front steered stub axles and the rear skewed axles may not all meet at one point. Nevertheless, they may almost merge so that very near true rolling can occur for a large proportion of the steering angle when the vehicle is in motion. The remainder of the rear axle wheel misalignment is absorbed by suspension spring distortion, shackle joints or torque arm rubber joints, and tyre compliance or as undesirable tyre scrub.

9.4.5 Dual front axle steering
Operating large rigid trucks with heavy payloads makes it necessary in addition to utilizing tandem axles at the rear to have two axles in the front of the vehicle which share out and support the load.

Both of the front axles are compelled to be steer axles and therefore need to incorporate steering linkage such as will produce true or near true rolling when the vehicle is driven on a curved track.

The advantages gained by using dual front steering axles as opposed to a single steer axle are as follows:

346

1 The static payload is reduced per axle so that static and dynamic stresses imposed on each axle assembly are considerably lessened.
2 Road wheel holding is improved with four steered wheels as opposed to two, particularly over rough ground.
3 Road wheel impact shocks and the subsequent vibrations produced will be considerably reduced as the suspension for both sets of wheels share out the vertical movement of each axle.
4 Damage to one axle assembly or a puncture to one of the tyres will not prevent the vehicle being safely steered to a standstill.
5 Tyre wear rate is considerably reduced for dual axle wheels compared to single axle arrangements for similar payloads. Because the second axle wheels have a smaller turning angle relative to the foremost axle wheels, the tyre wear is normally less with the second axle road wheels.

A major disadvantage with dual front axles is that it is unlikely in practice that both instantaneous centres of the first and second stub axle turning circles will actually intersect at one point for all angles of turn. Therefore tyre scrub may be excessive for certain angles of steering wheel rotation.

ual front a le steering geometry (Fig. 9.29) When a pair of axles are used to support the front half of a vehicle each of these axles must be steered
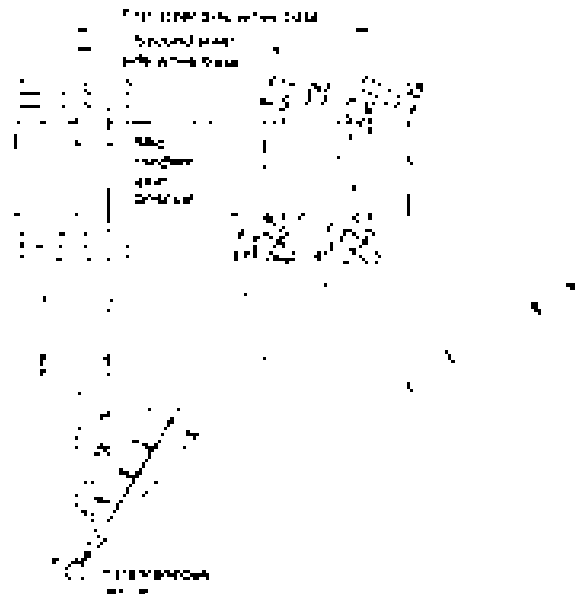
Fig. 9.29 Dual front axle steering Ackermann geometry
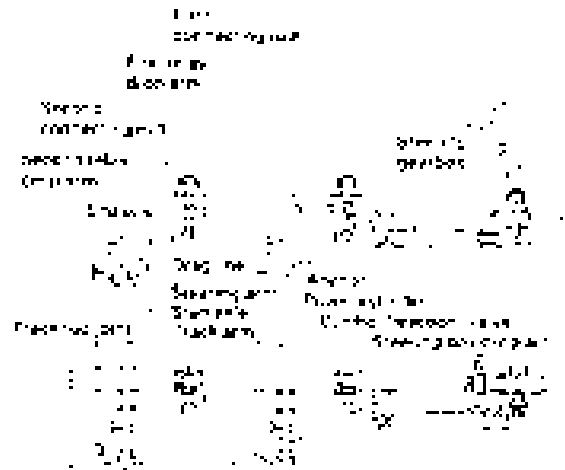
Fig. 9.30 Dual front axle steering linkage layout with power assistance

if the vehicle is to be able to negotiate a turning circle.

For a dual front axle vehicle to be steered, the Ackermann principle must apply to each of the front axles. This means that each axle has two wheels pivoted at each end of its beam. To enable true rolling of the wheels to take place when the vehicle is travelling along a curved track, lines drawn through each of these four stub axles must intersect at a common centre of rotation, somewhere along the extended line drawn between the tandem rear axles (Fig. 9.29).

Because the wheelbase between the first front axle is longer than the second front axle, relative to the mid-tandem axle position, the turning angles of both first front wheels will be greater than those of the second front axle wheels. The correct angle difference between the inner and outer wheels of each axle is obtained with identical Ackermann linkage settings, whereas the angle differential between the first and second axles is formed by the connecting rod ball joint coupling location on both relay drop arms being at different distances from their respective pivot point.

The dual steering linkage with power assistance ram usually utilizes a pair of swing relay drop arms bolted onto the chassis side member with their free ends attached to each axle drag link (Fig. 9.30).

The input work done to operate the steering is mainly supplied by the power cylinder which is coupled by a ball joint to the steering gearbox drop arm at the front and the power piston rod is anchored through a ball joint and bracket to the

chassis at the rear end. To transfer the driver's input effort and power assistance effort to both steer axles, a forward connecting rod links the front end of the power cylinder to the first relay drop arm. A second relay connecting rod then joins both relay arms together.

A greater swivel movement of the first pair of stub axles compared to the second is achieved (Fig. 9.31) by having the swing drop arm effective length of the first relay AB shorter than the second relay arm A B . Therefore the second relay arm push or pull movement will be less than the input swing of the first relay arm. As a result, the angular swing of the first relay, $= 20°$, will be less than for the second relay arm angular displacement, $= 14°$.

ual front a le alignment checks using unlop optical measurement equipment (Fig. 9.32(a–d))

1 Roll or drive forward. Check the toe-in or -out of both pairs of front steering wheels and adjust track rods if necessary (Fig. 9.32(a)).

2 Assemble mirror gauge stand with the mirror positioned at right angles to the tubular stand. Position the mirror gauge against a rear axle wheel (preferably the nearest axle to the front) with the mirror facing towards the front of the vehicle (Fig. 9.32(b)).

3 Place the view box gauge stand on the floor in a transverse position at least one metre in front of the vehicle so that the view box faces the mirror (Fig. 9.32(c)). Move the view box stand across until the reflected image is centred in the view box with a zero reading on the scale. Chalk mark the position of the view box tripod legs on the ground.

4 Bring the mirror gauge stand forward to the first steer axle wheel and place gauge prods against wheel rim (Fig. 9.34(c)).

5 If both pairs of steer axle wheels are set parallel (without toe-in or -out), set the pointer on the toe angle scale to zero. Conversely, if both steer axles have toe-in or -out settings, move the pointer on the toe angle scale to read half the toe-in or -out

Fig. 9.31 Dual front axle steering interconnecting relay linkage principle
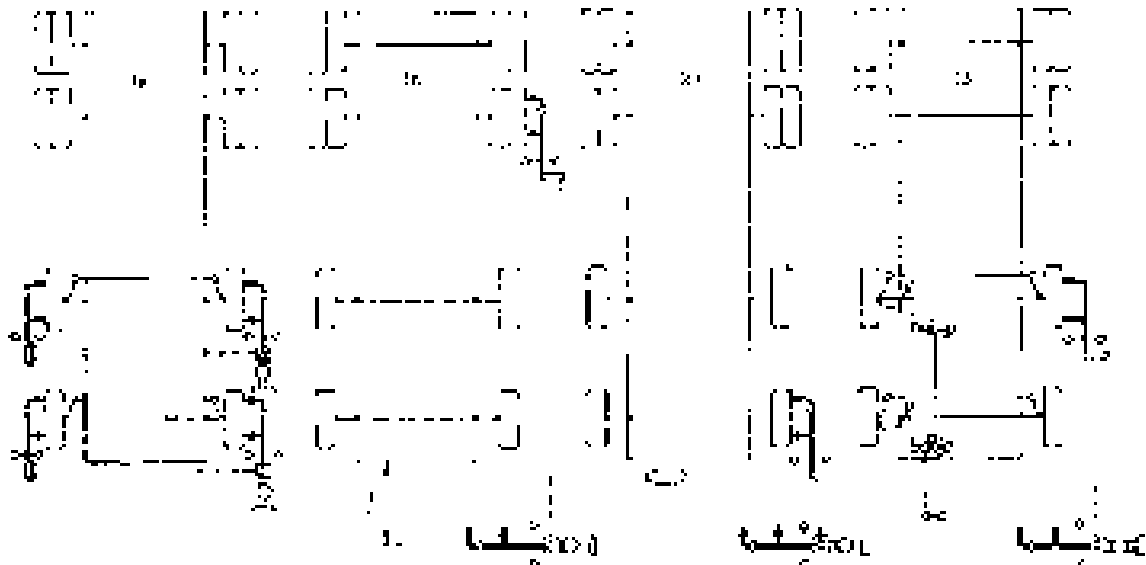
Fig. 9.32 Dual front axle wheel alignment procedure

figure, i.e. with a track toe angle of 30 set the pointer to read 15 .

6 Look through the periscope tube and with an assistant move the driver's steering wheel in the cab until the hairline is central in the view box (Fig. 9.32(c)). At the same time make sure that the mirror gauge prods are still in contact with the front wheel rim. The first front steer axle is now aligned to the first rigid rear axle.

7 Move the mirror gauge from the first steer axle wheel back to the second steer axle wheel and position the prods firmly against the wheel rim (Fig. 9.32(d)).

8 Look into the periscope. The hairline in the view box should be centrally positioned with the toe angle pointer still in the same position as used when checking the first steer axle (Fig. 9.32(d)).

If the hairline is off-centre, the relay connecting rod between the two relay idler arms should be adjusted until the second steer axle alignment relative to the rear rigid axle and the first steer axle has been corrected. Whilst carrying out any adjustment to the track rods or relay connecting rod, the overall wheel alignment may have been disturbed. Therefore a final check should be made by repeating all steps from 2 to 8.

9.4.6 Steer angle dependent four wheel steering system (Honda)

This steer angle dependent four wheel steering system provides dual steering characteristics enabling *same direction steer* to take place for small steering wheel angles. This then changes to *opposite direction steer* with increased steering wheel deviation from the straight ahead position. Both of these steer characteristics are explained as follows:

Opposite direction steer (Fig. 9.33)   At low speed and large steering wheel angles the rear wheels are turned by a small amount in the opposite direction to the front wheels to improve manoeuvrability when parking (Fig. 9.32). In effect opposite direction steer reduces the car's turning circle but it does have one drawback; the rear wheels tend to bear against the side of the kerb. Generally there is sufficient tyre sidewall distortion and suspension compliance to accommodate the wheel movement as it comes into contact with the kerb so that only at very large steering wheel angles can opposite direction steer becomes a serious problem.

Same direction steer (Fig. 9.33)   At high speed and small steering wheel angles the rear wheels are
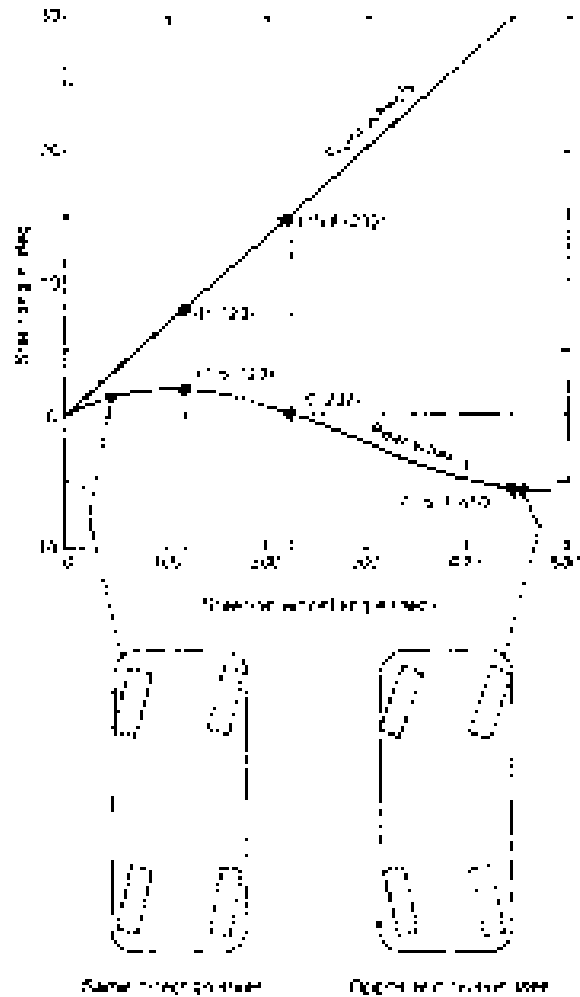
Fig. 9.33 Front and rear wheel steer relationship to driver's steering wheel angular movement

turned a small amount in the same direction as the front wheels to improve both steering response and stability at speed (Fig. 9.33). This feature is particularly effective when changing lanes on motorways. Incorporating a same direction steer to the rear wheels introduces an understeer characteristic to the car because it counteracts the angular steering movement of the front wheels and consequently produces a stabilizing influence in the high speed handling of the car.

 ront and rear road wheel response relative to the steering wheel angular movement (Fig. 9.33)   Moving the steering wheel approximately 120° from its central position twists the front wheels 8° from the

straight ahead position. Correspondingly, it moves the eccentric shaft peg to its maximum horizontal annular gear offset, this being equivalent to a maximum 1.5° same direction steer for the rear road wheels (Fig. 9.33).

Increasing the steering wheel rotation to 232° turns the front wheels 15.6° from the straight ahead position which brings the planetary peg towards the top of the annular gear and in vertical alignment with the gear's centre. This then corresponds to moving the rear wheels back to the straight ahead position (Fig. 9.33).

Further rotation of the steering wheel from the straight ahead position orbits the planetary gear over the right hand side of the annular gear. Accordingly the rear wheels steadily move to the opposite direction steer condition up to a maximum of 5.3° when the driver's steering wheel has been turned roughly 450° (Fig. 9.33).

our wheel steer (S) layout (Fig. 9.34)   The steering system is comprised of a rack and pinion front steering box and a rear epicyclic steering box coupled together by a central drive shaft and a pair of Hooke's universal end joints (Fig. 9.35). Both front and rear wheels swivel on ball suspension joints which are steered by split track rods actuating steering arms. The front road wheels are interlinked by a rack and transverse input movement to the track rods is provided by the input pinion shaft which is connected to the driver's steering wheel via a split steering shaft and two universal joints. Steering wheel movement is relayed to the rear steering box by way of the front steering rack which meshes with an output pinion shaft. This movement of the front rack causes the output pinion and centre drive shaft to transmit motion to the rear steering box. The rear steering box mechanism then con-

verts the angular input shaft motion to a transverse linear movement. This is then conveyed to the rear wheel swivels by the stroke rod and split track rods.

Rear steering bo   construction (Fig. 9.35)   The rear steering box is basically formed from an epicyclic gear set consisting of a fixed internally toothed annular ring gear in which a planetary gear driven by an eccentric shaft revolves (Fig. 9.35). Motion is transferred from the input eccentric shaft to the planetary gear through an offset peg attached to a disc which is mounted centrally on the eccentric shaft. Rotation of the input eccentric shaft imparts movement to the planetary gear which is forced to orbit around the inside of the annular gear. At the same time, motion is conveyed to the guide fork via a second peg mounted eccentrically on the face of the planetary gear and a slider plate which fits over the peg (Fig. 9.35). Since the slider plate is located between the fork fingers, the rotation of the planetary gear and peg causes the slider plate to move in both a vertical and horizontal direction. Due to the construction of the guide fork, the slider plate is free to move vertically up and down but is constrained in the horizontal direction so that the stroke rod is compelled to move transversely to and fro according to the angular position of the planetary gear and peg.

Adopting this combined epicyclic gear set with a slider fork mechanism enables a small same direction steer movement of the rear wheels to take place for small deviation of the steering wheel from the straight ahead position. The rear wheels then progressively change from a same direction steer movement into an opposite steer displacement with larger steering angles.

The actual steering wheel movement at which the rear wheels change over from the same direc-
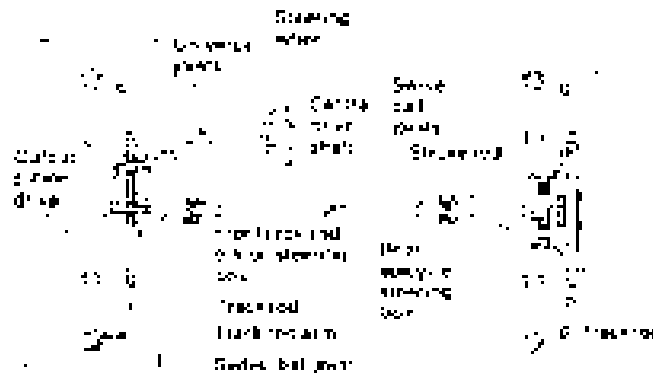
Fig. 9.34 Four wheel steering (4WS) system
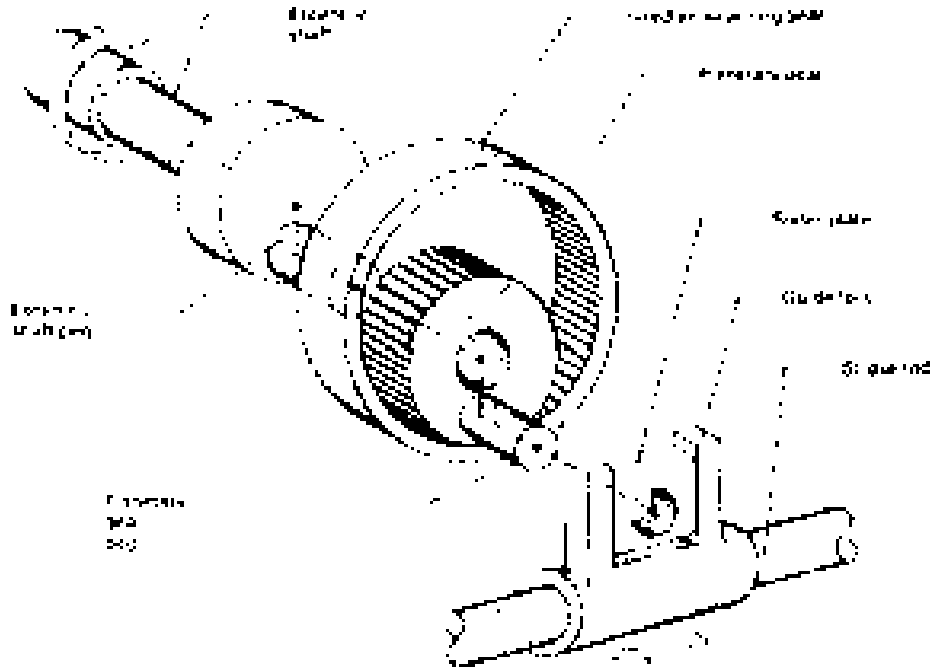
Fig. 9.35 Epicyclic rear steering box

tion steer to the opposite direction steer and the magnitude of the rear wheel turning angles relative to both conditions are dependent upon the epicyclic gear set gear ratio chosen.

Rear steering bo operation (Fig. 9.36(a–e)) The automatic correction of the rear road wheels from a same direction steer to opposite direction steer with increasing front road wheel turning angle and vice versa is explained by Fig. 9.36(a–e).

Central position With the steering wheel in the straight ahead position, the planetary gear sits at the bottom of the annular gear with both eccentric shaft and planetary pegs located at bottom dead centre in the mid-position (Fig. 9.36(a)).

° *eccentric shaft peg rotation* Rotating the eccentric shaft through its first quadrant (0–90°) in a chosen direction from the bottom dead centre position compels the planetary gear to roll in an anticlockwise direction up the left hand side of the annular ring gear. This causes the planetary peg and the stroke rod to be displaced slightly to the left (Fig. 9.36(b)) and accordingly makes the rear wheels move to a same direction steer condition.

° *eccentric shaft peg rotation* Rotating the eccentric shaft through its second quadrant (90–180°) causes the planetary gear to roll anticlockwise inside the annular gear so that it moves with the eccentric peg to the highest position. At the same time, the planetary peg orbits to the right hand side of the annular gear centre line (Fig. 9.36(c)) so that the rear road wheels turn to the opposite direction steer condition.

° *eccentric shaft peg rotation* Rotating the eccentric shaft through a third quadrant (180–270°) moves the planetary gears and the eccentric shaft peg to the 270° position, causing the planetary peg to orbit even more to the right hand side (Fig. 9.36(d)). Consequently further opposite direction steer will be provided.

° *eccentric shaft peg rotation* Rotating the eccentric shaft through a fourth quadrant (270–360°) completes one revolution of the eccentric shaft. It therefore brings the planetary gear back to the base of the annular ring gear with the eccentric shaft peg in its lowest position (Fig. 9.36(e)). The planetary peg will have moved back to the central position, but this time the peg is in its highest position. The front to rear wheel steering drive gearing is normally so arranged that

351
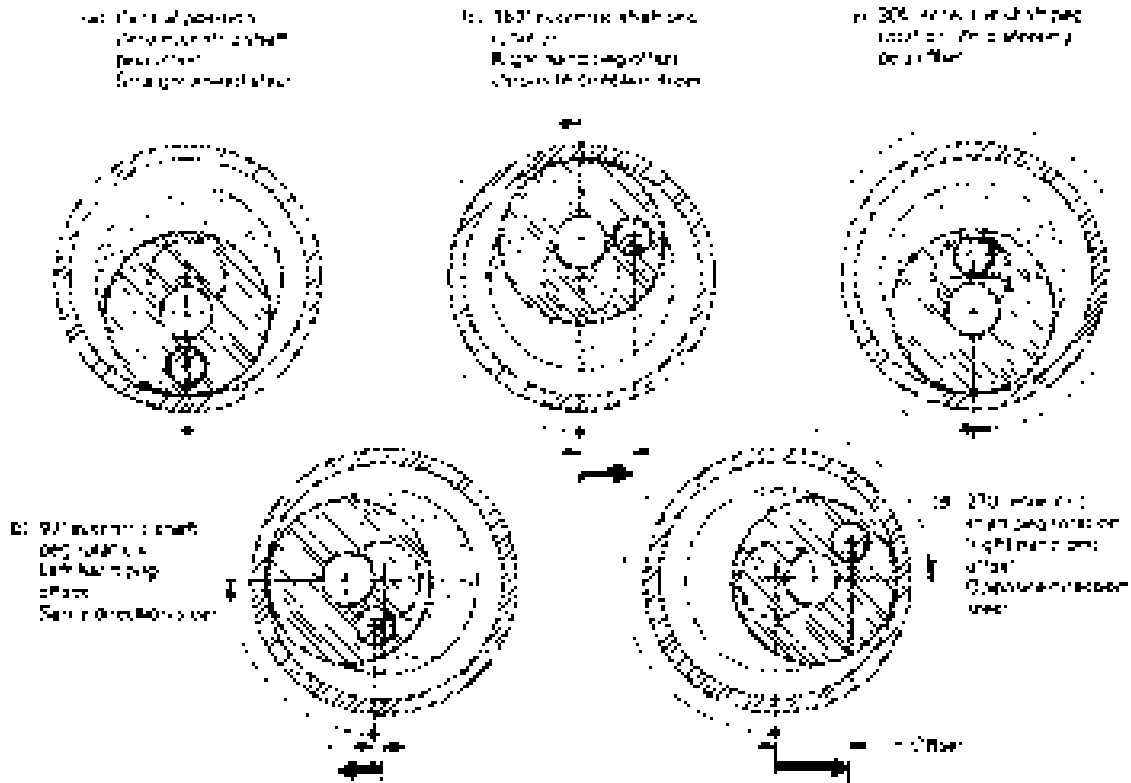
Fig. 9.36 (a–d) Principle of rear steering box mechanism

the epicyclic gear set does not operate in the fourth quadrant even under full steering lock conditions.

9.5 Variable-ratio rack and pinion
(Fig. 9.37(a–d))
Variable-ratio rack and pinion can be made to improve both manual and power assisted steering operating characteristics. For a manual rack and pinion steering system it is desirable to have a moderately high steering ratio to provide an almost direct steering response while the steering wheel is in the normally 'central position' for straight ahead driving and for very small steering wheel angular correction movement. Conversely for parking manoeuvres requiring a greater force to turn the steering wheel on either lock, a more indirect lower steering ratio is called for to reduce the steering wheel turning effort. However, with power assisted steering the situation is different; the steering wheel response in the straight ahead driving position still needs to be very slightly indirect with a relatively high steering ratio, but with the power assistance provided the off-centre steering response for

manoeuvring the vehicle can be made more direct compared with a manual steering with a slightly higher steering ratio. The use of a more direct low steering ratio when the road wheels are being turned on either lock is made possible by the servo action of the hydraulic operated power cylinder and piston which can easily overcome the extra tyre scrub and swivel-pin inclination resisting force. The variable-ratio rack is achieved by having tooth profiles of different inclination along the length of the rack, accordingly the pitch of the teeth will also vary over the tooth span.

With racks designed for manual steering the centre region of the rack has wide pitched teeth with a 40° flank inclination, whereas the teeth on either side of the centre region of the rack have a closer pitch with a 20° flank inclination. Conversely, power assisted steering with variable-ratio rack and pinion (see Fig. 9.37(c)) has narrow pitch teeth with 20° flank inclination in the central region; the tooth profile then changes to a wider pitch with 40° flank inclination away from the central region of the rack for both steering locks.

(a) Central rack teeth   (b) Off-centre rack teeth

Pressure angle 20°

Pressure angle 40°

(c) Variable-ratio tooth rack

Wide pitch (P)   Narrow pitch (p)   Wide pitch (P)
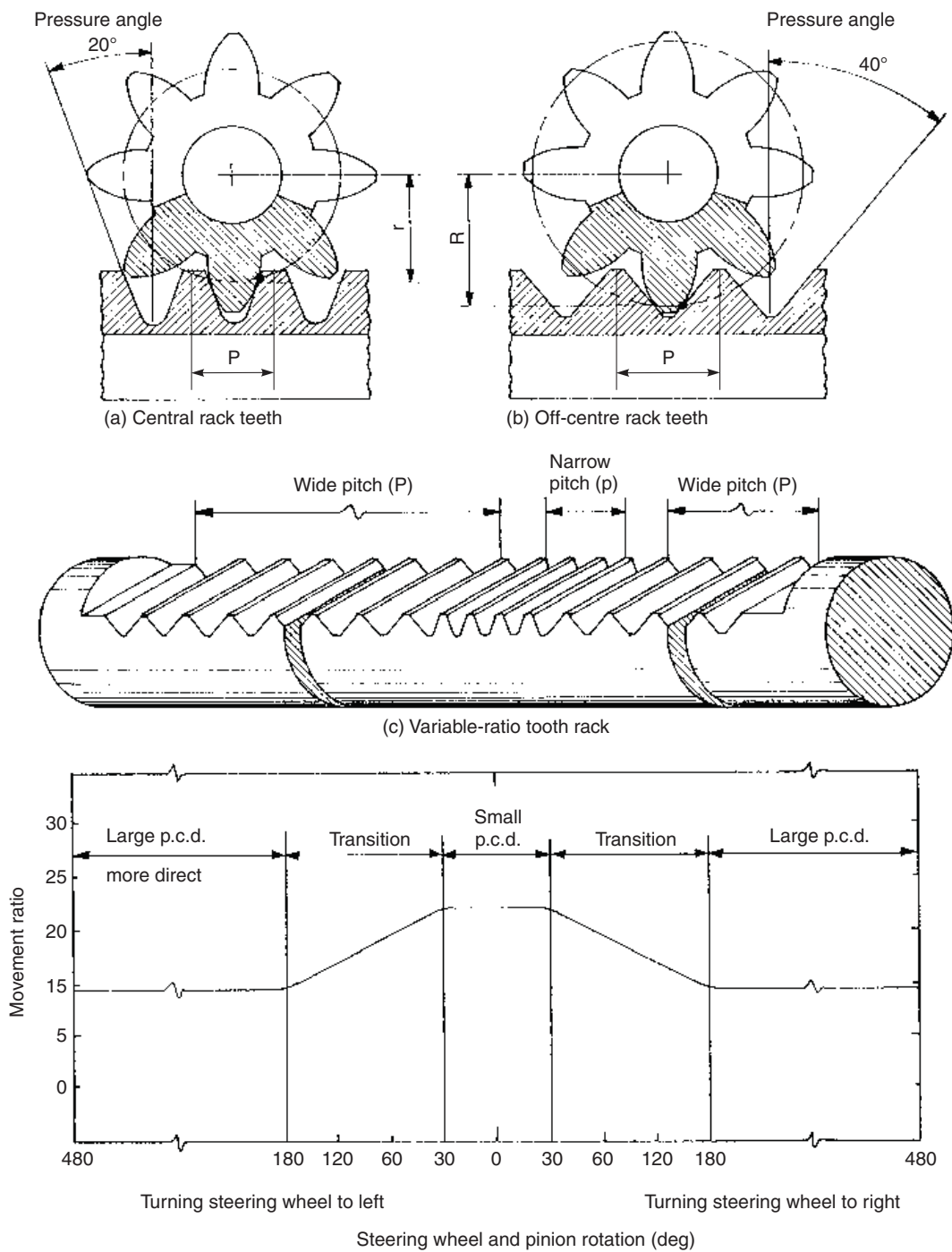
(d) Rack and pinion movement ratio from lock to lock of the steering wheel

Movement ratio

Large p.c.d. more direct   Transition   Small p.c.d.   Transition   Large p.c.d.

Turning steering wheel to left   Turning steering wheel to right

Steering wheel and pinion rotation (deg)

Fig. 9.37 (a–d)   Variable ratio rack and pinion steering suitable for power assisted steering

With variable-ratio rack and pinion involute teeth the rack has straight sided teeth. The sides of the teeth are normal to the line of action, therefore, they are inclined to the vertical at the pressure angle. If the rack has narrow pitch 'p' 20° pressure-angle teeth, the pitch circle diameter (2R) of the pinion will be small, that is, the point of contact of the meshing teeth will be close to the tip of the rack teeth (Fig. 9.37(a)), whereas with wide pitched 'P' 40° pressure-angle tooth contact between teeth will be near the root of the rack teeth (Fig. 9.37(b)) so its pitch circle diameter (2R) will be larger.

The ratio of steering wheel radius to pinion pitch circle radius (tooth contact radius) determines the movement ratio. Thus the smaller the pitch circle radius of the pinion for a given steering wheel size, the greater will be the movement ratio (see Fig. 9.37(d)), that is, a smaller input effort will be needed to steer the vehicle, but inversely, greater will be the steering wheel movement relative to the vehicle road wheel steer angle.

This design of rack and pinion tooth profile can provide a movement-ratio variation of up to 35% with the number of steering wheel turns limited to 2.8 from lock to lock.

9.6 Speed sensitive rack and pinion power assisted steering

9.6.1 Steering desirability

To meet all the steering requirements the rack and pinion steering must be precise and direct under normal driving conditions, to provide a sense of feel at the steering wheel and for the steering wheel to freely return to the straight ahead position after the steering has been turned to one lock or the other. The conventional power assisted steering does not take into account the effort needed to perform a steering function relative to the vehicle speed, particularly it does not allow for the extra effort needed to turn the road wheels when manoeuvring the vehicle for parking.

The ' F Servotronic' power assisted steering is designed to respond to vehicle speed requirements, 'not engine speed', thus it provides more steering assistance when the vehicle is at a standstill or moving very slowly than when travelling at speed; at high speed the amount of steering assistance may be tuned to be minimal, so that the steering becomes almost direct as with a conventional manual steering system.

9.6.2 Design and construction (Fig. 9.38(a–d))

The ' F Servotronic' speed-sensitive power assisted steering uses a conventional rotary control valve, with the addition of a reaction-piston device which modifies the servo assistance to match the driving mode.

The piston and rotary control valve assembly comprises a pinion shaft, valve rotor shaft with six external longitudinal groove slots, valve sleeve with six matching internal longitudinal groove slots, torsion bar, reaction-piston device and an electro-hydraulic transducer. The reaction-piston device is supported between the rotary valve rotor and valve sleeve, and guided internally by the valve rotor via three axially arranged ball grooves and externally guided by the valve sleeve through a multi-ball helix thread.

The function of the reaction-piston device is to modify the fluid flow gap formed between the valve rotor and sleeve longitudinal groove control edges for different vehicle driving conditions.

An electronic control unit microprocessor takes in speed frequency signals from the electronic speedometer, this information is then continuously evaluated, computed and converted to an output signal which is then transmitted to the hydraulic transducer mounted on the rotary control valve casing. The purpose of this transducer is to control the amount of hydraulic pressure reaching the reaction-piston device based on the information supplied to the electronic control unit.

9.6.3 Operation of the rotary control valve and power cylinder

eutral position (Figs 9.38(a) and 9.39(a)) With the steering wheel in its central free position, pressurized fluid from the pump enters the valve sleeve, passes though the gaps formed between the longitudinal groove control edges of both sleeve and rotor, then passes to both sides of the power cylinder. At the same time fluid will be expelled via corresponding exit 'sleeve/rotor groove' control-edge gaps to return to the reservoir. The circulation of the majority of fluid from the pump to the reservoir via the control valve prevents any build-up of fluid pressure in the divided power cylinder, and the equalization of the existing pressure on both sides of the power piston neutralizes any 'servo' action.

Anticlockwise rotation of the steering wheel (turning left — low speed) (Figs 9.38(b) and 9.39(b)) Rotating

Rack

Pinion shaft

Reservoir

Pump

Valve sleeve

Inner check valve

Outer check valve

Inner reaction chamber

Reaction piston (RP)

Outer reaction chamber

Torsion bar

(4)

(1)

6

5

7

Valve rotor shaft

Outer orifice

Inner orifice

Cut-off valve (CO-V)

(2)

6

(3)

Right hand side

Electro-hydraulic transducer (EHT)

Teflon ring seal

Electronic speedometer

Electronic control unit (ECU)

Left hand side

Power piston

Power cylinder

(a) Neutral position

Fig. 9.38 (a–d)   Speed sensitive rack and pinion power assisted steering with rotary reaction control valve

355
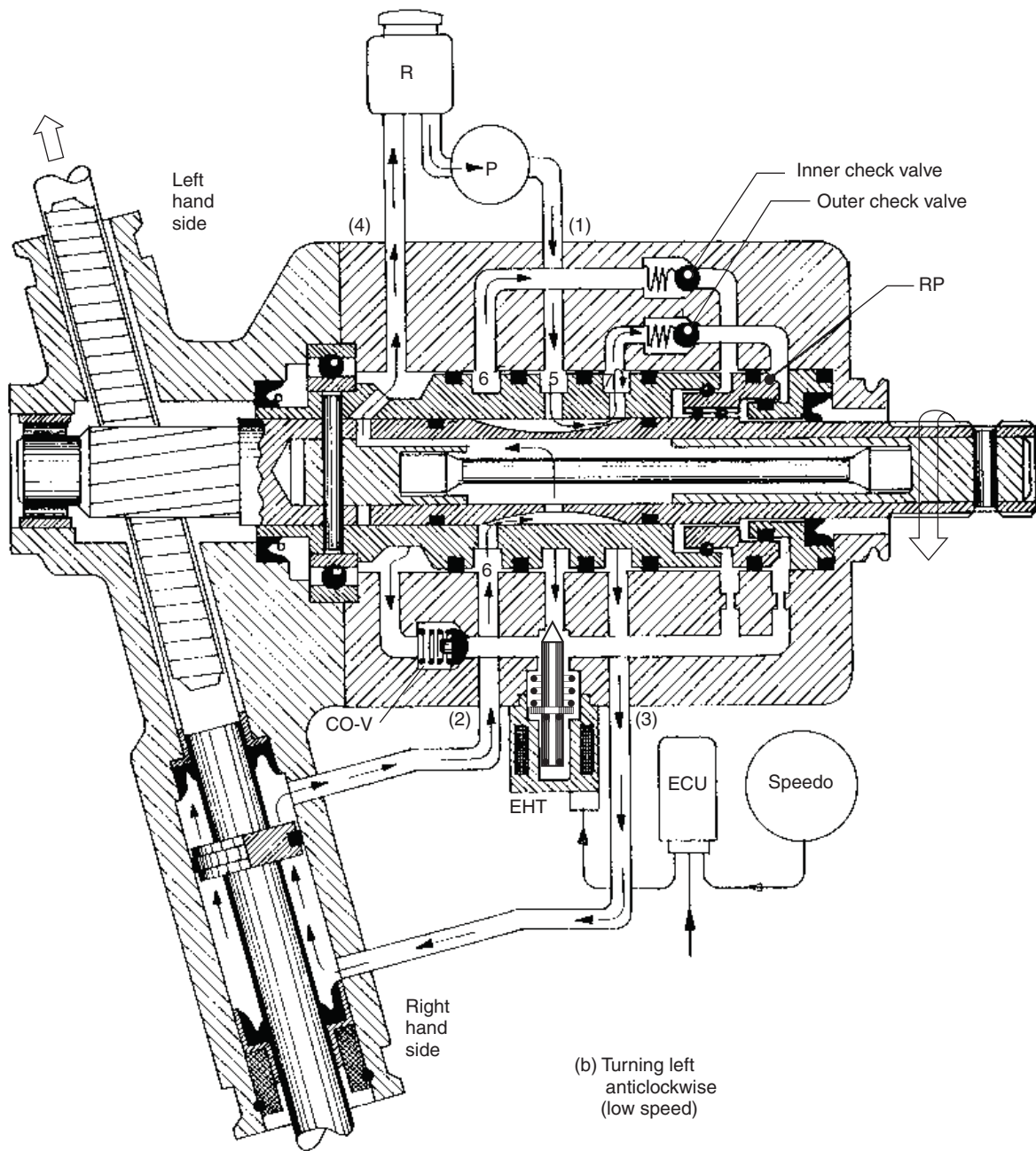
Fig. 9.38 *contd*

the steering wheel in an anticlockwise direction twists the control valve rotor against the resistance of the torsion bar until the corresponding leading edges of the elongated groove in the valve rotor and sleeve align. At this point the return path to the exit port '4' is blocked by control edges '2' while fluid from the pump enters port '1'; it then passes in between the enlarged control-edge gaps to come out of port '3', and finally it flows into the right-hand power cylinder chamber.

Left
hand
side

Inner check valve

Outer check valve

RP

(4)

R

P

(1)

6   5

6

CO-V

(2)

EHT

(3)

ECU

Speedo

Right
hand
side

Ball
guide
grooves

Ball
thread
grooves

Reaction
piston
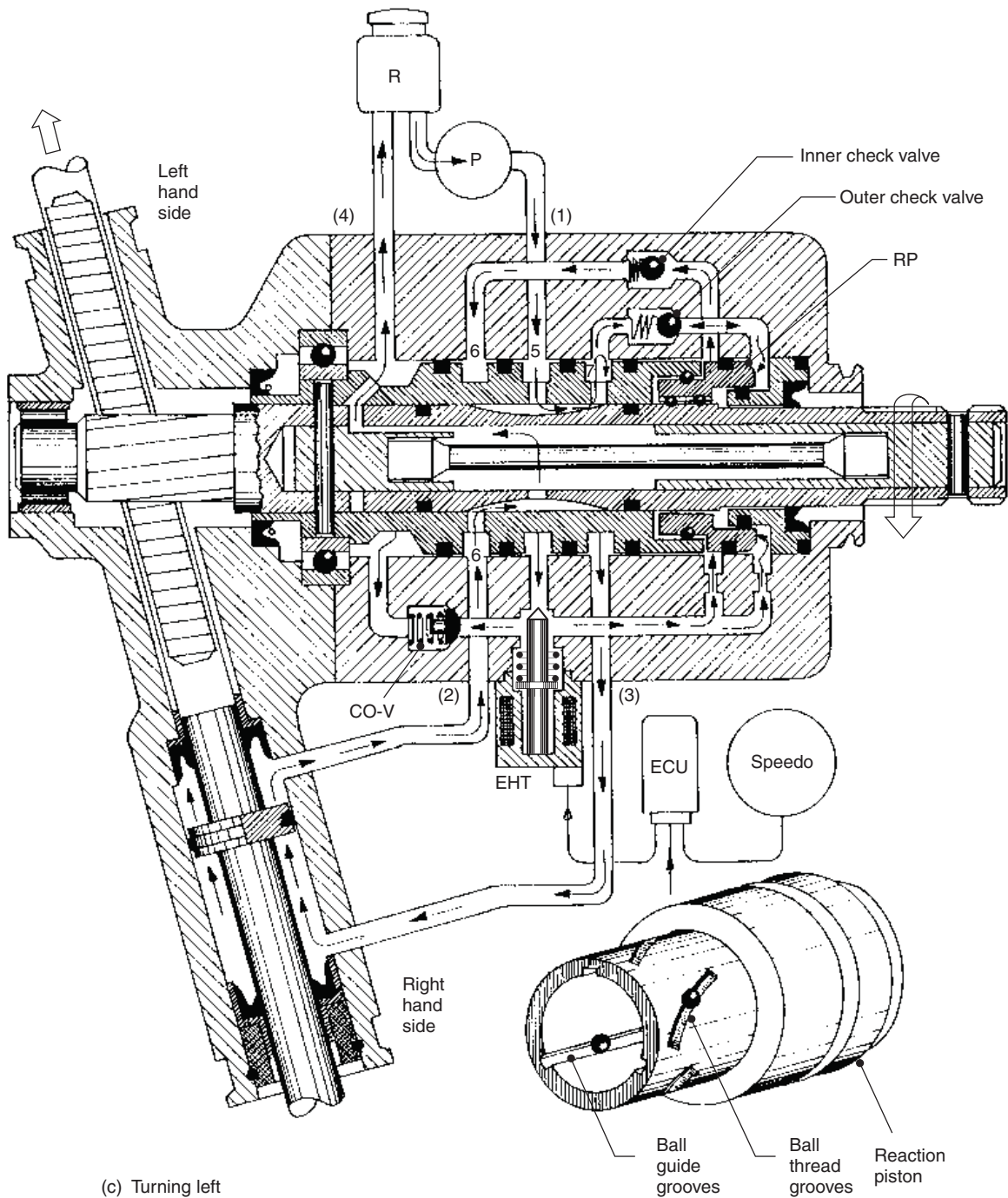
(c) Turning left
anticlockwise
(high speed)

Fig. 9.38   *contd*

357
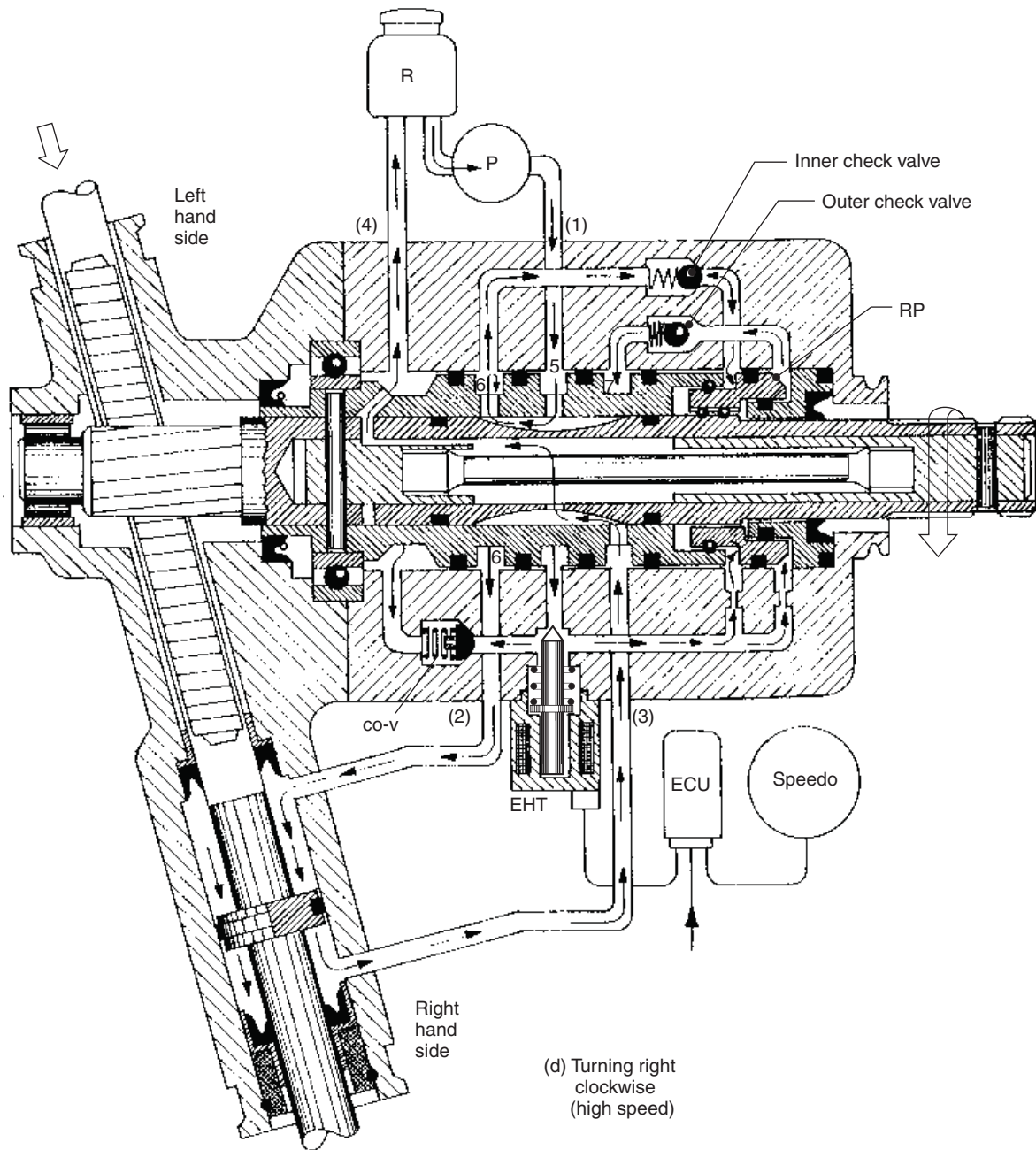
Fig. 9.38 *contd*

Conversely fluid from the left hand side power cylinder chamber is pushed towards port '2' where it is expelled via the enlarged trailing control-edge gap to the exit port '4', then is returned to the reservoir. The greater the effort by the driver to turn the steering wheel, the larger will be the control-edge gap made between the valve sleeve and rotor and greater will be the pressure imposed on the right hand side of the power piston.

Return long slot

Sleeve

Rotor

Torsion bar

Supply short slot

Reservoir

Pump

(4)

(4)

(2)

(1)

(3)

Left hand

Right hand

Power cylinder and piston
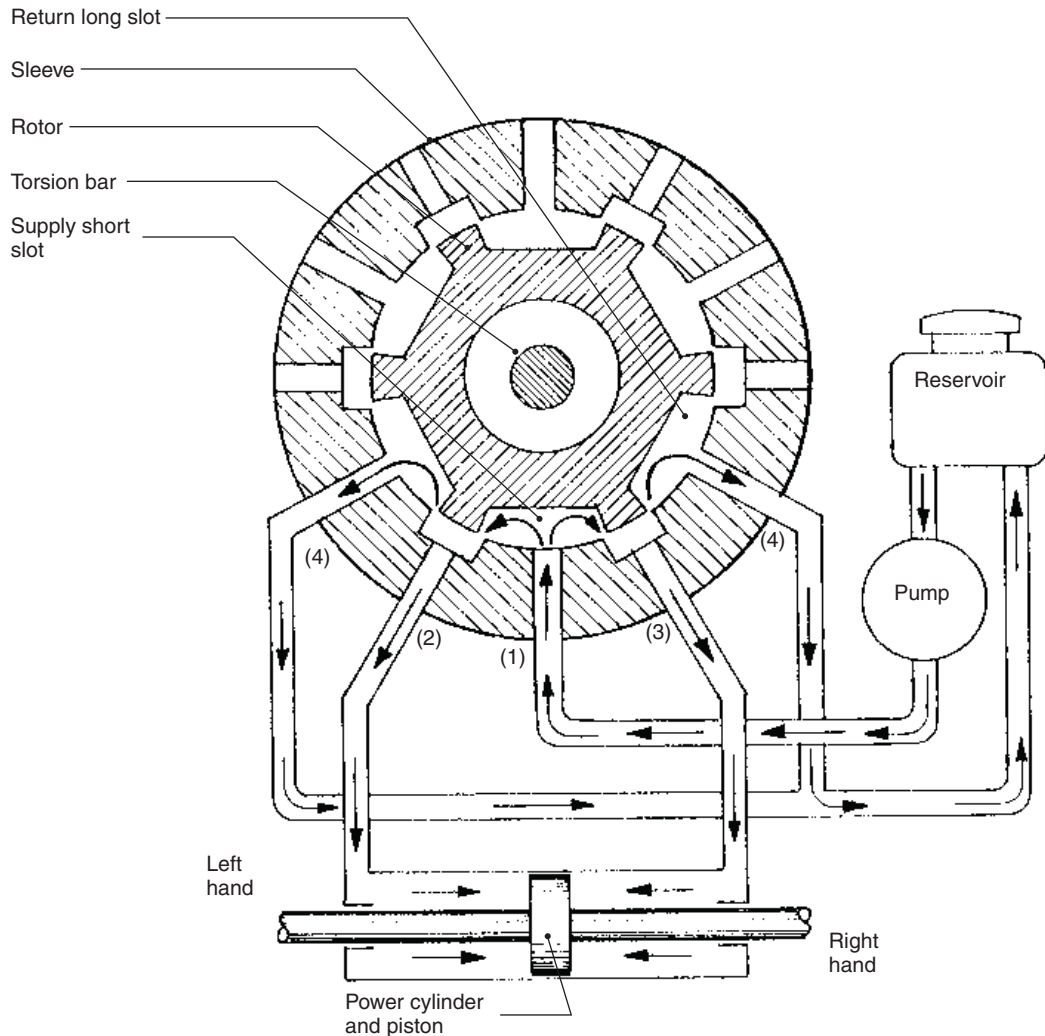
(a) Neutral position

Fig. 9.39 (a–c)   Rack and pinion power assisted steering sectional end views of rotary reaction control valve

When the vehicle is stationary or moving very slowly and the steering wheel is turned to man- oeuvre it into a parking space or to pull out from a kerb, the electronic speedometer sends out its minimal frequency signal to the electronic control unit. This signal is processed and a corresponding control current is transmitted to the electro- hydraulic transducer. With very little vehicle move- ment, the control current will be at its maximum; this closes the transducer valve thus preventing fluid pressure from the pump reaching the reaction valve piston device and for fluid flowing to and through the cut-off valve. In effect, the speed sen- sitive rotary control valve under these conditions now acts similarly to the conventional power assisted steering; using only the basic rotary con- trol valve, it therefore is able to exert relatively more servo assistance.

Anticlockwise rotation of the steering wheel (turning left — high speed) (Figs 9.38(c) and 9.39(b))   With increasing vehicle speed the frequency of the elec- tronic speedometer signal is received by the electro- nic control unit; it is then processed and converted to a control current and relayed to the electro- hydraulic transducer. The magnitude of this con- trol current decreases with rising vehicle speed,

359

Sleeve

Rotor

Torsion bar

Supply short slot

Return long slot

(4)   (4)

R

P

(2) (1)   (3)

Left
hand

(b) Turning left – anticlockwise
rotation of the steering
wheel
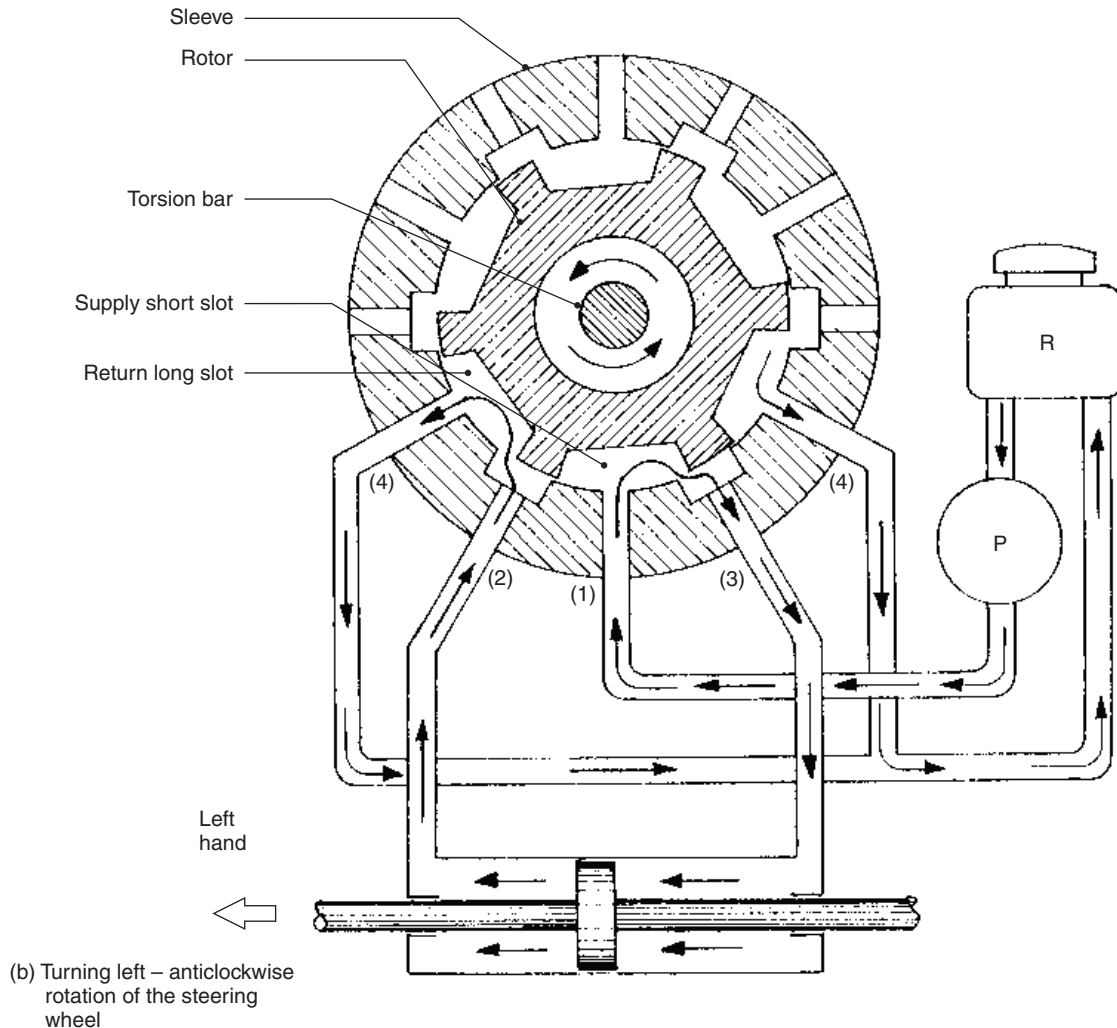
Fig. 9.39 *contd*

correspondingly the electro-hydraulic transducer valve progressively opens thus permitting fluid to reach the reaction piston at a pressure determined by the transducer-valve orifice opening. If the steering wheel is turned anticlockwise to the left (Fig 3.38(c)), the fluid from the pump enters radial groove '5', passes along the upper longitudinal groove to radial groove '7', where it circulates and comes out at port '3' to supply the right hand side of the power cylinder chamber with fluid.

Conversely, to allow the right hand side cylinder chamber to expand, fluid will be pushed out from the left hand side cylinder chamber; it then enters port '2' and radial groove '6', passing through the lower longitudinal groove and hollow core of the

rotor valve, finally returning to the reservoir via port '4'. Fluid under pressure also flows from radial groove '7' to the outer chamber check valve to hold the ball valve firmly on its seat. With the electro-hydraulic transducer open fluid under pump pressure will now flow from radial grooves '5' to the inner and outer reaction-piston device orifices. Fluid passing though the inner orifice circulates around the reaction piston and then passes to the inner reaction chamber check valve where it pushes the ball off its seat. Fluid then escapes through this open check valve back to the reservoir by way of the radial groove '6' through the centre of the valve rotor and out via port '4'. At the same time fluid flows to the outer piston

360
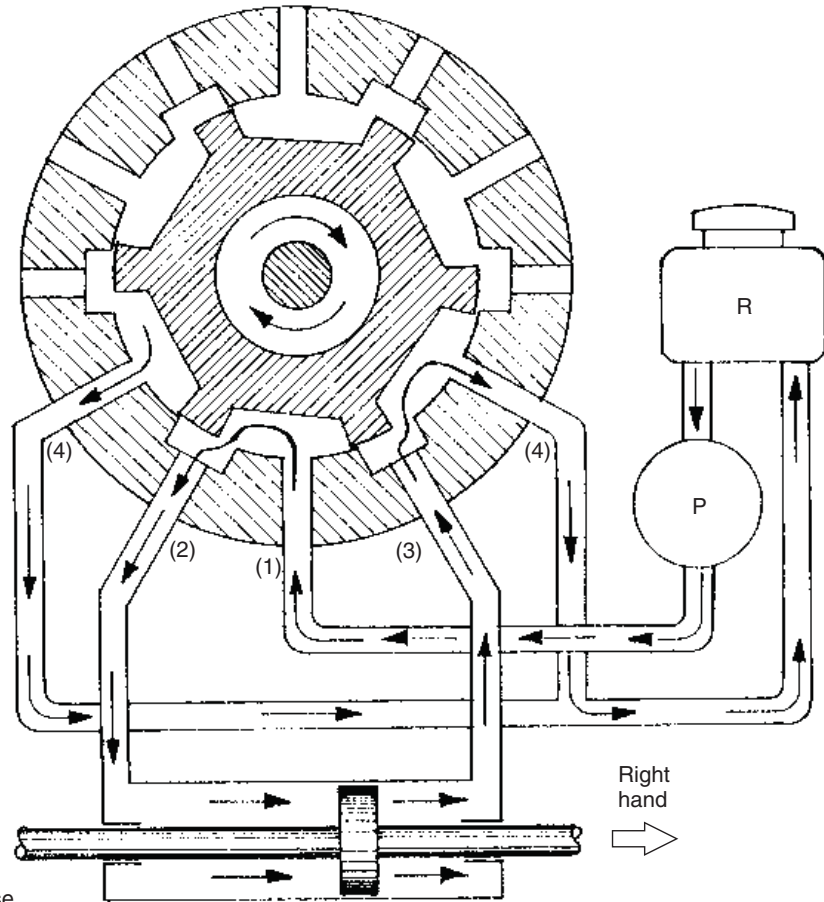
(c) Turning right – clockwise
   rotation of the steering
   wheel

Fig. 9.39 *contd*

reaction chamber and to the right hand side of
the outer check valve via the outer orifice, but
slightly higher fluid pressure from port '7' acting
on the opposite side of the outer check valve pre-
vents the valve opening. However, the fluid pres-
sure build-up in the outer piston reaction chamber
will tend to push the reaction piston to the left hand
side, consequently due to the pitch of the ball-
groove helix, there will be a clockwise opposing
twist of the reaction piston which will be trans-
mitted to the valve rotor shaft. Accordingly this
reaction counter twist will tend to reduce the fluid
gap made between the valve sleeve and rotor long-
itudinal control edges; it therefore brings about a
corresponding reaction in terms of fluid pressure

reaching the left hand side of the power piston and
likewise the amount of servo assistance.

In the high speed driving range the electro-
hydraulic transducer control current will be very
small or even nil; it therefore causes the transducer
valve to be fully open so that maximum fluid pres-
sure will be applied to the outer reaction piston.
The resulting axial movement of the reaction pis-
ton will cause fluid to be displaced from the inner
reaction chamber through the open inner reaction
chamber check valve, to the reservoir via the radial
groove '6', lower longitudinal groove, hollow rotor
and finally the exit port '4'.

As a precaution to overloading the power steer-
ing, when the reaction piston fluid pressure reaches

its pre-determined upper limit, the cut-off valve opens to relieve the pressure and to return surplus fluid to the reservoir.

Clockwise rotation of the steering wheel (turning right — low speed) (Fig. 9.39(c)) Rotation of the steering wheel clockwise twists the control valve against the resistance of the torsion bar until the corresponding leading control edges of the elongated grooves in the valve rotor and sleeve are aligned. When the leading groove control edges align, the return path to the exit port '3' is blocked while fluid from the pump enters port '1'; it then passes inbetween the enlarged control-edge gap to come out of port '2' and finally flows into the left hand power cylinder chamber.

Conversely, fluid from the right hand side power cylinder chamber is displaced towards port '3'where it is expelled via the enlarged gap made between the trailing control edges to the exit port '4'; the fluid then returns to the reservoir. The greater the misalignment between the valve sleeve and rotor control edges the greater will be the power assistance.

Clockwise rotation of the steering wheel (turning right — high speed) (Figs 9.38(d) and 9.39(c)) With increased vehicle speed the electro-hydraulic transducer valve commences to open thereby exposing the reaction piston to fluid supply pressure.

If the steering wheel is turned clockwise to the right (Fig. 9.38 (d)), the fluid from the pump enters the radial groove '5', passes along the upper longitudinal grooves to radial groove '6' where it circulates and comes out at port '2' to supply the power cylinder's left hand side chamber with fluid.

Correspondingly fluid will be displaced from the power cylinder's right hand chamber back to the reservoir via port '3' and groove '7', passing through to the lower longitudinal groove and hollow core of the rotor valve to come out at port '4'; from here it is returned to the reservoir.

Fluid under pressure will also flow from radial groove '6' to the reaction piston's outer chamber check valve thereby keeping the ball valve in the closed position. Simultaneously, with the electro-hydraulic transducer open, fluid will flow from radial groove '5' to the inner and outer reaction-piston orifices. Fluid under pressure will also pass though the outer orifice, and circulates around the reaction piston before passing to the reaction piston's outer chamber check valve; since the fluid pressure on the spring side of the check valve ball is much lower, the ball valve is forced to open thus

causing fluid to be returned to the reservoir via the radial groove '7', lower elongated rotor groove, hollow rotor core and out via port '4'. At the same time fluid flows to the inner chamber of the reaction piston via its entrance orifice. Therefore, the pressure on the spring side of its respective ball check valve remains higher thus preventing the ball valve opening. Subsequently pressure builds up in the inner chamber of the reaction piston, and therefore causes the reaction piston to shift to the right hand side; this results in an anticlockwise opposing twist to the reaction piston due to the ball-groove helices. Accordingly the reaction counter twist will reduce the flow gap between corresponding longitudinal grooves' control edges so that a reduced flow will be imposed on the left hand side of the power cylinder. Correspondingly an equal quantity of fluid will be displaced from the reaction piston outer chamber which is then returned to the reservoir via the now open outer check valve. Thus as the electro-hydraulic transducer valve progressively opens with respect to vehicle speed, greater will be the fluid pressure transmitted to the reaction piston inner chamber and greater will be the tendency to reduce the flow gap between the aligned sleeve and rotor valve control edges, hence the corresponding reduction in hydro-servo assistance to the steering.

9.6.4 Characteristics of a speed sensitive power steering system (Fig. 9.40)

Steering input effort characteristics relative to vehicle speed and servo pressure assistance are shown in Fig. 9.40. These characteristics are derived from the microprocessor electronic control unit which receives signals from the electronic speedometer and transmits a corresponding converted electric current to the electro-hydraulic transducer valve attached to the rotary control valve casing. Accordingly, the amount the electro-hydraulic transducer valve opens controls the degree of fluid pressure reaction on the modified rotary control valve (Fig. 9.38(c)). As a result the amount of power assistance given to the steering system at different vehicle speeds can be made to match more closely the driver's input to the vehicle's resistance to steer under varying driving conditions.

Referring to Fig. 9.40 at zero vehicle speed when turning the steering, for as little an input steering wheel torque of 2 Nm, the servo fluid pressure rises to 40 bar and for only a further 1 Nm input rise (3 Nm in total) the actuating pressure can reach 94 bar. For a vehicle speed of 20 km/h the rise in servo pressure is less steep, thus for an input effort torque of 2 Nm the actuating pressure has only risen to
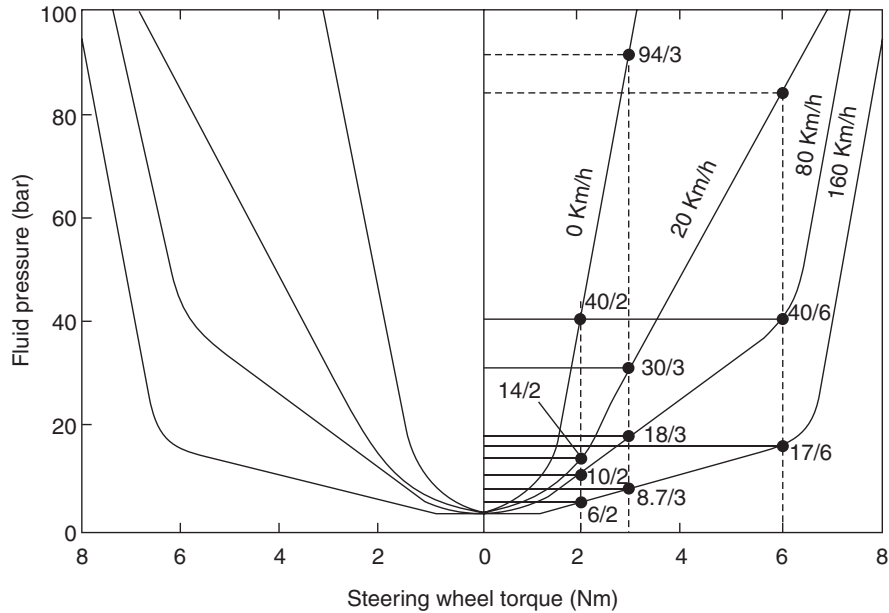
Fig. 9.40 Speed sensitive power steering steering wheel torque to servo fluid pressure characteristics for various road speeds

about 14 bar and for an input of 3 Nm the pressure just reaches 30 bar. With a higher vehicle speed of 80 km/h the servo pressure assistance is even less, only reaching 10, 18 and 40 bar for an input torque of 2, 3 and 6 Nm respectively; however, beyond an input torque of 6 Nm the servo pressure rises very steeply. Similarly for a vehicle speed of 160 km/h the rise in servo pressure assistance for an input torque rise ranging from 2 to 6 Nm only increases from 6 to 17 bar respectively, again beyond this input torque the servo pressure rises extremely rapidly. These characteristics demonstrate that there is considerable servo pressure assistance when manoeuvring the vehicle at a standstill or only moving slowly; conversely there is very little assistance in the medium to upper speed range of a vehicle, in fact the steering is almost operated without assistance unless a very high input torque is applied to the steering wheel in an emergency.

9.7 Rack and pinion electric power assisted steering

The traditional hydraulic actuated power assisted steering requires weighty high pressure equipment, which incorporates an engine driven high pressure pump, fluid reservoir and filter, reaction valve, high pressure hoses, servo cylinder, piston, ram and a suitable fluid. There is a tendency for fluid

to leak due to severe overloading of the steering linkage when driving against and over stone kerbs and when manoeuvring the car during parking in confined spaces. The electric power assisted steering unit is relatively light, compact, reliable and requires a maximum current supply of between 40 and 80 amperes when parking (depending on the weight imposed on the front road wheels) and does not consume engine power as is the case of a hydraulic power assisted steering system which does apply a relatively heavy load on the engine.

9.7.1 Description and construction (Fig. 9.41)
The essentials of a rack and pinion electric power assisted steering comprises an input shaft attached to the steering wheel via an intermediate shaft and universal joint and a integral output shaft and pinion which meshes directly with the steering rack, see Fig. 9.41. A torsion bar mounted in the centre of the hollow input shaft joins the input and output shafts together and transfers the driver's manual effort at the steering wheel to the pinion output shaft. Electrical servo assistance is provided by an electric motor which supplies the majority of the steering torque to the output pinion shaft when the car's steering is being manoeuvred. Torque is transferred from the electric motor to the output pinion shaft through a ball bearing supported worm gear and a worm wheel mounted and attached to the output pinion shaft.
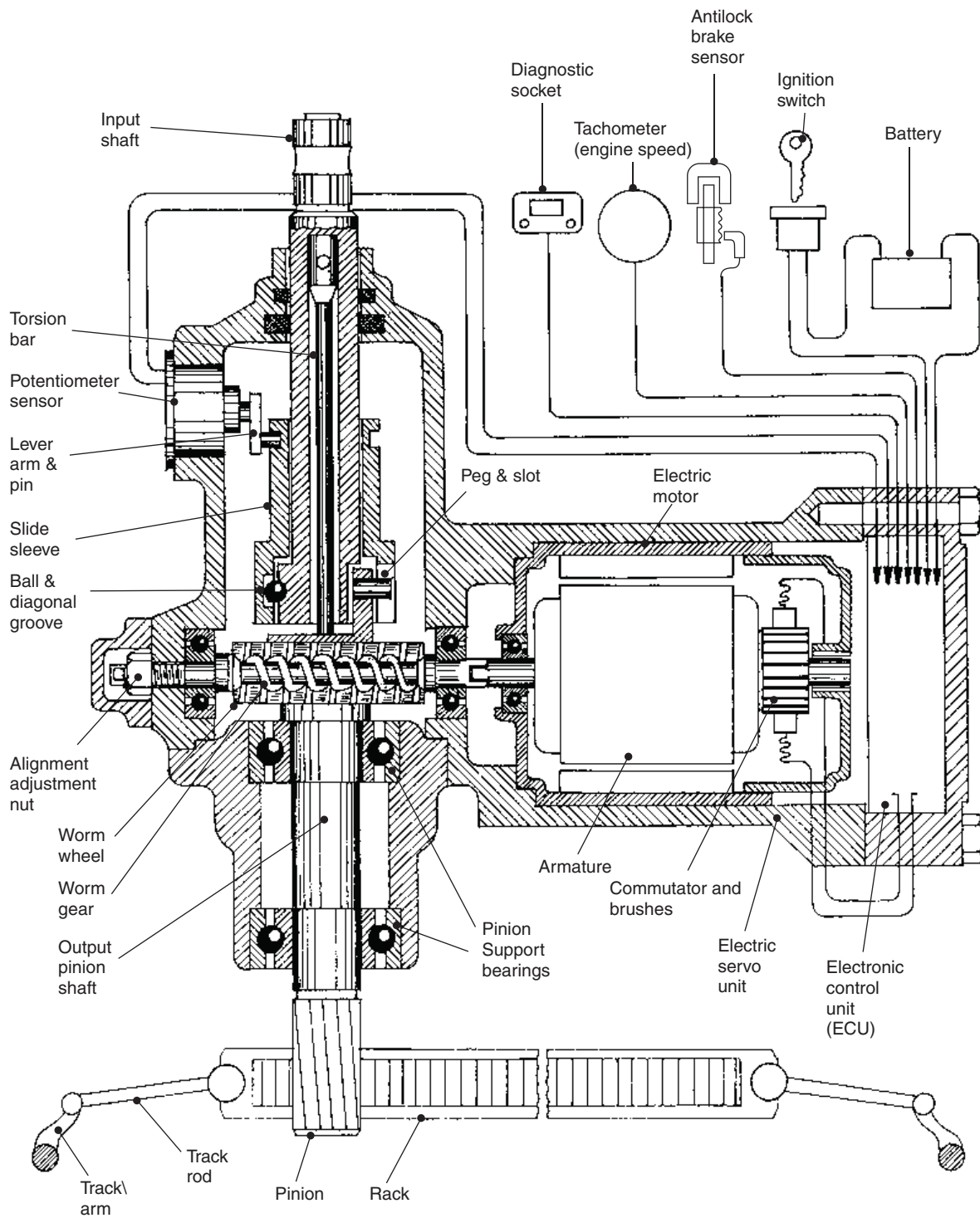
Fig. 9.41 Rack and pinion electric power assisted steering system

Relative angular misalignment between the input and output shafts is measured by transforming this angular movement into an axial linear movement along the input shaft by means of a slide sleeve, control ball, internal diagonal groove and a peg and slot. The slide sleeve which fits over the input shaft can move axially relative to the input shaft and rotates with the output shaft due to the peg and slot. Proportionate axial movement of the slide sleeve to the misalignment of the input to the output shafts is achieved by the internal diagonally formed groove in the slide sleeve and the control ball held in the shoulder part of the input shaft. Any axial slide-sleeve movement is registered by the rotary potentiometer (variable resistor) through the potentiometer arm and pin which is located in the slide sleeve's external groove.

When the steering is initially turned against the tyre to road surface grip resistance, the input torque applied to the steering is transferred to the pinion output shaft through the central torsion bar. The torsional twist of the torsion bar, that is, the angular misalignment of the input and output shafts, is proportional to the input effort at the steering wheel before the servo electric motor responds and supplies the extra input torque to the pinion output shaft to produce the desired amount of steering turn by the front road wheels. Should the electric servo assistance fail for any reason, then the steering input effort will be entirely provided by the driver though the torsion bar; under these conditions however the driver will experience a much heavier steering. A limit to the maximum torsion bar twist is provided when protruding ridges formed on the input and output shafts butt with each other.

An electronic control unit which is a microprocessor takes in information from various electrical sensors and then translates this from a programmed map into the required steering assistance to be delivered by the servo electric motor. Mechanical power is supplied by a servo electric motor which is able to change its polarity so that it can rotate either in a clockwise or anticlockwise direction as commanded by the direction of steering turn, the drive being transferred from the output pinion shaft via a warm gear and warm wheel. The large gear reduction ratio provided with this type of drive gearing enables the warm wheel to rotate at a much reduced speed to that of the warm gear and enables a relatively large torque to be applied to the output pinion shaft with a moderately small electric motor.

Steering wheel torque is monitored in terms of relative angular misalignment of the input and output shafts by the slide-sleeve movement, this is then converted into an electrical signal via the interlinked rotary potentiometer sensor. Engine and road speed sensors enable the electronic control unit to provide speed-sensitive assistance by providing more assistance at low vehicle speed when manoeuvring in a restricted space and to reduce this assistance progressively with rising speed so that the driver experiences a positive feel to the steering wheel. Note the engine and vehicle speeds are monitored by the tachometer and anti-lock brake sensors respectively.

9.7.2 *Operating principle* (Figs 9.42(a–c))

eutral position (Fig. 9.42(b)) When the input and output shafts are aligned as when the steering wheel is in a neutral no turning effort position, the control ball will be in the central position of the diagonal control groove. Correspondingly the potentiometer lever arm will be in the horizontal position, with zero signal feed current to the electronic control unit and the power supply from the electronic control unit to the servo electric motor switched off. Note the potentiometer is calibrated with the wiper arm in its mid-track position to signal a zero feed current.

Clockwise right hand turn (Fig. 9.42(a)) When the steering wheel is turned clockwise to give a right hand turn, the input torque applied by the steering wheel causes a relative angular misalignment between the input and output shaft, this being proportional to the degree of effort the driver applies. As a result the control ball rotates clockwise with the input shaft relative to the output shaft, and since the slide sleeve cannot rotate independently to the output pinion shaft due to the peg and slot, the flanks of the diagonal groove are compelled to slide past the stationary control ball, thus constraining the slide sleeve to an axial upward movement only.

Accordingly the rotary potentiometer lever arm will twist anticlockwise thereby causing the wiper arm to brush over the wire or ceramic resistive track. The change in resistance and current flow signals to the electronic control unit that servo assistance is required, being in proportion to the amount the slide sleeve and rotary potentiometer moves. Once the initial effort at the steering wheel has been applied the torsional twist of the torsion bar relaxes; this reduces the relative misalignment of the input and output shafts so that the rotary potentiometer lever arm moves to a reduced feed
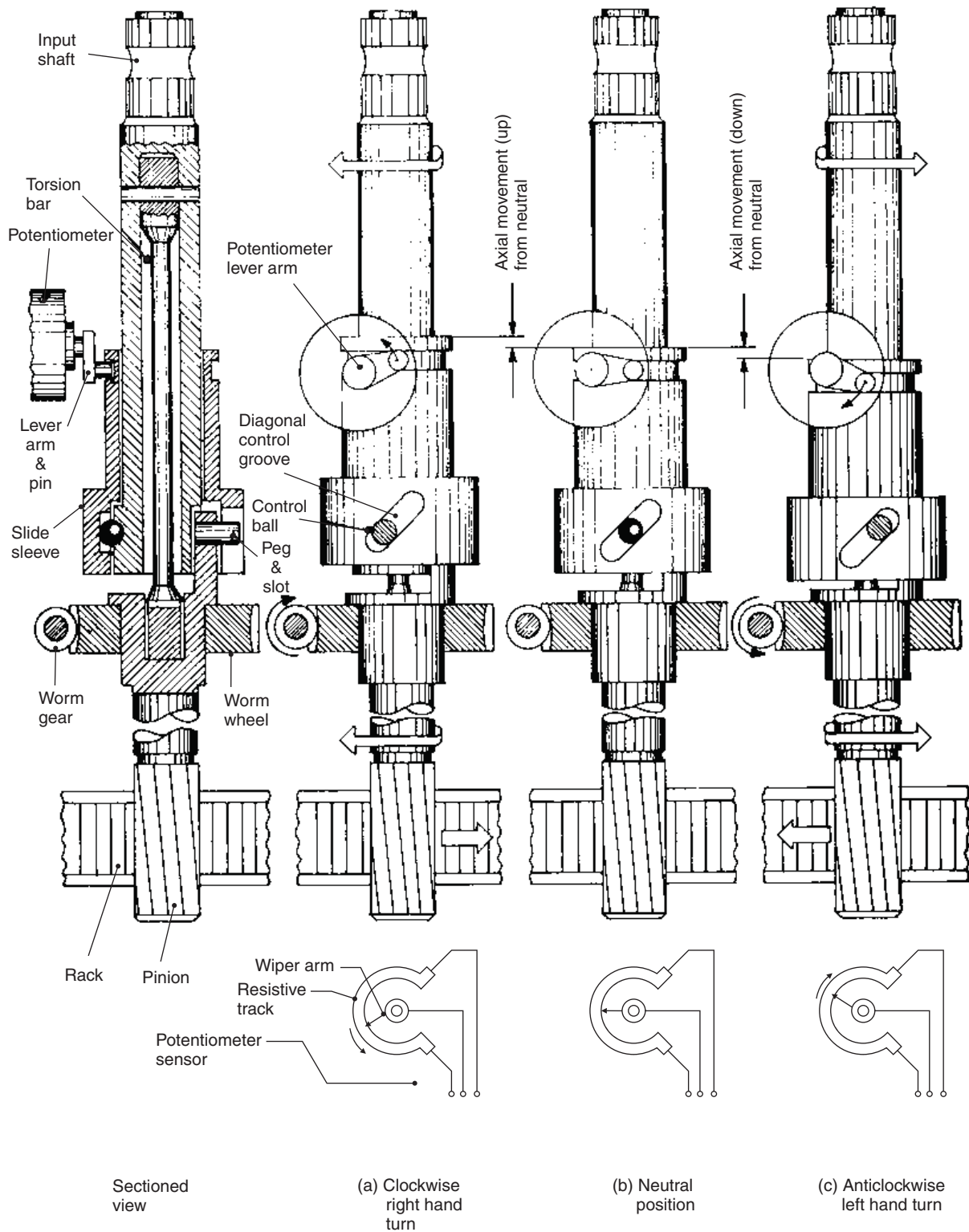
Fig. 9.42 (a–c) Operating principles for a rack and pinion electric power assisted steering

366

current position or even to zero feed current position. At this point the electronic control unit switched 'off' the electrical supply to the servo electric motor so that servo assistance via the warm gear and warm wheel to output pinion shaft comes to an abrupt end.

Anticlockwise left hand turn (Fig. 9.42(c)) When the steering wheel is turned anticlockwise to negotiate a left hand turn, the input effect applied by the driver to the steering wheel causes a relative angular misalignment between the input and output shafts, the relative twist of the torsion bar being proportional to the driver's input effort on the steering wheel. Due to the rotary movement of the input shaft, and control ball relative to the pinion output shaft, the diagonal groove in the sleeve will be forced to move over the stationary control ball in a downward axial direction since the peg and slot only permits the slide sleeve to move axially. The vertical downward displacement of the sleeve is relayed to the rotary potentiometer lever arm which will now partially rotate in a clockwise direction; its wiper arm will therefore brush over the resistive track, and an appropriate signal current will then be fed to the electronic control unit. The servo electric motor is then switched on, and thereby rotates the worm gear and in turn the worm wheel but at much reduced speed (due to the very large gear reduction ratio provided by a worm gear and worm wheel) in an anticlockwise direction. As the input torque effort by the driver on the steering wheel is reduced almost to nil, the relative misalignment of the input and output shaft will likewise be reduced; correspondingly the rotary potentiometer wiper arm will move to its mid-resistance position signalling zero current feed to the electronic control unit; it therefore switches off and stops the servo electric motor.

10 Suspension

10.1 Suspension geometry

The stability and effective handling of a vehicle depends upon the designers' selection of the optimum steering and suspension geometry which particularly includes the wheel camber, castor and kingpin inclination. It is essential for the suspension members to maintain these settings throughout their service life.

Unfortunately, the pivoting and swivelling joints of the suspension system are subject to both wear and damage and therefore must be checked periodically. With the understanding of the principles of the suspension geometry and their measurements it is possible to diagnose and rectify steering and suspension faults. Consideration will be given to the terminology and fundamentals of suspension construction and design.

10.1.1 Suspension terminology

Swivel oints or king pins   These are the points about which the steering wheel stub axles pivot.

ivot centre   The point where the swivel ball joint axis or kingpin axis projects and intersects the ground.

Contact patch   This is the flattened crown area of a tyre which contacts the ground.

Contact centres   This is the tyre contact patch central point which is in contact with the ground.

Track   This is the transverse distance between both steering wheel contact centres.

10.1.2   *heel camber angle* (Figs 10.1 and 10.2)

Wheel camber is the lateral tilt or sideway inclination of the wheel relative to the vertical (Fig. 10.1). When the top of the wheel leans inwards towards the body the camber is said to be negative, conversely an outward leaning wheel has positive camber.

Road wheels were originally positively cambered to maintain the wheel perpendicular to the early highly cambered roads (Fig. 10.2) and so shaped as to facilitate the drainage of rain water. With modern underground drainage, road camber has been greatly reduced or even eliminated and therefore wheel camber has been reduced to something like to 1 degrees.

The axis of rotation of a cambered wheel if projected outwards will intersect the ground at the apex of a cone generated if the wheel was permitted to roll freely for one revolution. The wheel itself then resembles the frustrum of a cone (Fig. 10.1). The path taken by the cambered wheel (frustrum of a cone) if free to roll would be a circle about the apex. Consequently both front wheels will tend to steer outwards in opposite directions as the vehicle moves forwards. In practice, the track rods and ball joints are therefore preloaded as they restrain the wheels from swivelling away from each other when the vehicle is in motion. If both wheels have similar camber angles, their outward pull on the track rods will be equal and therefore balance out. If one wheel is slightly more cambered than the other, due maybe to body roll with independent suspension or because of misalignment, the steering wheels will tend to wander or pull to one side as the vehicle is steered in the straight ahead position.
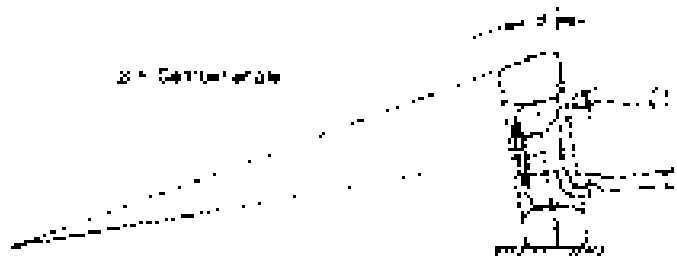
Fig. 10.1   Wheel camber geometry

Fig. 10.2 Road camber

A negatively cambered wheel leaning towards the radius of a curved track or bend increases its cornering power and reduces the tyre contact patch slip angle for a given cornering force compared to a wheel rolling in an upright position. Conversely, a positively cambered wheel leaning away from the centre of rotation reduces its cornering power and increases the tyre slip angle for a similar cornering force compared to a wheel rolling perpendicular to the ground.

To provide a small amount of understeer, the front wheels are normally made to generate a greater slip angle than the rear wheels by introducing positive wheel camber on the front wheels and maintaining the rear wheels virtually perpendicular to the ground.

When cornering with positive camber angles on both front wheels, the inner and outer wheels will lean inwards and outwards respectively relative to the centre of rotation of the turn. At the same time, body roll transfers weight from the inner wheel to the outer one. As a result the inner wheel will generate less slip angle than the outer wheel because it provides an inward leaning, more effective tyre grip with less vertical load than that of the less effective outward leaning tyre, which supports a greater proportion of the vehicle's weight. The front cambered tyres will generate on average more slip angle than the upright rear wheels and this causes the vehicle to have an understeer cornering tendency.

Steered positive cambered wheels develop slightly more slip angle than uncambered wheels. When they are subjected to sudden crosswinds or irregular road ridging, the tyres do not instantly deviate from their steered path, with the result that a more stable steering is achieved.

With the adoption of wider tyres as standard on some cars, wheel camber has to be kept to a minimum to avoid excessive edge wear on the tyres unless the suspension has been designed to cope with the new generation of low profile wide tread width tyre.

10.1.3 *Swivel or kingpin inclination*
(Figs 10.3–10.7)

Swivel pin or kingpin inclination is the lateral inward tilt (inclination) from the top between the upper and lower swivel ball joints or the kingpin to the vertical (Fig. 10.3). If the swivel ball or pin axis is vertical (perpendicular) to the ground, its contact centre on the ground would be offset to the centre of the tyre contact patch (Fig. 10.4). The offset between the pivot centre and contact patch centre is equal to the radius (known as the scrub radius) of a semicircular path followed by the rolling wheels when being turned about their pivots. When turning the steering the offset scrub produces a torque T created by the product of the offset radius r and the opposing horizontal ground reaction force F (i e T Fr (Nm)). A large pivot to wheel contact centre offset requires a big input torque to overcome the opposing ground reaction, therefore the steering will tend to be heavy. No offset (zero offset radius) (Fig. 10.5) prevents the tread rolling and instead causes it to scrub as the wheel is steered so that at low speed the steering also has a heavy response. A compromise is usually made by offsetting the pivot and contact wheel centres to roughly 10–25% of the tread width for a standard sized tyre. This small offset permits the pivot axis to remain within the contact patch, thereby enabling a rolling movement to still take place when the wheels are pivoted so that tyre scruff and creep (slippage) are minimized. One other

Fig. 10.3 Swivel (king) pin inclination

Fig. 10.4 Swivel (king) pin vertical axis offset

Fig. 10.5 Dished wheel centre point steering

effect of a large pivot to contact centre offset is when one of the wheels hits an obstacle like a bump or pothole in the road; a large opposing twisting force would be created momentarily which would be relayed back to the driver's steering wheel in a twitching fashion.

To reduce or even eliminate pivot to wheel centre offset, the whole stub axle, hub bearing assembly and disc or drum would have to be positioned within the centre region of the wheel rim and also extend, and therefore protrude, beyond the wheel rim flange (Fig. 10.5). A dished wheel arrangement of this type is known as centre point steering because both pivot centre and contact patch centres coincide in the middle of the wheel.

The alternative and realistic way of reducing the pivot to contact patch centre offset is to laterally incline the axis of the swivel joints so that the whole hub assembly and disc or drum is positioned inside the wheel and only the upper swivel joint may protrude outside the wheel rim.

The consequences of tilting the swivel pin axis is the proportional lowering of the stub axle axis in the horizontal plane as the wheel assembly swivels about its pivot points relative to the straight ahead position (Fig. 10.6(a and b)). Because the road wheels are already supported at ground level, the reverse happens, that is, both upper and lower wishbone arms or axle beam which supports the vehicle body are slightly raised. This unstable state produces a downward vehicle weight component which tends to return both steered wheel assemblies to a more stable straight ahead position. In other words, the pivot inclination produces a

self-centring action which is independent of vehicle speed or traction but is dependent upon the weight concentration on the swivel joints and their inclination. A very large swivel ball or pin inclination produces an excessively strong self-centring effect which tends to kick back on turns so that the swivel ball or pin inclination angle is usually set between 5 and 15°. A typical and popular value would be something like 8 or 12°.

The combination of both camber and swivel joint inclination is known as the *included angle* and the intersection of both of these axes at one point at ground level classifies this geometry as *centre point steering* (Fig. 10.7). In practice, these centre lines projected through the ball joints or pins and through the centre of the wheel are made to meet at some point below ground level. Thus an offset exists between the projected lines at ground level, which produces a small twisting movement when the wheels are steered. As a result, the wheels tend to roll about a circular path with the offset as its radius, rather than twist about its swivel centre with a continuous slip-grip action which occurs when there is no offset as with the centre point steering geometry.

10.1.4 *Castor angle* (Figs 10.8 and 10.9)

The inclination of the swivel ball joint axis or king-pin axis in the fore and aft direction, so that the tyre contact centre is either behind or in front of the imaginary pivot centre produced to the ground, is known as the *castor angle* (Fig. 10.8(b and c)). Positive castor angle is established when the wheel contact centre trails behind the pivot point at
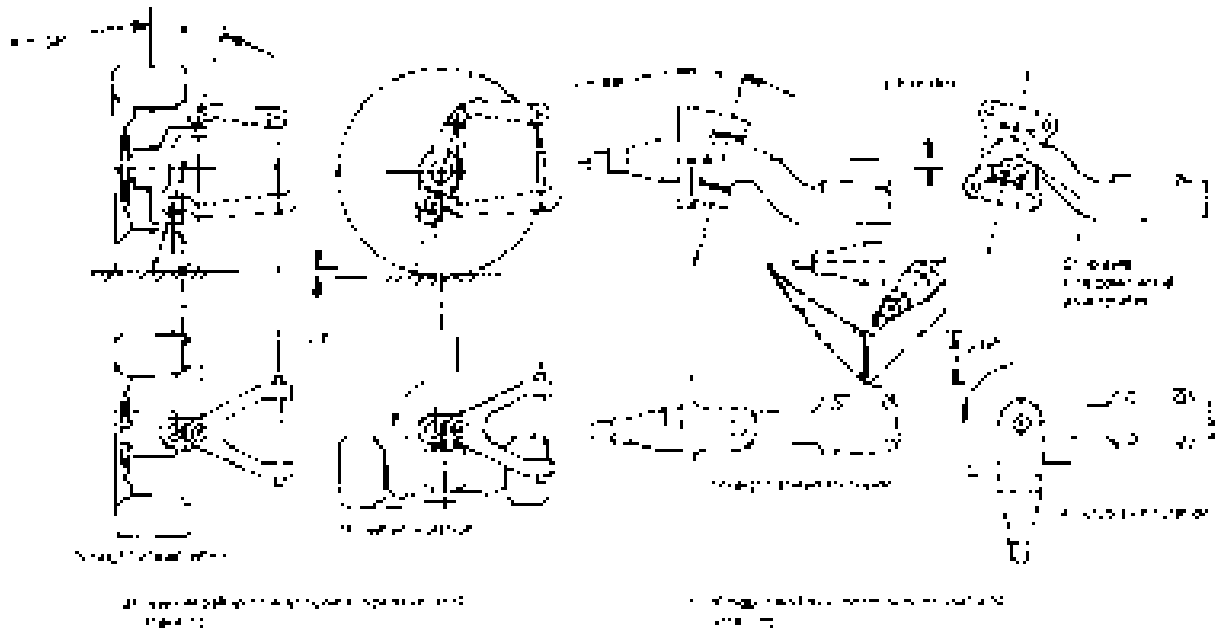
Fig. 10.6 (a and b)   Swivel and kingpin inclination self-straightening tendency

Fig. 10.7   Camber and swivel pin inclination centre point steering

ground level (Fig. 10.8(b)). Negative castor angle exists if the wheel contact centre leads the pivot axis intersection at ground level (Fig. 10.8(c)).

If the pivot centre and wheel contact patch centre coincide the castor is nil (Fig. 10.8(a)). Under these conditions the steered wheels become unstable as they tend to twitch from side to side when the vehicle travels along a straight path.

A rear wheel drive vehicle has the front wheel steer pivot axis inclined backward to produce positive castor (Fig. 10.9(a)). As the vehicle is propelled from the rear (the front wheels are pushed by the driving thrust transmitted by the rear drive wheels), it causes the front wheels to swing around their pivot axis until the tyre contact centre trails directly behind. This action takes place because the drag

force of the front tyres on the road causes both tyres to move until they are in a position where no out of balance force exists, that is, positioned directly to the rear of the pivot swivel balls or pin axis.

With front wheel drive vehicles the situation is different because the driving torque is transmitted through the steered front wheels (Fig. 10.9(b)). By inclining the pivot axis forwards, a negative castor is produced and instead of the pivot axis being pushed by the rear wheel drive thrust, traction is now transmitted through the front wheels so that the pivot axis is pulled forwards. The swivel balls or pin mounting swing to the rear of the contact patch centre, due to the vehicle rolling resistance acting through the rear wheels, opposing any forward motion.

The effects of castor angle can be seen in Fig. 10.9(a and b), when the steering is partially turned on one lock. The trail or lead distance between the pivot centre and contact patch centre rotates as the steered wheels are turned so that the forward driving force F_D and the equal but opposite ground reaction F_R are still parallel but are now offset by a distance x Therefore a couple (twisting movement)   is generated of magnitude $= Fx$, where $F = F_D = F_R$. With the vehicle in motion, the couple   will continuously try to reduce itself to zero by eliminating the offset x In other words, the driving and reaction forces F_D and F_R are at all

371
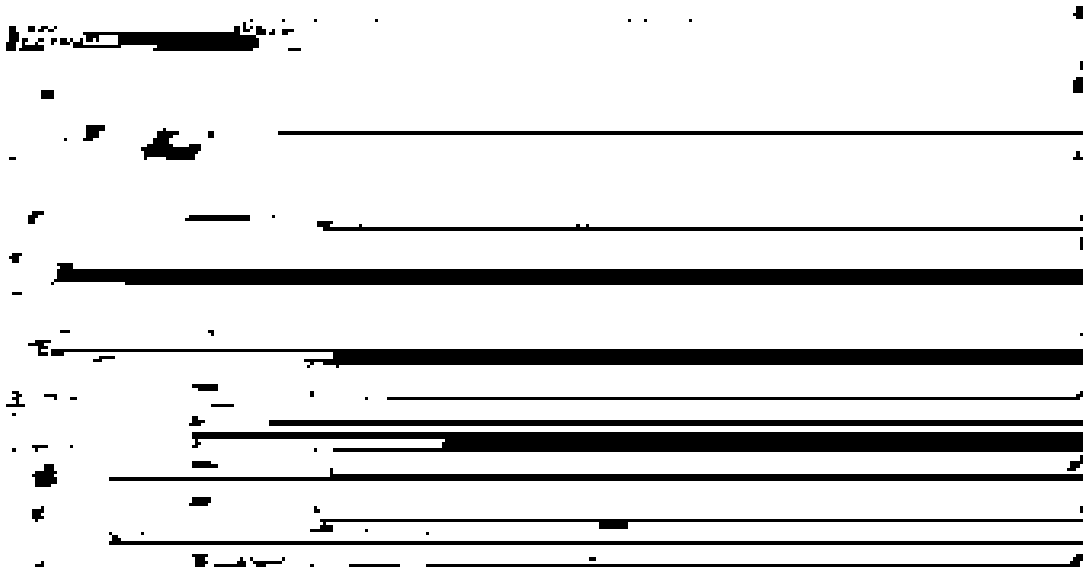
Fig. 10.8 Castor angle steering geometry

(a) Rear wheel drive castor
angle self-righting torque effect

(b) Front wheel drive castor
angle self-righting torque effect
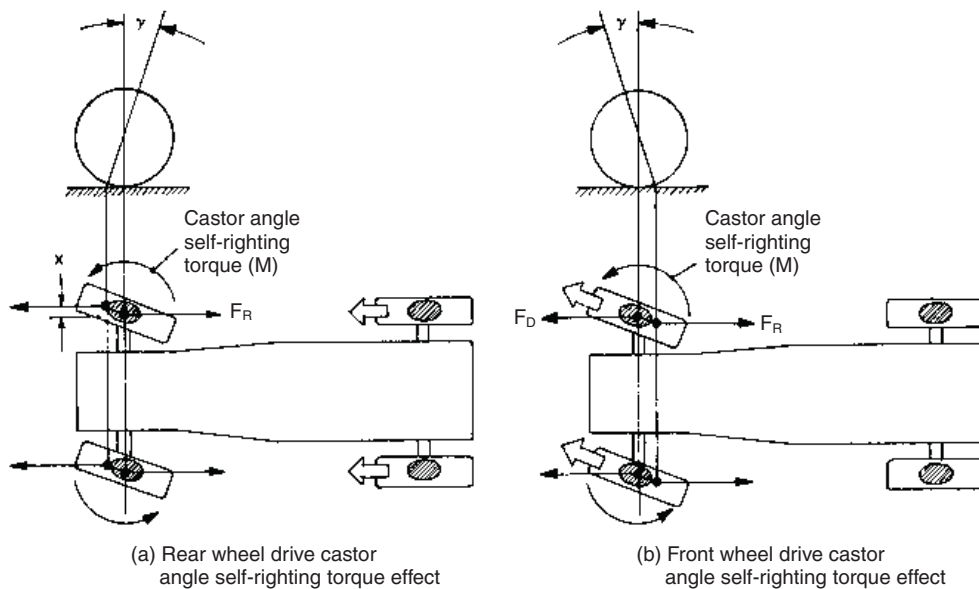
Fig. 10.9 (a and b) Illustration of steered wheel castor self-straightening tendency

times tending to align themselves with the wheels rolling when the steering has been turned to one lock. As a result the trailing or leading offset x produces a self-righting effect to the steered wheels. The greater the angle the wheels have been steered, the larger the pivot centre to contact patch centre offset x and the greater the castor self-centring action will be. The self-righting action which tends to straighten out the steering after it has

been turned from the straight position, increases with both wheel traction and vehicle speed.

10.1.5 Swivel joint positive and negative offset
(Figs 10.10–10.15)
When one of the front wheels slips during a brake application, the inertia of the moving mass will tend to swing the vehicle about the effective wheel which is bringing about the retardation because

372
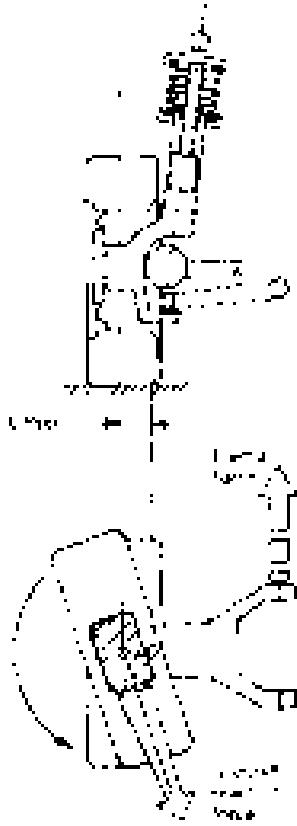
Fig. 10.10 Swivel pin inclination positive offset

Fig. 10.11 Swivel pin inclination negative offset

there is very little opposing resistance from the wheel on the opposite side (Fig. 10.12).

If the offset of the swivel ball joints is on the inside of the tyre contact patch the swivel inclination is known as *positive offset* (Fig. 10.10). When the wheels are braked the positive offset distance and the inertia force of the vehicle produce a turning movement which makes the wheels pivot about the contact patch centre in an outward direction at the front (Fig. 10.10). If the off side (right) wheel moves onto a slippery patch, the vehicle will not only veer to the left, due to the retarding effect of the good braked wheel preventing the vehicle moving forward, but the near side (left) wheel will also turn and steer to the left (Fig. 10.13). Therefore the positive offset compounds the natural tendency for the vehicle to swerve towards the left if the right hand wheel skids instead of continuing on a stable straight ahead path.

Arranging for the swivel ball joint inclination centre line to intersect the ground on the outside of the contact patch centre produces what is known as *negative offset* (Fig. 10.11). With negative offset the

Fig. 10.12 Directional stability when one wheel skids whilst being braked

373
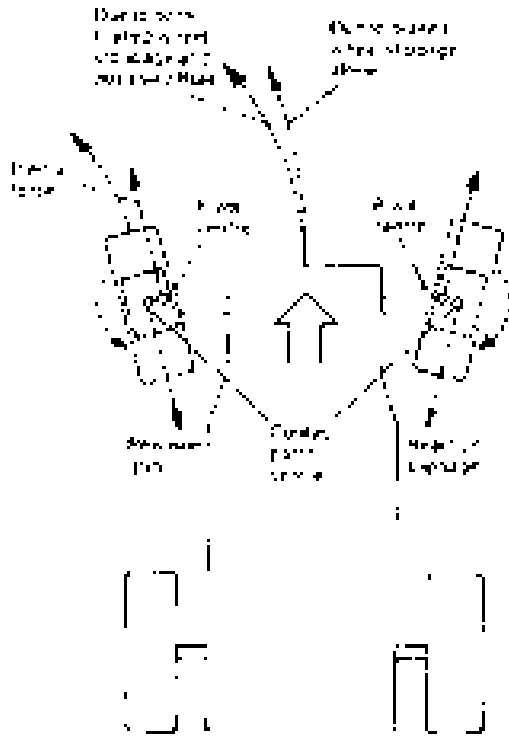
Fig. 10.13 Directional stability with positive offset when one wheel skids whilst being braked
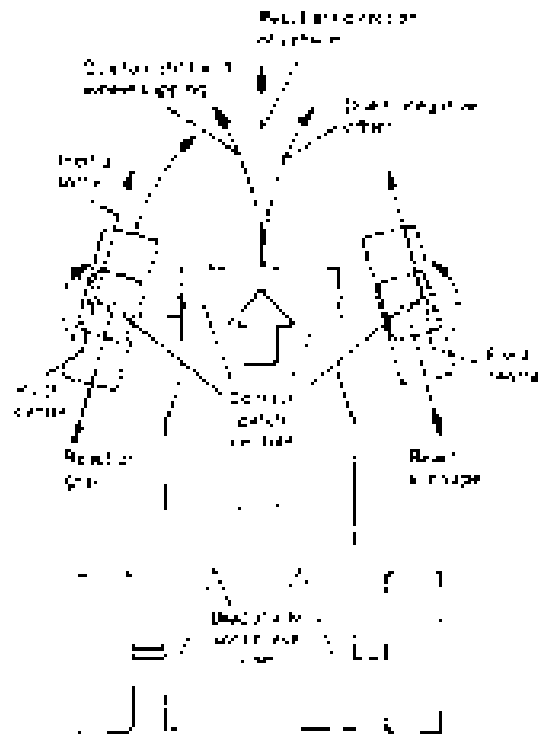
Fig. 10.14 Directional stability with negative offset when one wheel skids whilst being braked

momentum of the vehicle will produce a turning moment that makes the wheels swivel inwards at the front about the contact patch centre (Fig. 10.11) because the swivel ball joints and stub axle assembly are being pulled forwards and around the patch centre caused by the negative offset distance. The consequence of negative offset is that the effective braked wheel twists in the opposite direction to that to which the vehicle tends to veer (Fig. 10.14) and so counteracts the swerving tendency, enabling the vehicle to remain in a stable straight ahead direction.

In both positive and negative offset layouts, the skidding wheel turns in the same direction as the initial swerving tendency, but since it is not contributing greatly to the tyre to ground grip, its influence on directional stability is small.

The effect of negative offset is ideal for a split line braking system where if one brake line should fail, the front brake on the opposite side will still operate as normal (Fig. 10.14). The tendency for the car to veer to the side of the braked wheel is partially corrected by the wheel being turned due to the negative offset in the opposite direction (inwards), away from the direction in which the car wants to swerve.

When cornering, the sideways distortion of the tyre walls will misalign the wheel centre to that of the tread centre so that the swivel ball joint inclination offset will alter. The outer front wheel which supports the increase in weight due to body roll reduces positive offset (Fig. 10.15(a)), while negative offset becomes larger (Fig. 10.15(b)) and therefore makes it easier for the car to be steered when negotiating a bend in the road.

10.1.6 *acPherson strut friction and spring offset* (Figs 10.16 and 10.17)

The MacPherson strut suffers from stickiness in the sliding motion of the strut, particularly under light load with an extended strut since the cylinder rod bearing and the damper piston will be closer together. Because the alignment of the strut depends upon these two sliding members, extending and reducing their distance will increase the side loading under these conditions.

The problem of reducing friction between the inner and outer sliding members is largely overcome in two ways:

374
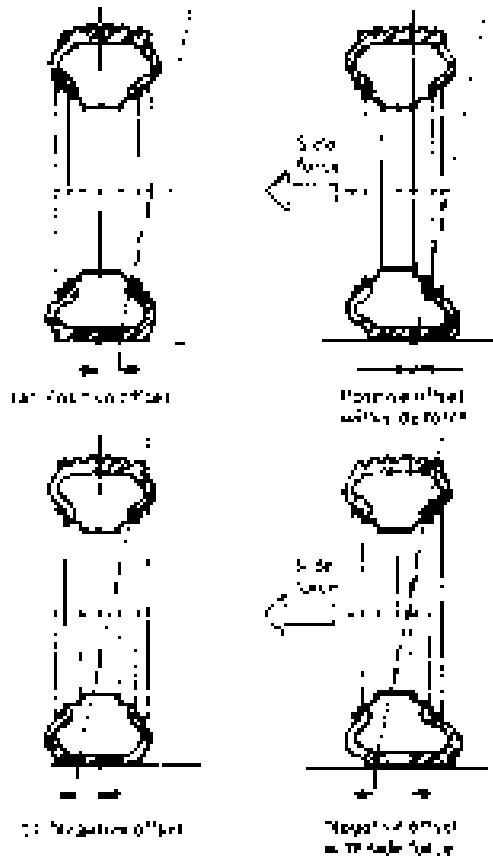
Fig. 10.15 (a and b)   Swivel pin inclination offset change when cornering
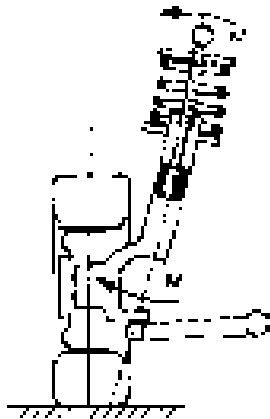
Fig. 10.16   Concentric coil spring and swivel pin axes permit bending moment reaction
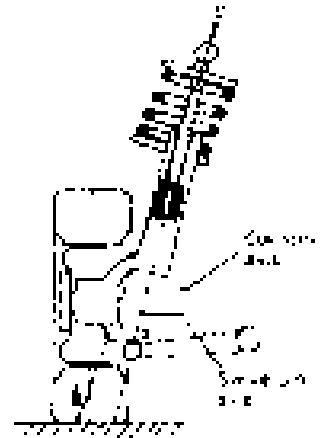
Fig. 10.17   Coil spring to swivel pin axis offset counteracts bending moment

(a) By reducing the friction, particularly with any initial movement, using a condition which is known as *stiction* This is achieved by facing the bearing surfaces with impregnated poly-tetra-fluorethytene (PTFE) which gives the rubbing pairs an exceptionally low coefficient of friction.

(b) By eliminating the bending moment on the strut under normal straight ahead driving although there will be a bending moment under cornering conditions.

The tendency for the strut to bend arises because the wheel is offset sideways from the strut, causing the stub axle to act as a cantilever from the base of the strut to the wheel it supports, with the result the strut bends in a curve when extended or under heavy loads (Fig. 10.16).

A simple solution which is commonly applied to reduce the bending moment on the strut is to angle the axis of the coil spring relative to the swivel joint axis causing the spring to apply a bending moment in the opposite sense to the vehicle load bending moment (Fig. 10.17). Under normal conditions this coil spring axis tilt is sufficient to neutralize the bending moment caused by the inclined strut and the stub axle offset, but the forces involved while cornering produce much larger bending moments which are absorbed by the rigidity of the strut alone.

10.2 Suspension roll centres

Roll centres (Fig. 10.29)   The roll centre of a suspension system refers to that centre relative to the ground about which the body will instantaneously

rotate. The actual position of the roll centre varies with the geometry of the suspension and the angle of roll.

Roll a is (Fig. 10.29) The roll axis is the line joining the roll centres of the front and the rear suspension. Roll centre height for the front and rear suspension will be quite different; usually the front suspension has a lower roll centre than that at the rear, causing the roll axis to slope down towards the front of the vehicle. The factors which determine the inclination of the roll axis will depend mainly on the centre of gravity height and weight distribution between front and rear axles of the vehicle.

10.2.1 Determination of roll centre height
(Fig. 10.18)
The determination of the roll centre height can be best explained using the three instantaneous centre method applied to the swing axle suspension, which is the basic design used for the development of almost any suspension geometry (Fig. 10.18).

A vehicle's suspension system involves three principal items; the suspended body B, the supporting wheels W and the ground G which provides the reaction to the downward load of the vehicle.

If a body which is suspended between two pairs of wheels is to be capable of rolling relative to the ground, then there must be three instantaneous centres as follows:

1 I_{BG} the instantaneous centre of the body relative to the ground which is more commonly known as the body roll centre,
2 I_{WB} the instantaneous centre of the wheel relative to the body which is the swing arm point of pivot,
3 I_{WG} the instantaneous centre of the wheel relative to the ground which is the contact centre between the tyre and ground. It therefore forms a pivot permitting the top of the wheel to tilt laterally inwards or outwards.

Fig. 10.18 Short swing axle

10.2.2 Short swing arm suspension
(Fig. 10.18)
When cornering, an overturning moment is generated which makes the body roll outwards from the centre of turn. The immediate response is that the inner and outer swing arm rise and dip respectively at their pivoted ends so that the inner and outer wheels are compelled to tilt on their instantaneous tyre to ground centres, I_{WG_1} and I_{WG_2}, in the opposite direction to the body roll.

For effective body roll to take place there must be two movements within the suspension geometry:

1 The swing arm pivot instantaneous centres I_{WB_1} and I_{WB_2} rotate about their instantaneous centres I_{WG_1} and I_{WG_2} in proportion to the amount of body roll.
2 The swing arm pivot instantaneous centres I_{WB_1} and I_{WB_2} move on a circular path which has a centre derived by the intersecting projection lines drawn through the tyre to ground instantaneous centres I_{WG_1} and I_{WG_2}.

The tilting, and therefore rotation, of both swing arms about the tyre to ground instantaneous centres I_{WG_1} and I_{WG_2} will thus produce an arc which is tangential to the circle on which the swing arm pivot instantaneous centres I_{WB_1} and I_{WB_2} touch. Therefore, the intersecting point I_{BG}, where the projection lines which are drawn through the wheel to ground contact points and the swing arm pivots meet, is the instantaneous centre of rotation for the body relative to the ground. This point is usually referred to as the *body roll centre*

Thus the body roll centre may be found by drawing a straight line between the tyre contact centre and swing arm pivot centre of each half suspension and projecting these lines until they intersect somewhere near the middle of the vehicle. The point of intersection becomes the body roll centre.

The roll centre height may be derived for a short swing arm suspension by consideration of similar triangles:

$$\frac{h}{t/2} = \frac{r}{l}$$

where
h	Roll centre height
t	Track width
r	Wheel radius
l	Swing arm length

Hence $h = \dfrac{tr}{2l}$

376

10.2.3 Long swing arm suspension (Fig. 10.19)

The long swing arm suspension is very similar to the short swing arm arrangement previously described, but the arms extend to the opposite side of the body relative to its wheel it supports and therefore both arms overlap with each other (Fig. 10.19).

The roll centre is determined by joining the tyre contact centre and the swing arm pivot centre by a straight line for each half suspension. The point where these lines meet is the body roll centre and its distance above or below the ground is known as the roll centre height. Because the long swing arm suspension has a much longer arm than used on the short swing arm layout, the slope of the lines joining the tyre contact centre and swing arm pivot is not so steep. Therefore the crossover point which determines the body roll centre height is lower for the long swing arm than for the short swing arm suspension.

The inherent disadvantage of the short swing arm suspension is that there is too much camber change with body roll and there is a tendency for the axle arms to jack the body up when cornering. Whereas the long swing arm suspension would meet most of the requirements for a good quality ride, it is impractical for a front suspension layout as it would not permit the engine to be situated relatively low between the two front wheels.

10.2.4 Transverse double wishbone suspension (Figs 10.20, 10.21 and 10.22)

If lines are drawn through the upper and lower wishbone arms and extended until they meet either inwards (Fig. 10.20) or outwards (Fig. 10.21), their intersection point becomes a virtual instantaneous centre for an imaginary (virtual) triangular swing arm suspension. The arc scribed by the wishbone arms pivoting relative to the body is almost identical to that of the imaginary or virtual arm which swings about the instantaneous virtual centres I_{BW_1}
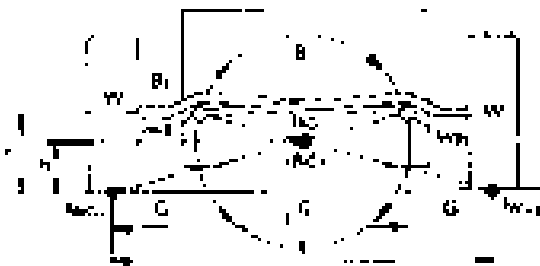
Fig. 10.19 Long swing axle

Fig. 10.20 Inward converging transverse double wishbone

Fig. 10.21 Outward converging transverse double wishbone

Fig. 10.22 Parallel transverse double wishbone

and I_{BW_2} for small movements of the suspension. Therefore, the body roll centre for a transverse double wishbone suspension can be derived similarly to a long swing arm suspension.

For inwardly converging transverse upper and lower wishbone arm suspension (Fig. 10.20) the body roll centre can be derived in two stages. Firstly, extend straight lines through the wishbone arms until they meet somewhere on the opposite side of the body at their virtual instantaneous centres I_{WB_1} and I_{WB_2}. Secondly, draw straight lines between the tyre contact centres I_{WG_1} and I_{WG_2} and the virtual centres I_{BW_1} and I_{BW_2} for each half suspension. The point where these inclined lines intersect is therefore the body roll centre I_{BG}.

For outward converging transverse upper and lower wishbone arm suspension (Fig. 10.21) the body roll centre is found again by drawing two

sets of lines. Firstly project straight lines through the wishbone arms for each side of the vehicle until they meet somewhere on the outside of each wheel at their virtual instantaneous centres I_{WB_1} and I_{WB_2}. Next draw straight lines between the tyre contact centres I_{WG_1} and I_{WG_2} and the virtual centres I_{WB_1} and I_{WB_2} for each half suspension, and at the same time extend these lines until they intersect near the middle of the vehicle. This point therefore becomes the body roll centre I_{BG}. It can be seen that inclining the wishbone arms so that they either converge inward or outward produces a corresponding high and low roll centre height.

With parallel transverse upper and lower wishbone arms suspension (Fig. 10.22) lines drawn through the double wishbone arms would be parallel. They would never meet and so the virtual instantaneous centres I_{WB_1} and I_{WB_2} would tend to infinity. Under these circumstances, lines normally drawn between the tyre contact centres I_{WG_1} and I_{WG_2} and the virtual instantaneous centres I_{WB_1} and I_{WB_2} would slope similarly to the wishbone extended lines. Consequently, the downwardly inclined parallel wishbone suspension predicts the tyre contact centre to virtual centre extended lines which meet at the roll centre would meet just above ground level. Therefore if the parallel wishbone arms were horizontally instead of downwardly inclined to the ground then the body roll centre would be at ground level.

10.2.5 Parallel trailing double arm and vertical pillar strut suspension (Figs 10.23 and 10.24)

In both examples of parallel double trailing arm (Fig. 10.23) and vertical pillar strut (Fig. 10.24) suspensions their construction geometry becomes similar to the parallel transverse double wishbone layout, due to both vertical stub axle members moving parallel to the body as they deflect up and down. Hence looking at the suspension from the front, neither the double trailing arms (Fig. 10.23) nor the sliding pillar (Fig. 10.24) layout has any trans-

Fig. 10.24 Vertical pillar strut

verse swing tendency about some imaginary pivot. Lines drawn through the two trailing arm pivot axes or sliding pillar stub axle, which represent the principle construction points for determining the virtual swing arm centres, project to infinity. The tyre contact centre to virtual instantaneous centre joining lines projected towards the middle of the vehicle will therefore meet at ground level, thus setting the body roll centre position. Inclining the trailing arm pivot axes or the vertical sliding pillar axis enables the roll centre height to be varied proportionally.

10.2.6 acPherson strut suspension (Fig. 10.25)

To establish the body roll centre height of any suspension, two of the three instantaneous centres, the tyre contact centre and the swing arm virtual centre must first be found. If straight lines are drawn between, and in some cases projected beyond, these instantaneous centres the third instantaneous centre which is the body roll centre becomes the point where both lines intersect.

The tyre contact centres (instantaneous centres I_{WG_1} and I_{WG_2}) where the wheels pivot relative to the ground are easily identified as the centres of the tyre where they touch the ground, but the second instantaneous virtual centre can only be found once the virtual or imaginary equivalent swing arm geometry has been identified.

For the MacPherson strut suspension (Fig. 10.25) the vertical swing arm and pivot centres I_{BW_1} and I_{BW_2} are obtained for each half suspension by projecting a line perpendicular to the direction

Fig. 10.23 Parallel trailing double arm

Fig. 10.25 MacPherson strut

of strut slide at the upper pivot. A second line is then drawn through and beyond the lower control arm until it intersects the first line. This point is the instantaneous virtual centre about which the virtual swing arm pivots.

Straight lines are then drawn for each half suspension between the tyre contact centre and the virtual swing arm centre. The point of intersection of these two lines will then be the third instantaneous centre I_{BG}, commonly referred to as the body roll centre.

10.2.7 Semi-trailing arm rear suspension
(Fig. 10.26)
A semi-trailing arm suspension has the rear wheel hubs supported by a wishbone arm pivoted on an inclined axis across the body (Fig. 10.26(a)).

If lines are projected through the wishbone arm pivot axis and the wheel hub axis they will intersect at the virtual instantaneous centres I_{BW_1} and I_{BW_2} (Fig. 10.26(a and b)). The distance between these centres and the wheel hub is the transverse equivalent (virtual) swing arm length a Projecting a third line perpendicular to the wheel hub axis so that it intersects the skewered wishbone arm axis produces the equivalent fore and aft (trailing) swing arm length b for the equivalent (virtual) semi-trailing triangular arm (Fig. 10.26(c)). The movement of this virtual swing arm changes the wheel camber and moves the wheel hub axis forward as the wheel deflects in bump or bounce from the horizontal position.

The body roll centre can now be determined by drawing a rear view of both virtual swing arms (Fig. 10.26(b)) and then drawing lines between each half swing arm instantaneous pivot centres I_{WB_1} and I_{WB_2} and the tyre contact centres I_{WG_1} and I_{WG_2}. The point where these two sloping lines cross over can then be defined as the body roll centre I_{BG}.

10.2.8 High load beam axle leaf spring sprung body roll stability (Fig. 10.27)
The factors which influence the resistance to body roll (Fig. 10.27) are as follows:

a) The centrifugal force acting through the centre of gravity of the body load.
b) The arm length from the centre of load to the effective roll centre h_1 or h_2.
c) The spring stiffness in Newtons/metre of vertical spring deflection.
d) The distance between the centres of both springs known as the spring stability base t_s.
e) The distance between road wheel centres known as the tyre stability base t_w.

Considering the same side force acting through the centre of gravity of the body load and similar spring stiffness for both under- and over-slung springs (Fig. 10.27), two fundamental observations can be made.

Firstly it can be seen (Fig. 10.27) that with over-slung springs the body roll centre RC_1 is much higher than that for underslung springs RC_2 and therefore the overslung springs provide a smaller overturning arm length h_1 as opposed to h_2 for the underslung springs. As a result, the high roll centre with the small overturning arm length offers a greater resistance to body roll than a low roll centre with a long overturning arm.

Secondly it can be seen (Fig. 10.27) that the triangular projection lines produced from the centre of gravity through the centres of the springs to

Fig. 10.26 Semi-trailing arm
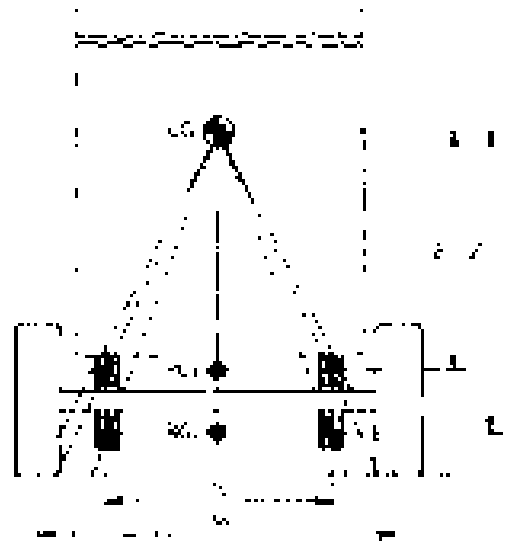
Fig. 10.27 Effects of under- and over-slung springs on the roll centre height

the ground provide a much wider spring stability base for the high mounted springs compared to the low mounted underslung springs. In fact the overslung spring centre projection lines nearly approach the tyre stability base width t_w which is the widest possible for such an arrangement without resorting to outboard spring seats.

10.2.9 Rigid axle beam suspension
(Fig. 10.28(a–d))
An axle beam suspension is so arranged that both wheel stub axles are rigidly supported by a common transverse axle beam member which may be a steered front solid axle beam, a live rear axle hollow circular sectioned casing or a DeDion tubular axle beam.

With a rigid axle beam suspension there cannot be any independent movement of the two stub axles as is the case with a split swing axle layout. Therefore any body roll relative to the ground must take place between the axle beam and the body itself. Body roll can only take place about a mechanical pivot axis or about some imaginary axis somewhere near mid-spring height level.

Methods used to locate and control the axle movement are considered as follows:

Longitudinally located semi-elliptic springs
(Fig. 10.28(a)) When semi-elliptic leaf springs support the body, the pivoting point or body roll centre will be roughly at spring-eye level but this will become lower as the spring camber (leaves bow) changes from positive upward bowed leaves when unloaded to negative downward bowed leaves with increased payload.

Transverse located Panhard rod (Fig. 10.28(b)) The use of coil springs to support the body requires some form of lateral body to axle restraint if a torque tube type axle is to be utilized. This may be provided by a diagonally positioned Panhard rod attached at its ends to both the axle and body. When the body tilts it tends to move sideways and either lifts or dips depending which way the side force is applied. Simultaneously the body will roll about the mid-position of the Panhard rod.

Diagonally located tie rods (Fig. 10.28(c)) To provide both driving thrust and lateral support for
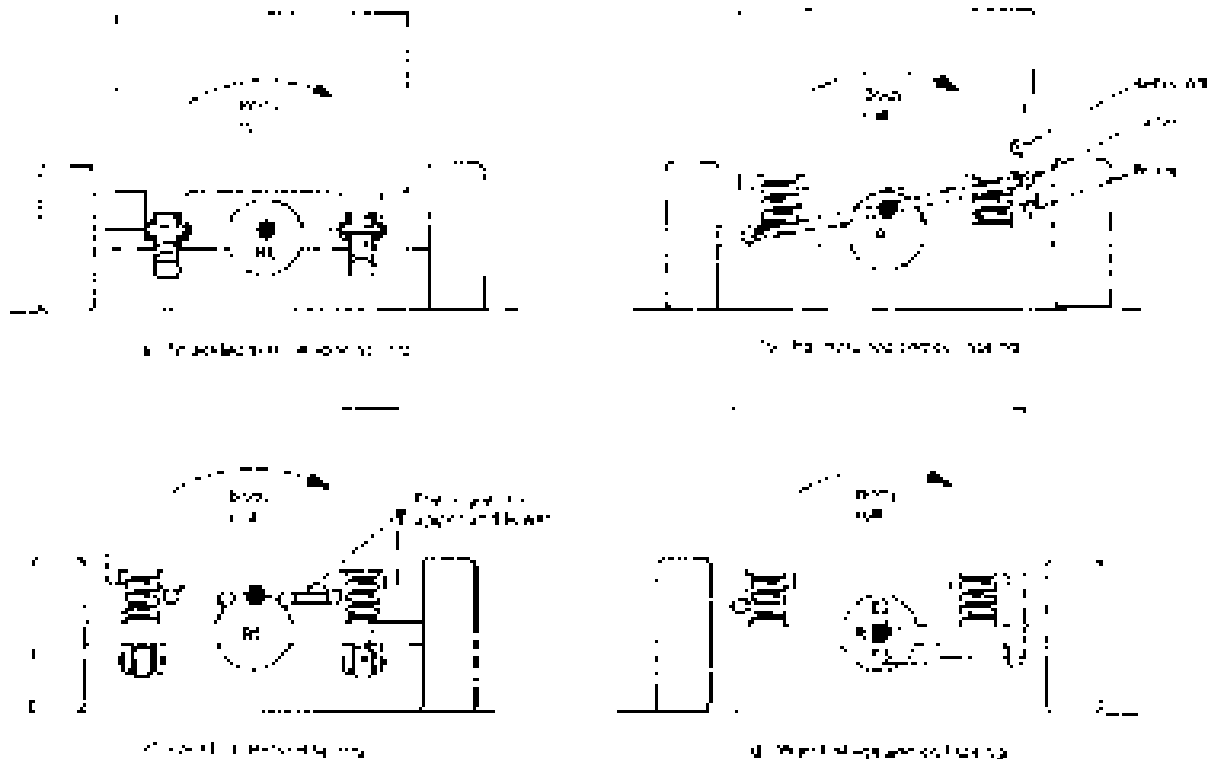
Fig. 10.28 (a–d) Body roll centres for rigid beam axle suspensions

380

a helical coil spring live axle layout, a trailing four link suspension may be adopted which has a pair of long lower trailing arms which absorb both the driving and braking torque reactions and a pair of short upper diagonally located tie rods to control any lateral movement. Any disturbing side forces which attempt to make the body tilt sideways will cause it to roll about a centre roughly in line with the upper tie rod height.

ransverse att linkage (Fig. 10.28(d)) An alternative arrangement for controlling the sideways movement for a coil spring suspension when used in conjunction with either a live axle or a DeDion tube is the Watt linkage. Suspension linkages of this type consist of a pair of horizontal tie rods which have their outer ends anchored to the body and their inner ends coupled to a central balance lever which has its pivot attachment to the axle beam. If the body is subjected to an overturning moment it will result in a body roll about the Watt linkage balance lever pivot point. This instantaneous centre is therefore the body roll centre.

10.3 Body roll stability analysis

When a vehicle turns a corner the centrifugal force produced acts outwards through the centre of gravity of the sprung mass, but it is opposed by the tyre to ground reaction so that the vehicle will tend to overturn. An overturning moment is therefore generated which tends to transfer weight from the inner wheels to the outside wheels. At the same time due to the flexibility and softness of the suspension, the body rolls so that in effect it overhangs and imposes an additional load to the outer wheels.

The opposition to any body roll will be shared out between the front and rear suspension according to their roll resistance. Thus if the front suspension roll stiffness with an anti-roll bar is twice that of the rear, then the front wheels will sustain two thirds of the roll couple while the rear ones only carry one third.

10.3.1 Body roll couple (Fig. 10.29)

The body roll couple (moment) consists of two components:

Centrifugal moment about the roll centre =
$$Fa \ \text{Nm}$$
Transverse displacement moment $= w \ a \ \tan$
$$\simeq \quad a \ \text{(Nm)}$$

where F centrifugal side force
       a distance between the centre of gravity and roll centre
       w unsprung weight
         angle of body roll

Hence

Total roll movement or couple $= Fa + \quad a$
$$= (F + \quad) a \ \text{(Nm)}$$
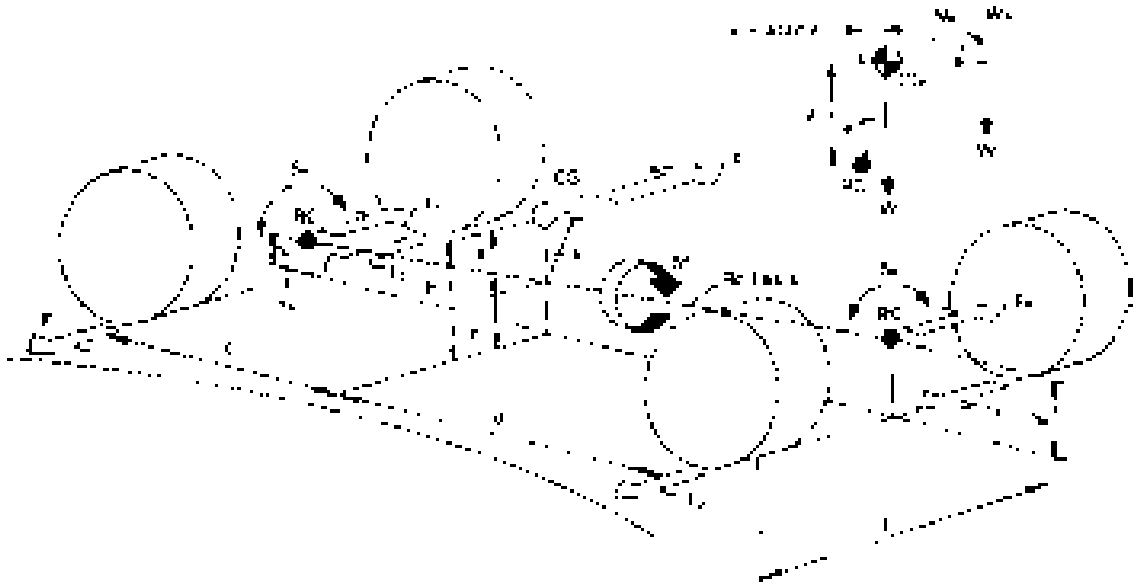
Fig. 10.29 Body roll centres and roll axis

381

The sum of these couples are resisted by the springs in proportion to their torsional stiffness at the front and rear.

Body roll stiffness (Fig. 10.29) The body roll stiffness is defined as the roll couple produced per degree of body roll.

i e Roll stiffness $= \dfrac{\text{Roll couple}}{\text{Roll angle}}$ (Nm/deg)

hence $\qquad S = \text{---}$ (Nm/deg)

where $\quad S \quad$ roll stiffness (Nm/deg)
$\qquad\qquad$ roll couple (Nm)
$\qquad\qquad$ angle of roll (deg)

The fraction of torsional stiffness for the front and rear suspensions will therefore be:

$$S_F = \frac{S_F}{S_F + S_R} \left(\frac{\text{Nm/deg}}{\text{Nm/deg}} \right)$$

$$S_R = \frac{S_R}{S_R + S_F}$$

where $\quad S_F \quad$ fraction of front torsional stiffness
$\qquad\quad S_R \quad$ fraction of rear torsional stiffness

10.3.2 Body overturning couple (Fig. 10.30)
The centrifugal force F created when a vehicle is travelling on a circular track acts through the body's centre of gravity CG at some height h and is opposed by the four tyre to ground reaction forces F_1, F_2, F_3 and F_4.

Consequently an overturning couple Fh is produced which transfers weight from the inside wheels to the outer wheels which are spaced the track width t apart. Thus the overturning couple will also be equivalent to t, that is, $t = Fh$.

i e Weight transferred $= \dfrac{Fh}{t}$ (N)

It should be noted that the centre of gravity height h is made up from two measurements; the distance between the ground and the body roll centre b and the distance between the roll centre and the centre gravity a.

Therefore
Total body roll couple $= Fh = F(a + b)$ (N)
$$= Fa + Fb \text{ (N)}$$

10.3.3 Body roll weight transfer (Fig. 10.31)
The product Fa is the overturning couple rotating about the roll centre which causes the body to roll. This couple is opposed by a reaction couple Rt where R is the vertical reaction force due to the weight transfer and t is the wheel track width.

Therefore $\qquad Rt = Fa$
$$R = \frac{Fa}{t} \text{ (N)}$$

This shows that as the distance between the ground and the body roll centre known as the couple arm becomes smaller, the overturning couple and therefore the body roll will also be reduced in the same proportion. Thus if the couple arm a is reduced to zero the reaction force R will likewise approach zero. A small couple is desirable so that the driver experiences a sense of body roll as a warning for cornering stability. If both roll centre and centre of gravity height coincided there would be no indication to the driver that the lateral forces acting on the body were reaching the limit of the tyre to ground sideway grip. Consequently suspensions in which the centre of gravity and the roll centre are at the same height can cause without warning a sudden tyre to ground breakaway when cornering at speed.

10.3.4 Body direct weight transfer couple
(Fig. 10.32)
If the centrifugal force acted through the roll centre axis instead of through the centre of gravity, a

Fig. 10.30 Overturning couple
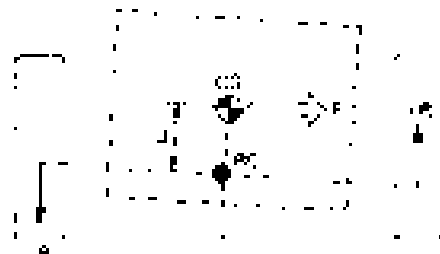
Fig. 10.31 Body roll weight transfer

Fig. 10.32 Direct weight transfer

moment F_b about the ground would be produced so that a direct transference of weight from the inner to the outer wheels occurs. The reaction to this weight transfer for a track width t is a resisting moment R_t which is equal but opposite in sense to the moment F_b.

Hence $\quad Rt = Fb$,

therefore $\quad R = \dfrac{Fb}{t}(\text{N})$

If the fore and aft weight distribution is proportional between the front and rear axle roll centres, the centrifugal force F acting through the roll centre axis would be split into two forces F_F and F_R which act outwards from the front and rear roll centres.

Thus $\quad R_F = \dfrac{F_F\, b_F}{t}(\text{N})$

$\qquad\quad R_R = \dfrac{F_R\, b_R}{t}(\text{N})$

where R, R_F and R_R = Total, front and rear vertical reaction forces respectively

Thus lowering the body roll centre correspondingly reduces the vertical reaction force R and by having the roll centre at ground level the direct weight transfer couple will be eliminated.

Therefore if the roll axis slopes from the ground upwards from front to rear, all the direct weight transfer couple will be concentrated on the rear wheels.

10.3.5 Body roll couple distribution (Fig. 10.29)
The body roll couple on the front and rear tyres is proportional to the front and rear suspension stiffness fraction.

i.e. Roll couple on front tyres

$$_F = \frac{S_F}{S_F + S_R}(F + \quad)a + F_F h_F(\text{Nm})$$

Roll couple on rear tyres

$$_R = \frac{S_R}{S_R + S_F}(F + \quad)a + F_R h_R(\text{Nm})$$

Body roll angle The body roll angle may be defined as the roll couple per unit of roll stiffness

i e $\quad$ Total roll angle $= \dfrac{\text{Roll couple}}{\text{Roll stiffness}}\ \dfrac{\text{Nm}}{\text{Nm/deg}}$

$$= \frac{}{S_F + S_R}(\text{deg})$$

10.3.6 Body roll weight transfer (Fig. 10.29)
The body roll weight transferred may be defined as the roll couple per unit width of track

i.e. $\quad$ Total roll weight transfer

$$= \frac{\text{Roll couple}}{\text{Track width}}\left(\frac{\text{Nm}}{\text{m}}\right)$$

hence $\quad = \dfrac{}{t}(\text{N})$

Front suspension weight transfer

$$_F = \frac{S_F}{S_F + S_R} \times \frac{}{t}(\text{N})$$

Rear suspension weight transfer

$$_R = \frac{S_R}{S_R + S_F} \times \frac{}{t}(\text{N})$$

where $\quad$, $_F$ and $_R$ $\quad$ Total, front and rear weight transfer respectively (N)

$\qquad\qquad\qquad t \quad$ Wheel tract (m)

10.3.7 Lateral (side) force distribution
(Fig. 10.33)
The total lateral resisting forces generated at all tyre to ground interfaces must equal the centrifugal

Fig. 10.33 Longitudinal weight distributions

383

Fig. 10.34 (a and b) Comparison of rigid and independent suspension body roll stiffness

force acting through the body's centre of gravity. Thus the fore and aft position of the centre of gravity determines the weight distribution between the front and rear wheels and therefore the proportion of cornering force necessary to be generated by their respective tyres.

If F_F and F_R are the front and rear tyre to ground cornering forces, then taking moments about F_R

$$F_F l = F_b$$

Therefore $\quad F_F = \dfrac{F_b}{l}(N)$

$$F_R l = F_a$$

Therefore $\quad F_R = \dfrac{F_a}{l}(N)$

Thus the amount of load and cornering force carried by either the front or rear tyres is proportional to the distance the centre of gravity is from the one or the other axle. Normally there is slightly more weight concentrated at the front half of the vehicle so that greater cornering forces and slip angles are generated at the front wheels compared to the rear.

10.3.8 Comparison of rigid axle beam and independent suspension body roll stiffness
(Fig. 10.24)

A comparison between roll stiffness of both rigid axle beam and independent suspension can be derived in the following manner:

Consider the independent suspension (Fig. 10.34(a)). Let the centrifugal force F act through the centre of gravity CG at a height h above the roll centre RC. The overturning couple Fh produced must be equal and opposite to the reaction couple t_w created by a reduction in the inside wheel reaction − and a corresponding increase in the

outside wheel reaction + between the effective spring span t_w.

If the vertical spring stiffness is S N/m and the vertical deflection at the extremes of the spring span is x m then the angle of body roll degrees can be derived as follows:

$$\tan = \frac{x}{t_w/2} = \frac{2x}{t_w} \quad (1)$$

Weight transfer $= x\,S$

Therefore Overturning couple $= Fh$

and Reaction couple $= t = Sxt$

$$(\text{since} = Sx)$$

$$Fh = Sxt_w$$

$$\text{or} \quad x = \frac{Fh}{t_w S} \quad (2)$$

From (1) $\quad \tan = \dfrac{2x}{t_w}$

but $\quad x = \dfrac{Fh}{St_w}$

so $\quad \tan = \dfrac{2}{t}\dfrac{Fh}{St_w}$

When is small, $\tan \simeq$

$$= \frac{2Fh}{St_w{}^2} \quad (3)$$

This formula shows that the body roll angle is proportional to both centrifugal force F and the couple arm height h but it is inversely proportional to both the spring stiffness k and the square of the spring span $t_w{}^2$, which in this case is the wheel track.

i e $\quad \propto F, \quad \propto h, \quad \propto \dfrac{1}{S}$ and $\propto \dfrac{1}{t_w{}^2}$

A similar analysis can be made for the rigid axle beam suspension (Fig. 10.34(b)), except the spring span then becomes the spring base t_s instead of t_w. Because the spring span for a rigid axle beam suspension is much smaller than for an independent suspension (t_w^2 t_s^2), the independent wide spring span suspension offers considerably more roll resistance than the narrow spring span rigid axle beam suspension and is therefore preferred for cars.

10.4 Anti-roll bars and roll stiffness (Fig. 10.35)

10.4.1 Anti-roll bar function
A torsion anti-roll bar is incorporated into the suspension of a vehicle to enable low rate soft springs to be used which provides a more comfortable ride under normal driving conditions. The torsion bar does not contribute to the suspension spring stiffness (the suspension's resistance to vertical deflection) as its unsprung weight is increased or when the driven vehicle is subjected to dynamic shock loads caused possibly by gaps or ridges where concrete sections of the road are joined together. However, the anti-roll bar does become effective if one wheel is raised higher than the other (Fig. 10.35) as the vehicle passes over a hump in the road or the body commences to roll while cornering. Under these conditions, the suspension spring stiffness (total spring rate) increases in direct proportion to the relative difference in deflection of each pair of wheels when subjected to the bump and rebound of individual wheels or body roll when the vehicle is moving on a circular path.

10.4.2 Anti-roll bar construction (Fig. 10.36)
Generally the anti-roll bar is formed from a medium carbon steel solid circular sectioned rod which is positioned transversely and parallel to the track (Fig. 10.36) so that it nearly spans the distance between the road wheels (Fig. 10.35). The bar is bent at both ends in right angles to form cracked arms. These arms can then be actuated by short link rods attached to the unsprung portion of the suspension such as the axle beam or transverse wishbone arms for independent suspension. The main transverse span of the rod is supported by rubber bearings positioned on the inside of the cranked arms at each end. These bush bearings are either mounted directly onto the body structure when incorporated
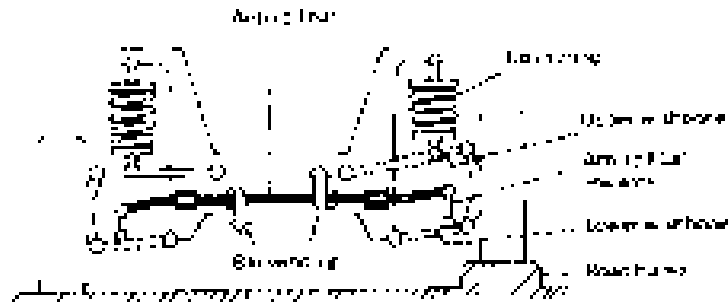
Fig. 10.35 Transverse double wishbone coil spring independent suspension with anti-roll bar
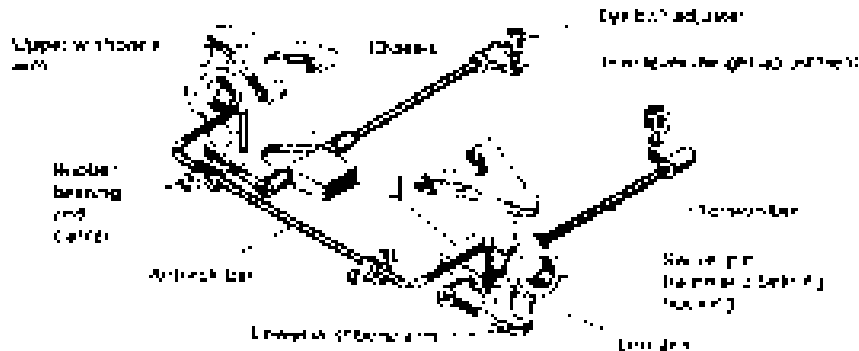
Fig. 10.36 Transverse double wishbone torsion bar independent suspension with anti-roll bar
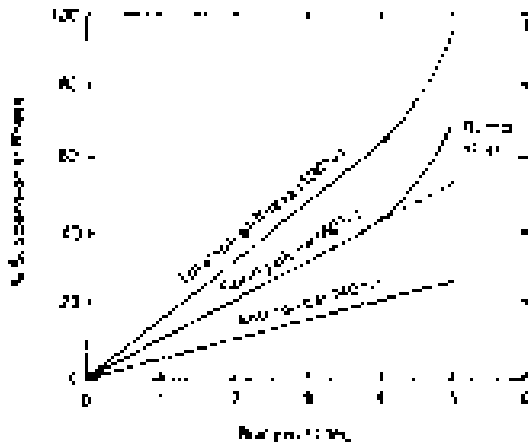
Fig. 10.37 Relationship of body roll and suspension spring and anti-roll bar stiffness

Fig. 10.38 Relationship of body roll and the understeer tendency with and without an anti-roll bar

on cars (Fig. 10.35) or indirectly for commercial vehicles (Fig. 10.39) on short vertical arms which provide a swing attachment to the chassis.

10.4.3 Anti-roll bar operating principle

When a pair of road wheels supported on an axle travel over a bumpy road one or other wheel will lift and fall as they follow the contour of the road surface. If the springs were relatively hard, that is they have a high spring rate, then the upthrust caused by the bumps would be transmitted to the body which would then lift on the side being disturbed. Thus the continuous vertical deflection of either wheel when the vehicle moves forward would tend to make the body sway from side to side producing a very uncomfortable ride. On the other hand if softer springs were used for the suspension, the small road surface irregularities would be adequately absorbed by the springs and dampers, but when cornering there would be insufficient spring stiffness to resist the overturning moment; this would therefore permit excessive body roll which could not be tolerated. Incorporating an anti-roll bar with relatively soft suspension springs mostly overcomes the difficulties discussed and therefore greatly improves the vehicle's ride. This is possible because the soft springs improve the suspension's response on good straight roadways (Fig. 10.37), with the benefits of the anti-roll bar automatically increasing the suspension roll stiffness when the vehicle is cornering.

10.4.4 Anti-roll bar action caused by the body rolling (Fig. 10.39(a and b))

When cornering, the centrifugal force acting through the centre of gravity of the sprung body produces an overturning moment created by its offset to the body's roll centre which will therefore tend to make the body roll (Fig. 10.39(a and b)). The rolling body will tilt the transverse span of the roll bar with it so that the cranked arms on the outside wheel to the turn will be depressed downward, whereas the cranked arm on the opposite end near the inside wheel to the turn will tend to rise. The consequence of this misalignment of the anti-roll bar arms is that the two cranked arms will rotate in opposite directions to each other and so transmit a torque from the inside wheel which is subjected to less load to the outside wheel which is now more heavily loaded. The effect of the torsional wind-up in the bar is that it tries to rotate the outside wheel cranked arm and since the arm is attached to the axle or indirectly to the wishbone arm it cannot move. The alternative is for the roll bar and the rubber bearing mount near the outside wheel to lift in proportion to the degree of twisting torque. It therefore counteracts some of the downward push due to the increase in weight to the outside wheel and as a result stiffens the roll resistance of the springing on the outside wheel as a whole. Consequently a larger slip angle is generated on the front outside wheel relative to the rear wheel, and as a result, the vehicle will develop a small initial but progressive understeer tendency approximately proportional to the amount the body rolls (Fig. 10.38).

10.4.5 Anti-roll bar action caused by single wheel lift (Fig. 10.39(c and d))

If one of a pair of axle wheels lifts as it climbs over a bump (Fig. 10.39(c)) in the road, then the vertical

386
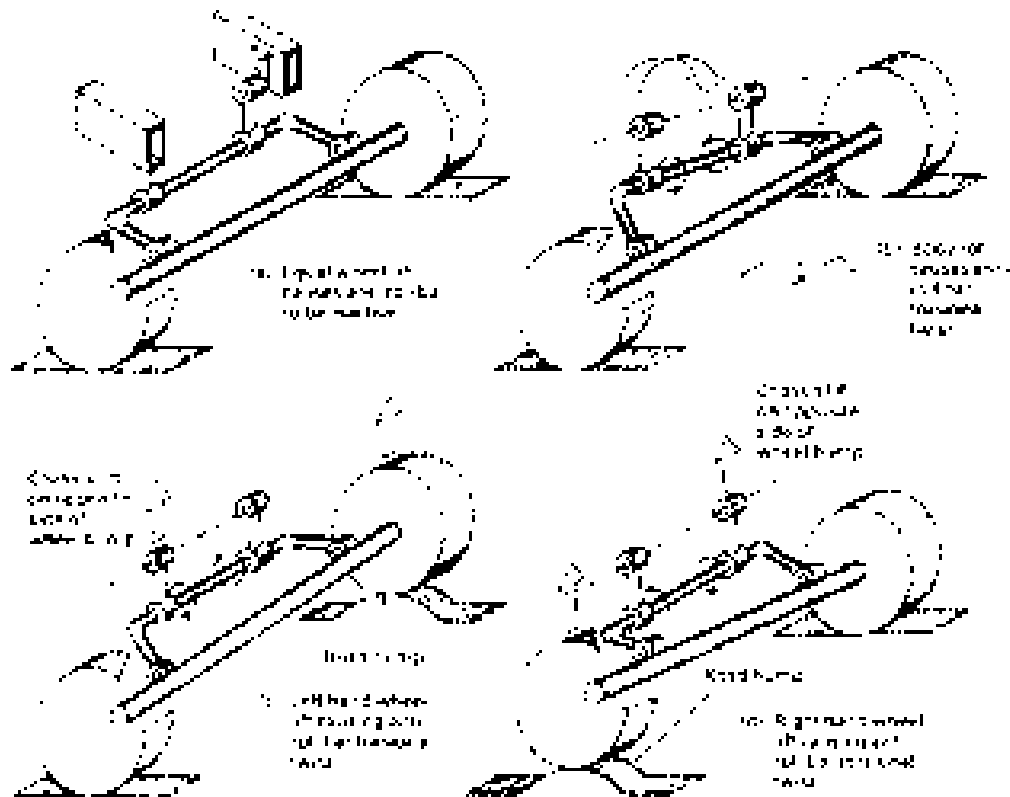
Fig. 10.39 (a–d)   Anti-roll bar action

deflection of the wheel and spring raises and rotates the anti-roll bar's cranked arm on that side so that the transverse span of the bar is twisted. The bar is therefore subjected to a torque which is proportional to its angle of rotation.

This twisting torque is transferred to the opposite cranked arm which then applies a downward force onto the axle and wheel. However, because the wheel cannot sink into the ground, the reaction occurs on the rubber bearing mount arm which therefore tends to lift up the side of the chassis on the opposite side to the vertically deflected wheel. As a result, both sides of the chassis (body) will have been raised, thereby enabling the vehicle's body to remain upright instead of tilting to one side. Similarly, if the opposite wheel hits an obstacle in the road (Fig. 10.39(d)), the torsional wind-up of the bar transfers an upward thrust to the other side, which again tends to lift the chassis on the undisturbed wheel side and so maintains the sprung chassis and body on an even keel (Fig. 10.39(c)).

10.5 Rubber spring bump or limiting stops

10.5.1 Bump stop function
(Figs 10.40 and 10.42)
Suspension bump and body roll control depends upon the stiffness of both the springs and anti-roll bar over the normal operating conditions, but if the suspension deflection approaches maximum bump or roll the bump stop (Fig. 10.40(a, b, c and d)) becomes active and either suddenly or progressively provides additional resistance to the full deflection of the wheel and axle relative to the body (Fig. 10.42). The bump stop considerably stiffens the resisting spring rate near the limit of its vertical movement to prevent shock impact and damage to the suspension components. The stop also isolates the sprung and unsprung members of the suspension under full deflection conditions so that none of the noise or vibrations are transmitted through to the body structure. In essence the bump stop enables an anti-roll bar to be used which has a slightly lower spring rate, therefore permitting

387

a more cushioned ride for a moderate degree of body roll.

10.5.2 Bump stop construction (Fig. 10.40(a–d))

Bump stops may be considered as limiting springs as they have elastic properties in compression similar to other kinds of spring materials. Solid and hollow spring stops are moulded without reinforcement from natural rubber compound containing additives to increase the ozone resistance. The deflection characteristics for a given size of rubber stop spring are influenced by the hardness of the rubber, this being controlled to a large extent by the proportion of sulphur and carbon black which is mixed into the rubber compound. The most common rubber compound hardness used for a rubber spring stop is quoted as a shore hardness of 65; other hardness ranging from 45 to 75 may be selected to match a particular operating requirement. A solid cylindrical rubber block permits only 20% deflection when loaded in compression, whereas hollow rubber spring stops have a maximum deflection of 50–75% of their free height. The actual amount of deflection for a given spring stop height and response to load will depend upon a number of factors such as the rubber spring stop size, outer profile, wall thickness, shape of inner cavities, hardness of rubber compound and the number of convolution folds.

Bump rubber spring stops may be solid and conical in shape or they may be hollow and cylindrical or rectangular shaped with a bellow profile (Fig. 10.40(a, b, c and d)) having either a single, double or triple fold (known as convolutions). The actual profile of the rubber bump stop selected will depend upon the following:

1 How early in the deflection or load operating range of the suspension the rubber begins to compress and become active.
2 Over what movement and weight change the bump stop is expected to contribute to the sudden or progressive stiffening of the suspension so that it responds to any excessive payload, impact load and body roll.

10.5.3 Bump stop characteristics
(Figs 10.41 and 10.42)

The characteristics of single, double and triple convolution rubber spring stops, all using a similar rubber hardness, are shown in Fig. 10.41. It can be seen that the initial deflection for a given load is large but towards maximum deflection there is very little compression for a large increase in load. The relation between load and deflection for bump is not quite the same on the release rebound so that the two curves form what is known as a hysteresis loop. The area of this loop is a measure of the energy absorbed and the internal damping within
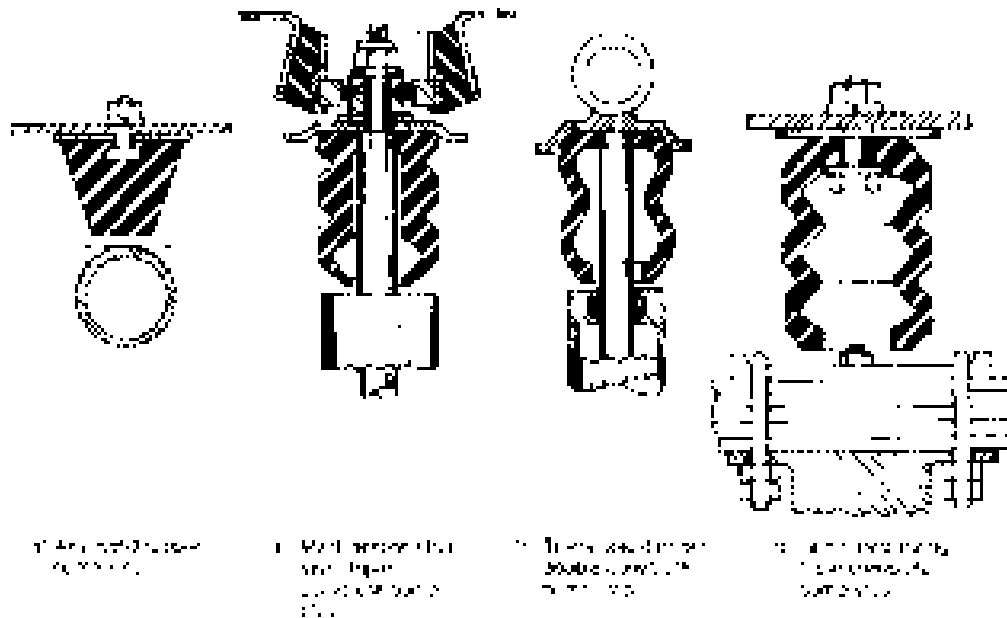
Fig. 10.40 (a–d) Suspension bump stop limiter arrangements

388
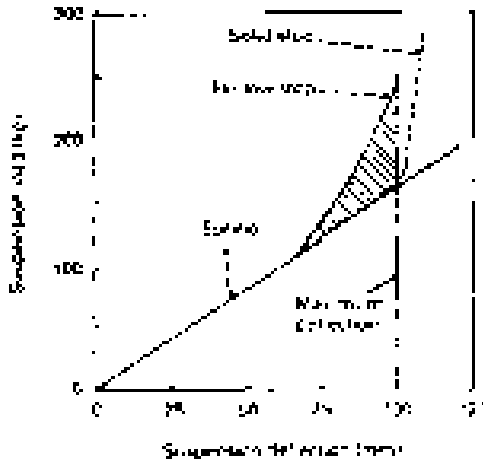
Fig. 10.41 Characteristics of hollow rubber single, double and triple convolute progressive bump stops
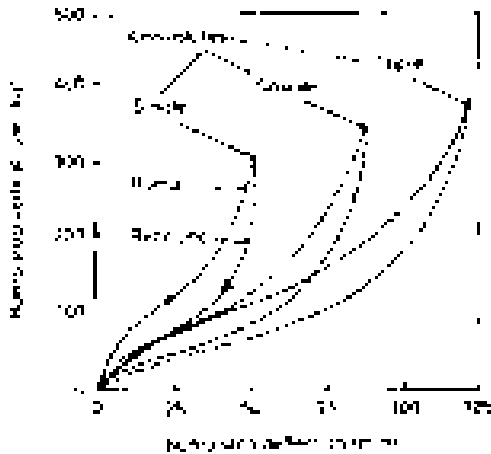
Fig. 10.42 Combined characteristics of suspension spring and rubber bump stops

the rubber in one cycle of compression and expansion of the rubber spring stop. For hollow rubber spring stops they always end in a point; this means for any load change there will be some spring deflection.

Fig. 10.42 shows how the bump spring stop deviates from the main spring load-deflection curve at about two-thirds maximum deflection and that the resultant stiffness (steepness of curve) of the steel spring, be it leaf, coil or torsion bar, and that of the bump spring stops considerably increases towards full load.

10.6 Axle location

10.6.1 Torque arms (Figs 10.28(c) and 10.44)
Torque arms, sometimes known as radius arms or rods, are mounted longitudinally on a vehicle between the chassis/body structure and axle or unsprung suspension member. Its purpose is to permit the axle to move up and down relative to the sprung chassis/body and to maintain axle alignment as the torque arm pivots about its pin, ball or conical rubber joint. Sometimes the upper torque rods are inclined diagonally to the vehicle's lengthwise axis to provide lateral axle stability (Figs 10.28(c) and 10.44). These arms form the link between the unsprung suspension members and the sprung chassis/body frame and are therefore able to transmit both driving and braking forces and to absorb the resulting torque reactions.

10.6.2 Panhard rod (Fig. 10.28(b))
Panhard rods, also known as transverse control rods or arms, are positioned across and between both rear wheels approximately parallel to the axle (Fig. 10.28(b)). One end of the rod is anchored to one side of the axle span while the other end is anchored to the body structure; both attachments use either pin or ball type rubber joints. A Panhard rod restrains the body from moving sideways as the vehicle is subjected to lateral forces caused by sidewinds, inclined roads and centrifugal forces when cornering. When the body is lowered, raised or tilted relative to the axle, the Panhard rod is able to maintain an approximate transverse axle alignment (Fig. 10.28(b)) relative to the chassis/body thus relieving the suspension springs from side loads.

10.6.3 Transverse located Watt linkage
(Fig. 10.43(a, b and c))
A Watt linkage (Fig. 10.43) was the original mechanism adopted by James Watt to drive his beam steam engine. This linkage is comprised of two link rods pivoting on the body structure at their outer ends and joined together at their inner ends by a coupler or equalizing arm which is pivoted at its centre to the middle of the rear axle. When in mid-position the link rods are parallel whereas the equalizing arm is perpendicular to both (Fig. 10.43(b)).

If vertical movement of the body occurs either towards bump (Fig. 10.43(c)) or rebound (Fig. 10.43(a)) the end of the link rods will deviate an equal amount away from the central pivot point of the coupling arm. Thus the left hand upper link rod

389
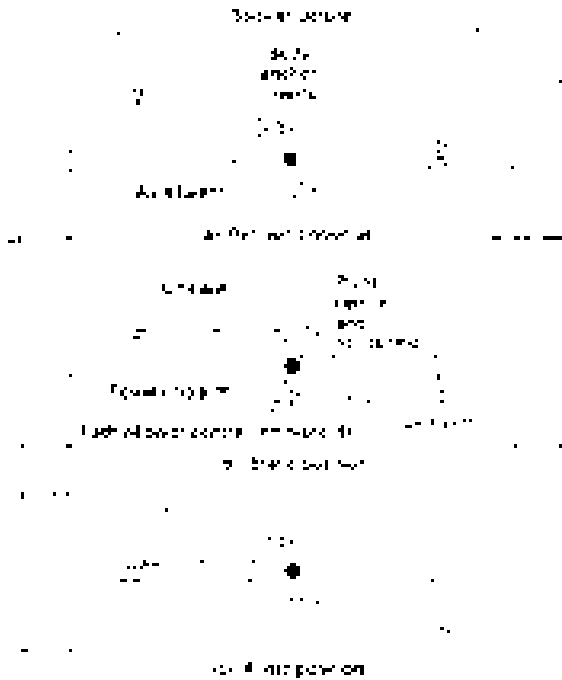
Fig. 10.43 (a–c) Transversely located Watt linkage

will tend to pull towards the left and the right hand lower link rod will apply an equal pull towards the right. The net result will be to force the equalizing arm to rotate anticlockwise to accommodate the inclination to the horizontal of both link rods. If the left hand link rod were made the lower link and the right hand rod the upper link, then the direction of tilt for the equalizing arm would now become clockwise.

For moderate changes in the inclination of the link rods, the body will move in a vertical straight line, thus maintaining a relatively accurate body to axle lateral alignment. Excessive up and down movement of the body will cause the pivot centre to describe a curve resembling a rough figure eight, a configuration of this description being known as a *lemniscoid* (Fig. 10.43(b)).

Under body roll conditions when cornering, the whole body relative to the axle and wheels will be restrained to rotate about the equalizing arm pivot centre at mid-axle height; this point therefore becomes the roll centre for the rear end of the body.

A similar Watt linkage arrangement can be employed longitudinally on either side of the wheels to locate the axle in the fore and aft direction.

10.7 Rear suspension arrangements

10.7.1 Live rigid axle rear suspension

Suspension geometry characteristics of a live axle are as follows:

1 Wheel camber is zero irrespective if the vehicle is stationary or moving round a bend in the road.
2 If one wheel moves over a hump or dip in the road then the axle will tilt causing both wheels to become cambered.
3 Because both wheels are rigidly joined together the wheel track remains constant under all driving conditions.
4 Because the axle casing, half shafts and final drive are directly supported by the wheels, the unsprung weight of a live axle is very high.
5 With a live rigid axle, which is attached to the body by either leaf or coil springs, the body will tilt about some imaginary roll centre roughly mid-way between the upper and lower spring anchorage points.
6 Horizontal fore and aft or lateral body location is achieved by using the leaf springs themselves as restraining members or, in the case of coil springs which can only support the vehicle's vertical load and therefore cannot cope with driving thrust and side loads, horizontally positioned control rods.

Without accurate control of horizontal body movement relative to the axle casing caused by vertical deflection of the springs or longitudinal and transverse forces, the body's weight distribution would be unpredictable which would result in poor road holding and steering response.

Hotchkiss drive suspension (Fig. 10.84(a)) This is the conventional semi-elliptic spring suspension which has each spring positioned longitudinally on each side of the axle and anchored at the front end directly to a spring hanger attached to the body structure and at the rear end indirectly via swing shackle plates to the rear spring hangers, the axle being clamped to the springs somewhere near their mid-span position. Thus fore and aft driving and braking forces are transmitted through the front half of the springs and lateral forces are accommodated by the rigidity of the spring leaves and spring anchorage.

our link coil spring live a le rear suspension (Fig. 10.44) Substituting coil springs for semi-elliptic springs requires a separate means of locating and maintaining body and axle alignment when

390
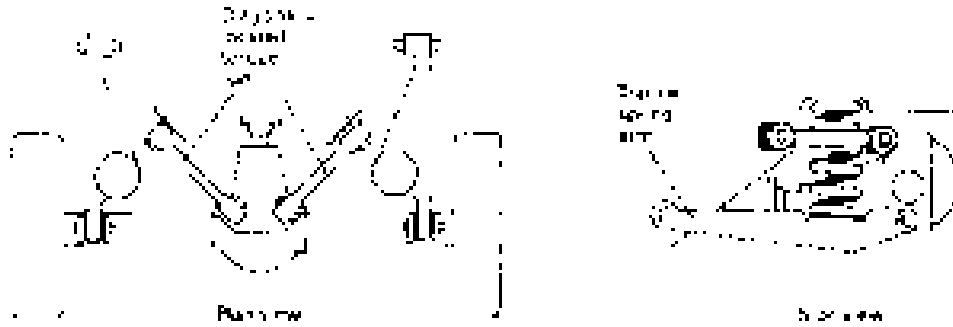
Fig. 10.44 Four link coil spring live axle rear suspension

subjected to longitudinal and transverse forces caused by spring deflection, body roll or driving and braking thrust loads.

The locating links are comprised of a pair of long trailing lower arms and a pair of short diagonally positioned upper torque arms (Fig. 10.44). Rubber pin joints secure the forward ends of the arms to the body structure but the lower rear ends are attached underneath the axle tubes as far apart as possible and the upper short torque arms attached much closer together onto the final drive housing. The coil springs are mounted between the upper body structure and the lower pressed steel trailing arms. These springs only provide vertical support and cannot restrain any horizontal movement on their own. Spring deflection due to a change in laden weight causes both sets of arms to swivel together, thereby preventing the axle assembly rotating and possibly making the universal joints operate with very large angles. Both driving and braking thrust are transmitted through the lower trailing arms which usually are of a length equal to roughly half the wheel track so that when the arms swing the change in wheelbase is small. The upper

arms are normally inclined at 45° to the car's centre line axis so that they can absorb any axle reaction torque tending to rotate the axle, and at the same time prevent relative lateral movement between the body and axle. Body roll or axle tilt are permitted due to the compliance of the rubber pin joints.

A relatively high roll centre is obtained which will be roughly at the upper torque arm height.

orque tube rear wheel drive suspension (Fig. 10.45) One of the major problems with the Hotchkiss drive layout is that the axle torque reaction tends to spin the axle casing when transmitting drive torque in the opposite direction to the rotating wheels and when braking to twist the axle casing in the same direction as the revolving wheels. The result is a considerably distorted semi-elliptic spring and body to axle misalignment. To overcome this difficulty, a rigid tube may be bolted to the front of the final drive pinion housing which extends to the universal joint at the rear of the gearbox or a much shorter tube can be used which is supported at its front end by a rubber pin or ball joint attached to a reinforcing cross-member
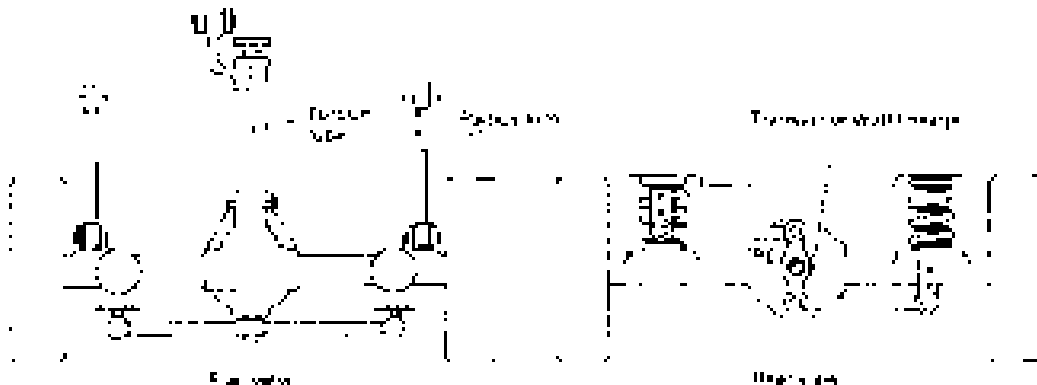
Fig. 10.45 Torque tube with trailing arm and transverse Watt linkage live axle rear suspension

391

(Fig. 10.45). On either side of the torque tube is a trailing arm which locates the axle and also transmits the driving and braking thrust between the wheels and body. Coil springs are mounted vertically between the axle and body structure, their only function being to give elastic support to the vehicle's laden weight. Lateral body to axle alignment is controlled by a transverse Watt linkage. The linkage consists of an equalizing arm pivoting centrally on the axle casing with upper and lower horizontal link arms anchored at their outer ends by rubber pin joints to the body structure. Thus when the springs deflect or the body rolls, the link arms will swing about their outer body location centres causing the equalizing arm to tilt and so restrain any relative lateral body to axle movement without hindering body vertical displacement.

With the transversely located Watt linkage, the body roll centre will be in the same position as the equalizing arm pivot centre. The inherent disadvantages of this layout are still the high amount of unsprung weight and the additional linkage required for axle location.

10.7.2 Non-drive rear suspension
The non-drive (dead) rear axle does not have the drawback of a large unsprung weight and it has the merit of maintaining both wheels parallel at all times. There is still the unwanted interconnection between the wheels so that when one wheel is raised off the ground the axle tilts and both wheels become cambered.

The basic function of a rear non-drive rear suspension linkage is to provide a vertical up and down motion of the axle relative to the body as the springs deflect and at the same time prevent longitudinal and lateral axle misalignment due to braking thrust, crosswinds or centrifugal side force.

ive link coil spring leading and trailing arm att linkage and Panhard rod non-drive a le rear suspension (Fig. 10.46) One successful rigid axle beam and coil spring rear suspension linkage has incorporated a Watt linkage parallel to each wheel to control the axle in the fore and aft direction (Fig. 10.46). A transversely located Panhard rod connected between the axle and body structure is also included to restrict lateral body movement when it is subjected to side thrust.

railing arms with central longitudinal wishbone and anti-roll tube non-drive a le rear suspension (Fig. 10.47) A rectangular hollow sectioned axle beam spans the two wheels and on either side are mounted a pair of coil springs. A left and right hand trailing arm links the axle beam to the body structure via rubber bushed pivot pins located at both ends of the arms at axle level (Fig. 10.47). To locate the axle beam laterally and to prevent it rotating when braking, an upper longitudinal wishbone arm ('A' arm) is mounted centrally between the axle and body structure. The 'A' arm maintains the axle beam spring mounting upright as the spring deflects in either bump or rebound, thus preventing the helical coil springs bowing. It also keeps the axle beam aligned laterally when the body is subjected to any side forces caused by sloping roads, crosswinds and centrifugal force.

Situated just forward of the axle beam is a transverse anti-roll tube welded to the inside of each trailing arm. When body roll occurs while the car is cornering, the inner and outer trailing arms will tend to lift and dip respectively. This results in both trailing arms twisting along their length. Therefore the anti-roll tube, which is at right angles to the arms, will be subjected to a torque which will be resisted by the tube's torsional stiffness. This torsional resistance thus contributes to the coil spring

Fig. 10.46 Five link coil spring leading and trailing arm Watt linkage and Panhard rod dead axle rear suspension
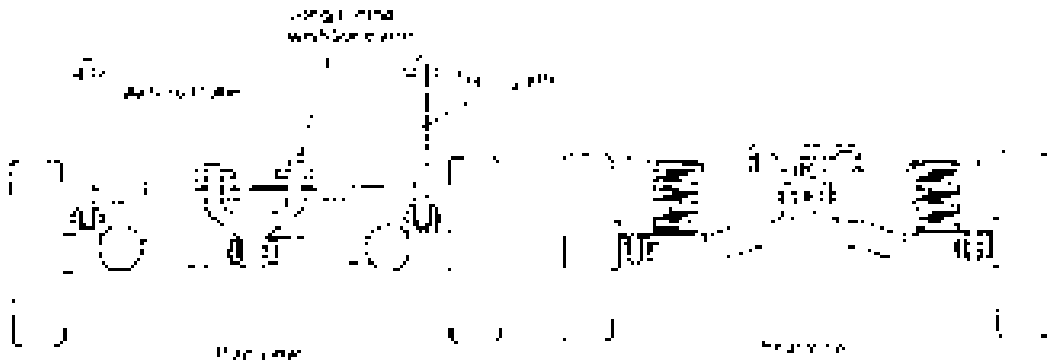
Fig. 10.47 Trailing arm coil spring with central longitudinal wishbone and anti-roll tube dead axle suspension

roll stiffness and increases in proportion to the angle of roll. With this type of suspension the unsprung weight is minimized and the wheels remain perpendicular to the ground under both laden weight and body roll changes.

railing arm and torsion bar spring with non-drive a le rear suspension (Fig. 10.48) The coil springs normally intrude into the space which would be available for passengers or luggage, therefore torsion bar springs transversely installed in line with the pivots of the two trailing arms provide a much more compact form of suspension springing (Fig. 10.48). During roll of the body, and also when the wheels on each side are deflected unequally, the axle beam is designed to be loaded torsionally, to increase the torsional flexibility and to reduce the stress in the material. The axle tube which forms the beam is split underneath along its full length. This acts as an anti-roll bar or stabilizer when the springs are unevenly deflected. The pivot for each trailing arm is comprised of a pair of rubber bushes pressed into each end of a transverse tube which forms a cross-member between the two longitudinal members of the floor structure of the body. The inner surface of the rubber bush is bonded to a hexagonal steel sleeve which is mounted on a boss welded to the outside of the trailing arm. In the centre of the trailing arm boss is a hexagonal hole which receives the similar shaped end of the torsion bar. To prevent relative movement between the male and female joint made between the boss and torsion bar, a bolt locked by a nut in a tapped radial hole in the boss presses against one of the flats on the torsion bar.

One torsion bar spring serves both suspension arms so that a hexagon is forged mid-way between the ends of the bar. It registers in a hexagonal hole formed in the steel collar inserted in and spot
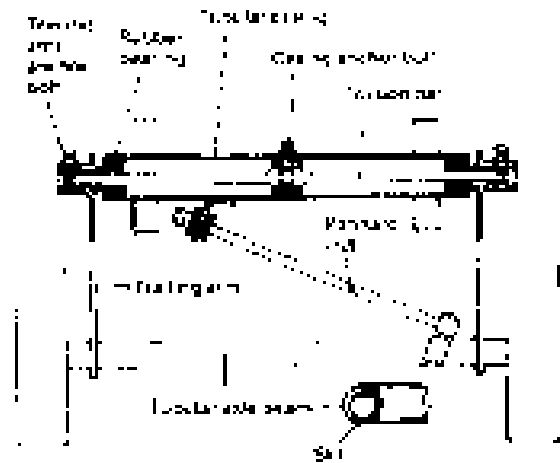
Fig. 10.48 Trailing arm and torsion bar spring with dead axle rear suspension

welded to the transverse tube that houses the torsion bar spring. Again the torsion bar and collar are secured by a radial bolt locked by a nut.

In the static laden position a typical total angular deflection of the spring would be 20° and at full bump about 35°. To give lateral support for the very flexible trailing arms a Panhard rod is diagonally positioned between the trailing arms so that it is anchored at one end to the axle beam and at the other end to the torsion bar tubular casing. All braking torque reaction is absorbed by both trailing arms.

railing arm and coil spring twist a le beam non-drive a le rear suspension (Fig. 10.49(a, b and c)) The pivoting trailing arms are joined together at their free ends by an axle beam comprised of a tubular torsion bar enclosed by a inverted 'U' channel steel section, the ends of the beam being

393
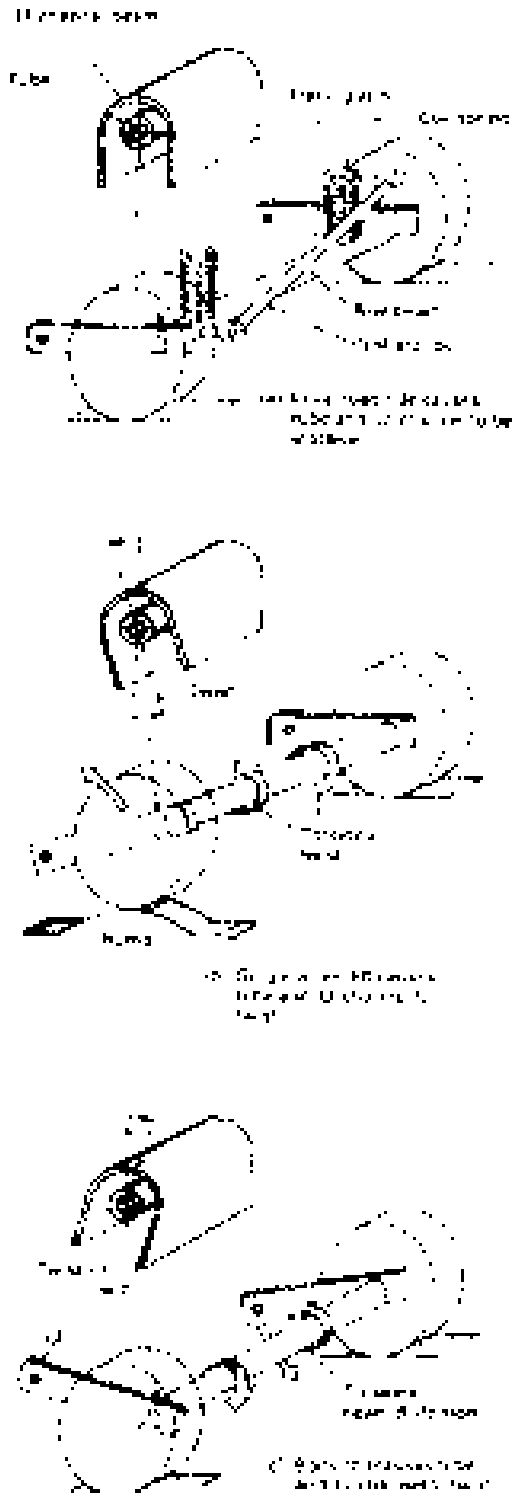
Fig. 10.49 (a–c) Trailing arm twist axle beam rear suspension

butt welded to the insides of the both trailing arms (Fig. 10.49(a, b and c)).

When both wheels are deflected an equal amount, caused by increased laden weight only, the coil springs are compressed (Fig. 10.49(a)). If one wheel should be raised more than the other, its corresponding trailing arm rotates about its pivot causing the axle beam to distort to accommodate the difference in angular rotation of both arms (Fig. 10.49(b)). Consequently the twisted axle beam tube and outer case section will transfer the torsional load from the deflected trailing arm to the opposite arm. This will also cause the undeflected arm to rotate to some degree, with the result that the total body sway is reduced.

During cornering when the body rolls, the side of the body nearest the turn will lift and the opposite side will dip nearer to the ground (Fig. 10.49(c)). Thus the inner trailing arm will be compelled to rotate clockwise, whereas the outer trailing arm rotates in the opposite direction anticlockwise. As a result of this torsional wind-up of the axle beam, the outer wheel and trailing arm will tend to prevent the inner trailing arm from rotating and lifting the body nearest the turn. Hence the body roll tendency will be stabilized to some extent when cornering.

With this axle arrangement much softer coil springs can be used to oppose equal spring deflection when driving in the straight ahead direction than could otherwise be employed if there were no transverse interconnecting beam.

Strut and link non-drive rear independent suspension (Fig. 10.50) With this suspension the wheel hub carrier's up and down motion is guided by the strut's sliding action which takes place between its piston and cylinder. The piston rod is anchored by a rubber pivot to the body structure and the cylinder member of the strut is rigidly attached to the wheel hub carrier (Fig. 10.50). A transverse link (wishbone arm) connects the lower part of the hub carrier to the body, thereby constraining all lateral movement between the wheels and body. The swing link arm and sliding strut member's individual movements combine in such a way that the hub carrier's vertical motion between bump and rebound produces very little change to the static wheel camber, either when the laden weight alters or when cornering forces cause the body to roll.

Braking fore and aft inertia forces are transmitted from the body to the hub carrier and wheel by trailing radius arms which are anchored at their
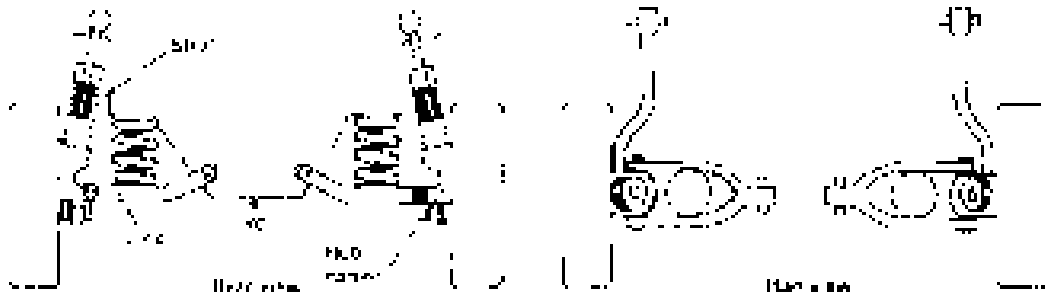
Fig. 10.50 Strut and link non-drive independent rear suspension

forward ends by rubber pin joints to the body under-structure. Owing to the trailing radius arms being linked between the body and the underside of each wheel hub carrier, deflection of the coil springs will cause a small variation in wheel toe-in to occur between the extremes in vertical movement.

The positioning of the body roll centre height will be largely influenced by the inclination of the swing arm relative to the horizontal; the slope of these transverse arms are usually therefore chosen so that the roll centre height is just above ground level.

10.7.3 Rear wheel drive suspension

Swing arm rear wheel drive independent suspension (Fig. 10.51) This suspension normally takes the form of a pair of triangular transverse ('A' arm) swing arm members hinging on inboard pivot joints situated on either side of the final drive with their axes parallel to the car's centre line (Fig. 10.51). Coil springs are mounted vertically on top of the swing arm members near the outer ends. The wheels are supported on drive hubs mounted on ball or tapered roller bearings located within the swing arm frame.

Each drive shaft has only one universal joint mounted inboard with its centre aligned with that of the swing arm pivot axes. If the universal joints

and swing arm pivot axes are slightly offset (above and below in diagram), the universal joints must permit a certain amount of sliding action to take place to compensate for any changes in drive shaft length as the spring deflects. Usually the outer end of the drive shaft forms part of the stub axle wheel hub.

Any increase in static vehicle weight causes the swing arms to dip so that the wheels which were initially perpendicular to the road now become negatively cambered, that is, both wheels lean towards the body at the top. Consequently, when the body rolls during cornering conditions, the inner and outer wheels relative to the turn become cambered negatively and positively respectively; they both lean towards the centre of rotation. With a change in static vehicle weight both swing arms pivot and dip an equal amount which reduces the wheel track width. Similarly, if the body rolls the inner swing arm pivot centre rises and the outer swing arm pivot drops, so in fact both the swing arm pivots tend to rotate about their roll centres thus reducing the width of the wheel track again. Both wheels at all times will remain parallel as there is no change in wheel toe-in or -out.

Low pivot split a le coil spring rear wheel drive independent suspension (Fig. 10.52) The conventional transverse swing arm suspension suffered from three major limitations:

Fig. 10.51 Transverse swing arm coil spring rear wheel drive independent suspension

Fig. 10.52 Low pivot split axle coil spring rear wheel drive suspension

1 The swing arms were comparatively short because the pivot had to be mounted on either side of the final drive housing; it therefore caused a relatively large change in wheel camber as the car's laden weight increased or when wheel bounce occurred.
2 Due to the projection lines extending from the tyre to ground centre contact to and beyond the swing arm pivot centres, the body roll centre with this type of suspension was high.
3 There was a tendency when cornering for the short swing arms to become jacked up and with the load concentrated on the outside, the highly positively cambered wheel reduced its ability to hold the road so that the rear end of the car was subjected to lateral breakaway.

To overcome the shortcomings of the relatively large change in wheel camber and the very high roll centre height, the *low pivot split axle suspension* was developed.

With this modified swing axle arrangement the axle is split into two, with the adjacent half-axles hinged on a common pivot axis below the final drive housing (Fig. 10.52). A vertical strut supports the final drive assembly; at its upper end it is mounted on rubber discs which bear against the rear cross-member and at its lower end it is anchored to a pin joint situated on the hinged side of the final drive pinion housing. The left hand half-axle casing houses a drive shaft, crownwheel and differential unit. A single universal joint is positioned inside the casing so that it aligns with the pivot axis of the axles. The right hand half-axle houses its own drive shaft and a rubber boot protects the final drive assembly from outside contamination, such as dirt and water. A horizontal arm forms a link between the pivot axis and body structure and controls any lateral movement of the body relative to the axles. Fore and aft support for each half-axle is given by trailing radius arms which also carry the vertically positioned coil springs. The body roll centre thus becomes the pivot axis for the two half-axles which is considerably lower than for the conventional double pivot short swing arm suspension.

railing arm rear wheel drive independent suspension (Fig. 10.53) The independent trailing arm suspension has both left and right hand arms hinged on an axis at right angles to the vehicle centre line (Fig. 10.53). Each arm, which is generally semi-triangular shaped, is attached to two widely spaced pivot points mounted on the car's rear subframe. Thus the trailing arms are able to transfer the drive thrust from the wheel and axle to the body structure, absorb both drive and braking torque reactions and to restrain transverse body movement when the vehicle is subjected to lateral forces. The
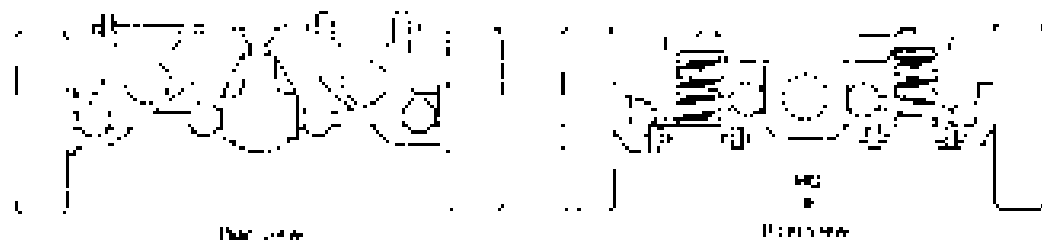
Fig. 10.53 Trailing arm coil spring rear wheel drive independent suspension

396

rear ends of each arm support a live wheel hub, the drive being transmitted from the final drive to each wheel via drive shafts and inner and outer universal joints to accommodate the angular deflection of the trailing arms. The inner joints also incorporate a sliding joint to permit the effective length of the drive shafts to vary as the trailing arms articulate between bump and rebound.

When the springs deflect due to a change in laden weight, both wheels remain perpendicular to the ground. When the body rolls on a bend, the inner wheel becomes negatively cambered and the outside wheel positively cambered; both wheels lean away from the turn. Spring deflection, caused by either an increase in laden weight or wheel impact, does not alter the wheel track toe-in or -out or the wheel track width, but body roll will cause the wheel track to widen slightly.

Semi-trailing arm rear wheel drive independent suspension (Fig. 10.54) With the semi-trailing arm suspension each arm pivots on an axis which is inclined (skewed) to something like 50 to 70 degrees to the car's centre line axis (Fig. 10.54). The pivot axes of these arms are neither transverse nor longitudinally located but they do lie on an axis which is nearer the trailing arm pivot axis (which is at right angles to the car's centre line axis). Consequently the arms are classified as semi-trailing.

Swivelling of these semi-trailing arms is therefore neither true transverse or true trailing but is a combination of both. The proportion of each movement of the semi-trailing arm will therefore depend upon its pivot axis inclination relative to the car's centre line. With body roll the transverse swing arm produces positive camber on the inside wheel and negative camber on the outer one (both wheels lean inwards when the body rolls), whereas with a trailing arm negative camber is produced on the inside wheel and positive camber on the outer one (both wheels lean outwards with body roll).

Skewing the pivot axis of the semi-trailing arm suspension partially neutralizes the inherent ten-

dencies when cornering for the transverse swing arm wheels to lean towards the turn and for the trailing arm wheels to lean away from the turn. Therefore the wheels remain approximately perpendicular to the ground when the car is subjected to body roll.

Because of the relatively long effective swing arm length of the semi-trailing arm, only a negligible change to negative camber on bump and positive camber on rebound occurs when both arms deflect together. However, there is a small amount of wheel toe-in produced on both inner and outer wheels for both bump and rebound arm movement, due to the trailing arm swing action pulling the wheel forward as it deflects and at the same time the transverse arm swing action tilting the wheel laterally.

By selecting an appropriate semi-trailing arm pivot axis inclination, an effective swing arm length can be produced to give a roll centre height somewhere between the ground and the pivot axis of the arms. By this method the slip angles generated by the rear tyres can be adjusted to match the understeer cornering characteristics required.

Transverse double link arm rear wheel drive independent suspension (Figs 10.55 and 10.56) This class of suspension may take the form of an upper and lower wishbone arm linking the wheel hub carrier to the body structure via pivot joints provided at either end of the arms. Drive shafts transfer torque from the sprung final drive unit to the wheel hub through universal joints located at the inner and outer ends of the shafts. Driving and braking thrust and torque reaction is transferred through the wide set wishbone pivot joints. One form of transverse double link rear wheel drive independent suspension uses an inverted semi-elliptic spring for its upper arm (Fig. 10.55).

A double wishbone layout has an important advantage over the swing axle and trailing arm arrangements in that the desired changes of wheel camber, relative to motions of the suspension, can

Fig. 10.54 Semi-trailing arm coil spring rear wheel drive independent suspension
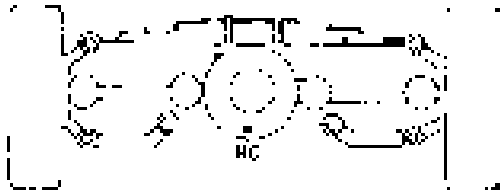
Fig. 10.55 Transverse swing arm and inverted semi-elliptic spring rear wheel drive independent suspension

Fig. 10.56 Transverse swing arm and double universal joint load bearing drive shaft rear independent suspension

be obtained more readily. With swing axles, camber changes tend to be too great, and the roll centre too high. Wheels located by trailing arms assume the inclination of the body when it rolls, thereby reducing the cornering forces that the tyres produce. Generally, transverse double link arm suspensions are designed to ensure that, when cornering, the outer wheel should remain as close to the vertical as possible.

A modified version (Fig. 10.56) of the transverse double link suspension comprises a lower transverse forked tubular arm which serves mainly to locate the wheel transversely; longitudinal location is provided by a trailing radius arm which is a steel pressing connecting the outer end of the tubular arm to the body structure. With this design the upper transverse link arm has been dispensed with, and a fixed length drive shaft with Hooke's

universal joints at each end now performs the task of controlling the wheel hub carrier alignment as the spring deflects. Compact twin helical coil springs are anchored on both sides of the lower tubular forked arms with telescopic dampers positioned in the middle of each spring.

e ion a le rear wheel drive suspension (Figs 10.57 and 10.58)   The DeDion axle is a tube (sometimes rectangular) sectioned axle beam with cranked (bent) ends which are rigidly attached on either side to each wheel hub. This permits the beam to clear the final drive assembly which does not form part of the axle beam but is mounted independently on the underside of the body structure (Figs 10.57 and 10.58).

To attain good ride characteristics the usual sliding couplings at the drive shaft to the wheels are dispensed with in this design since when transmitting drive or braking torque, such couplings generate considerable frictional resistance which opposes the sliding action. A sliding joint is provided in the axle tube to permit wheel track variation during suspension movement (Fig. 10.57). Axle lateral location is therefore controlled by the drive shafts which are permitted to swing about the universal joint centres but are prevented from extending or contracting in length. Fore and aft axle location is effected by two Watt linkages. These comprise two lower trailing fabricated pressed steel arms, which also serve as the lower seats for the coil springs. Their rear ends are carried on pivots below the hub carriers. The other parts of the Watt linkage consist of two rearward extending tubular arms, each attached to a pivot above the hub carrier. The upper and lower unequal length link arm pivot centres on the body structure are arranged in such a way that the axle has a true vertical movement as the spring deflects so that there are no roll steer effects. When the body rolls
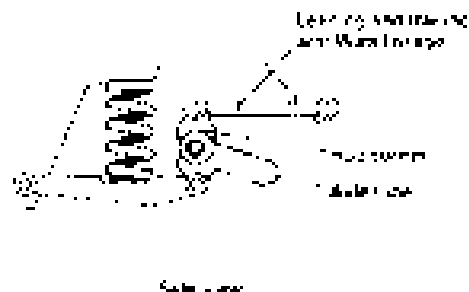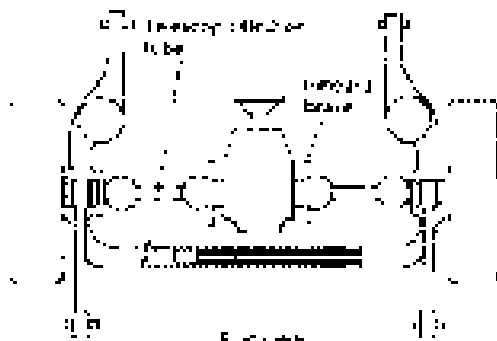
Fig. 10.57 DeDion axle with leading and trailing arm Watt linkage rear suspension
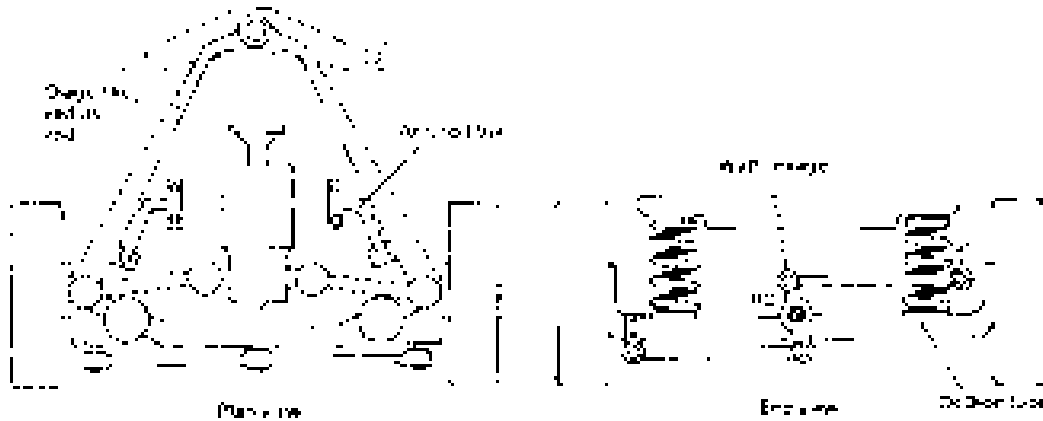
Fig. 10.58 DeDion tube with diagonal radius arms and Watt transverse linkage rear suspension

one hub carrier tends to rotate relative to the other, which is permitted by the sliding joint in the axle tube. The inner and outer sliding joints of the axle tube are supported on two widely spaced bronze bushes. The internal space between the inner and outer axle tube is filled about two thirds full of oil and lip seals placed on the outboard end of each bearing bush prevents seepage of oil. A rubber boot positioned over the axle sliding joint prevents dirt and water entering between the inner and outer tube members.

A DeDion axle layout reduces the unsprung suspension weight for a rear wheel drive car, particularly if the brakes are situated inboard. It keeps both road wheels parallel to each other under all driving conditions and transfers the driving and braking torque reactions directly to the body structure instead of by the conventional live axle route by way of the axle casing and semi-elliptic springs or torque rods to the body. The wheels do not remain perpendicular to the ground when only one wheel lifts as it passes over a hump or dip in the road. The body roll centre is somewhere near the mid-height position of the wheel hub carrier upper and lower link arm pivot points; a typical roll centre height from the ground would be 316 mm.

An alternative DeDion axle layout forms a triangle with the two diagonal radius arms which are rigidly attached to it (Fig. 10.58). The apex where the two radius arms meet is ahead of the axle and is pivoted by a ball joint to the body cross-member so that the driving and braking thrust is transferred from the axle to the body structure via the diagonal arms and single pivot. A transverse Watt linkage mounted parallel and to the rear of the axle beam controls lateral body movement relative to the axle. Therefore the body is constrained to roll on an axis

which passes between the front pivot supporting the radius arms and the central Watt linkage pivot to the rear of the axle.

The sprung final drive which is mounted on the underside of the rear axle arch transmits torque to the unsprung wheels by way of the drive shaft and their inner and outer universal joints. The effective length of the drive shaft is permitted to vary as the suspension deflects by adopting splined couplings or pot type joints for both inner universal joints.

10.8 Suspension design consideration

10.8.1 Suspension compliance steer
(Fig. 10.59(a and b))
Rubber bush type joints act as the intermediates between pivoting suspension members and the body to reduce the transmission of road noise from the tyres to the body. The size, shape and rubber hardness are selected to minimize noise vibration and ride hardness by operating in a state of compressive or torsional distortion.

If the rubber joints are subjected to any abnormal loads, particularly when the suspension pivots are being articulated, the theoretical geometry of the swing members may be altered so that wheel track misalignment may occur.

The centrifugal force when cornering can produce lateral accelerations of 0.7 to 8.0 g which is sufficient to compress and distort the rubber and move the central pin off-centre to the outer hole which supports the rubber bush.

With transverse or semi-trailing arms suspension (Fig. 10.59(a)) the application of the brakes retards the rotation of the wheels so that they lag behind the inertia of the body mass which is still trying to
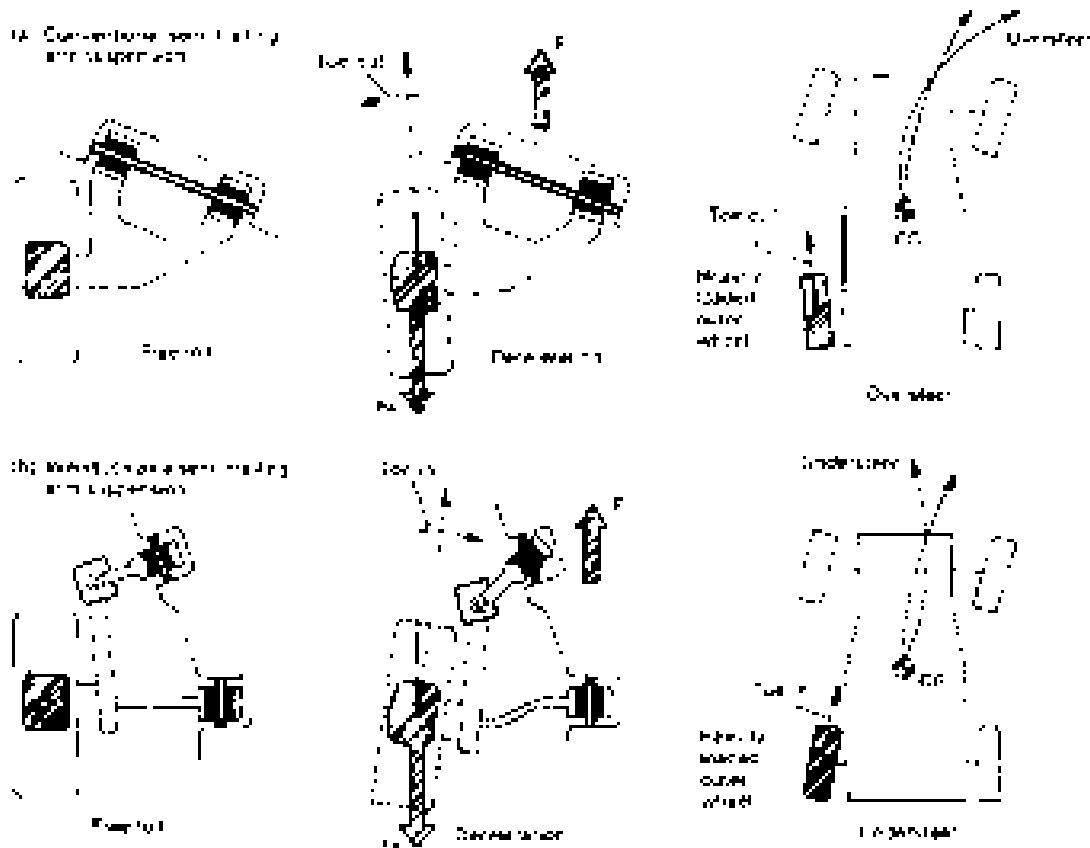
Fig. 10.59 (a and b) Semi-trailing suspension compliance steer

thrust itself forward. Consequently the opposing forces between the body and suspension arms will distort the rubber joint, causing the suspension arms to swing backwards and therefore make the wheel track toe outwards.

The change in the wheel track alignment caused by the elastic deflection of the suspension rubber pivot joints is known as *suspension compliance steer* since it introduces an element of self-steer to vehicle.

Compliance steer is particularly noticeable on cornering if the brakes are being applied since the heavily loaded outside rear wheel and suspension is then subjected to both lateral forces and fore and aft force which cause an abnormally large amount of rubber joint distortion and wheel toe-out (Fig. 10.59(a)), with the result that the steering will develop an unstable oversteer tendency.

A unique approach to compliance steer is obtained with the Weissuch axle used on some Porsche cars (Fig. 10.59(b)). This rear transverse upper and lower double arm suspension has an additional lower two piece link arm which takes

the reaction for both the accelerating and decelerating forces of the car. The lower suspension links consist of a trailing tubular steel member which carries the wheel stub axle and the transverse steel plate arm. The trailing member has its front end pivoted to a short torque arm which is anchored to the body by a rubber bush and pin joint pivoted at about 30° to the longitudinal car axis. When the car decelerates the drag force pulls on the rubber bush pin joint (Fig. 10.59(b)) so that the short torque arm is deflected backward. As a result, the transverse steel plate arm distorts towards the rear and the front end of the trailing tubular member supporting the wheel is drawn towards the body, thus causing the wheel to toe-in. Conversely, when the car is accelerated the wheel tends to toe-out, but this is compensated by the static (initial) toe-in which is enough to prevent them toeing-out under driven conditions. The general outcome of the lower transverse and trailing link arm deflection is that when cornering the more heavily loaded outside wheel will toe-in and therefore counteract

400

some of the front wheel steer, thus producing a degree of understeer.

10.8.2 Suspension roll steer
(Fig. 10.60(a, b and c))
When a vehicle is cornering the body tilts and therefore produces a change in its ground height between the inside and outside wheels. By careful design, the suspension geometry can be made to alter the tracking direction of the vehicle. This self-steer effect is not usually adopted on the front suspension as this may interfere with steering geometry but it is commonly used for the rear suspension to increase or decrease the vehicle's turning ability in proportion to the centrifugal side force caused by cornering. Because it affects the steering handling characteristics when corner-ing it is known as *roll oversteer* and *roll understeer* respectively.

Roll steer can be designed to cancel out large changes in tyre slip angles when cornering, particu-larly for the more heavily loaded outer rear wheel since the slip angle also increases roughly in pro-portion to the magnitude of the side force.

The amount of side force created on the front or rear wheels is in proportion to the load distribution on the front and rear wheels. If the car is lightly laden at the front the rear wheels generate a greater slip angle than at the front, thus producing an oversteer tendency. When the front is heavily loaded, the front end has a greater slip angle and so promotes an understeer response.

The object of roll steer on the rear wheels is for the suspension geometry to alter in such a way that
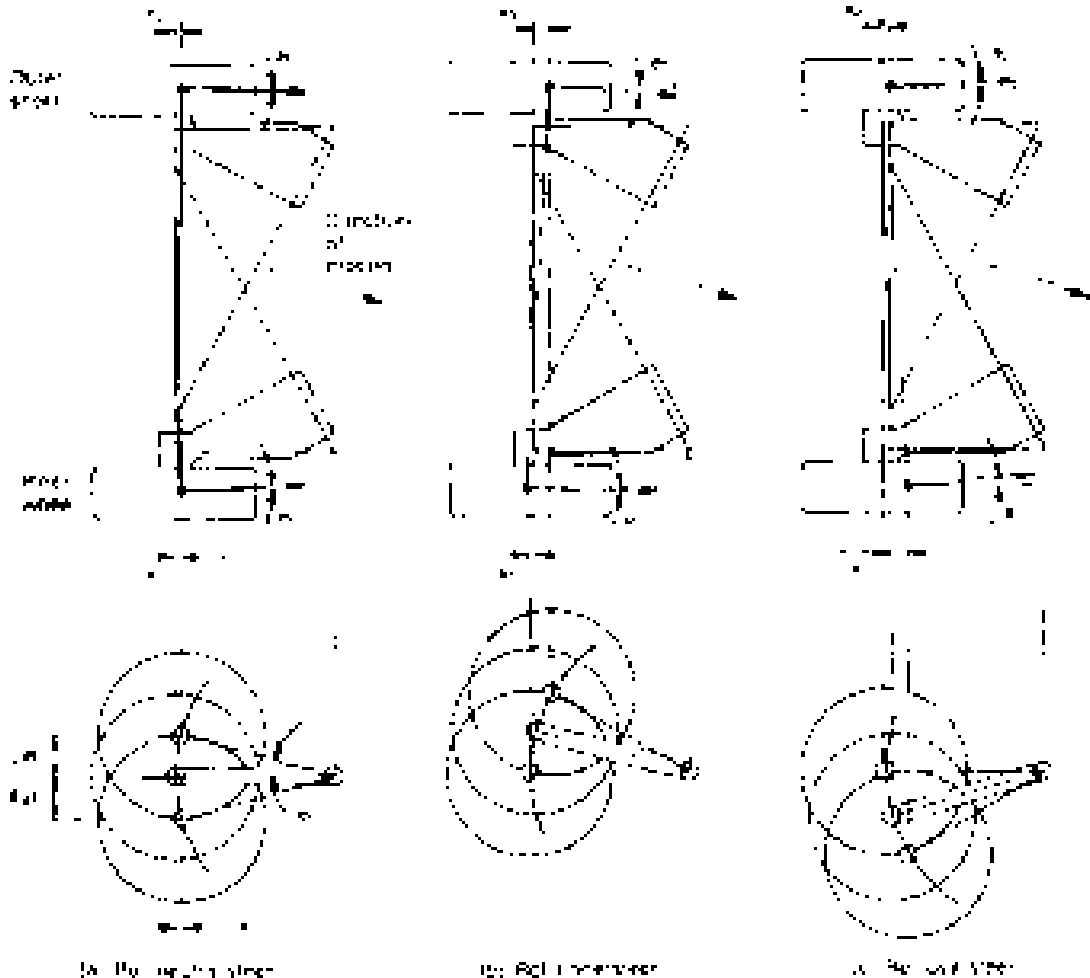
Fig. 10.60 (a–c) Semi-trailing suspension roll steer

the rear wheels steer the back end of the vehicle either outwards or inwards to compensate for the deviation in directional steer caused by changes in tyre slip angle.

A good example which illustrates suspension roll steer is with the semi-trailing arm steer rear suspension (Fig. 10.60(a, b and c)). If the body tilts when the vehicle corners the arms swing about their pivots so that the wheel axle attached to their free ends scribes circular arcs as they deflect up or down.

When the body rolls with the trailing arms set horizontally in their static position (Fig. 10.60(a)), the outer wheel and arm swings upwards towards the body whereas the inner wheel and arm rotates downwards and away from the body.

The consequence of the movement of the arms is that both axles move forward a distance x but because the axles of both trailing arms pivot at an inclined angle to the central axis of the vehicle the axis of end wheel axle will be slightly skewed inward so that both wheels now toe-in.

If the static position of the trailing arms were now set upwards an angle from the horizontal (Fig. 10.60(b)), when the body rolls the outer wheel and arm swing further upwards, whereas the inner wheel swings in the opposite direction (downwards towards the horizontal position). The outcome is that the outer wheel axle moves forwards whereas the inner wheel axle moves slightly to the rear. As a result, both the outer and inner stub axles skew the wheels towards the turn so that the outer wheel track toes-in and the inner wheel toes-out. Thus the change in tracking would tend to counteract any increase in slip angle due to cornering and so cause more understeer.

Setting the trailing arm static position so that both arms are inclined downwards an angle from the horizontal (Fig. 10.60(c)) produces the opposite effect to having an upward tilt to the trailing arms. With body roll the outer wheel and arm now swings towards the horizontal and moves backwards slightly whereas the inner wheel and arm pivots further downward and moves forwards. Consequently both wheels are skewed outward from the turn, that is, the inner wheel toes-in and the outer wheel toes-out. The tracking in this situation compounds the increase in slip angle which is experienced while cornering and therefore produces an oversteer tendency.

10.8.3 Anti-dive and squat suspension control
(Fig. 10.61)
All vehicles because of their suspended mass suffer from weight transfer when they are either acceler-
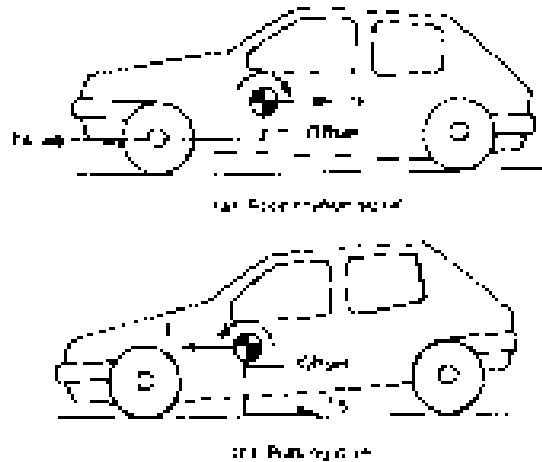
Fig. 10.61 (a and b)   Vehicle squat and dive

ated, as when pulling away from a standstill, or when retarding while being braked.

A vehicle driven from a standstill (Fig. 10.61(a)) experiences a rapid change in speed in a short time interval so that a large horizontal accelerating force F_A is delivered at axle level to overcome the opposing body's inertia force F_I which acts in the opposite sense through the centre of gravity but which is generally situated well above axle height somewhere between the two axles. Due to the vertical offset distance between the accelerating force F_A and the inertia reaction force F_I, a pitch moment will be produced which transfers weight from the front to the rear wheels as the front of the car lifts and the rear sinks, thereby making the car body squat at the rear.

Likewise weight transfer occurs from the rear to the front wheels when the vehicle is braked (Fig. 10.61(b)) which causes the body to pitch forward so that the rear rises and the front suspension dips, which gives a front nose dive appearance to the vehicle. The forces involved when braking are the ground level retarding frictional force F_B and the inertia reaction force F_I at the centre of gravity height. Therefore there is a larger offset between the two opposing forces when braking than when accelerating because with the latter the driving force acts at axle level. Consequently when the brakes are applied, the offset opposing retarding frictional force and the inertia reaction force produce a couple which attempts to make the body pitch and dive towards the front.

A leading and trailing arm suspension layout can be designed to counteract both squat (Figs 10.62 and 10.63) and dive (Fig. 10.64) tendencies.

When the vehicle accelerates forwards, the reaction to the driving torque pivots the suspension arm about the axle in the opposite direction to the input driving torque. Thus in the case of a front wheel drive vehicle (Fig. 10.62) the arm swings downwards and opposes the front upward lift caused by the reluctant inertia couple. Likewise with a rear wheel drive vehicle (Fig. 10.63) the reaction to the driving torque swivels the suspension arm upward and so resists the rearward pitch caused by the weight transference from the front to the rear axle.

For both drive acceleration and braking the amount of squat and dive is controlled by the length of the leading and trailing arms. The shorter they are, the greater their resistance to weight transference will be, and from that point of view alone, the better the quality of ride will be.

A large number of modern suspensions are based on trailing or semi-trailing arm designs which can build in anti-squat and -dive control but leading arm front suspension has inherent undesirable features and therefore is rarely used. However, anti-squat and -dive control can be achieved by producing a virtual lead arm front suspension, that is, by arranging the swing axis of a double wishbone arm suspension to converge longitudinally along the wheelbase at some point.

The double transverse wishbone arm suspension geometry (Fig. 10.65) is laid out so that the top wishbone arm axis tilts downwards and the lower slightly upward towards the rear so that lines drawn through these pivot axes intersect somewhere towards the rear.

When the brakes are applied, the body will tend to pitch downward at the front but the clamped disc caliper or back plate will attempt to rotate with the road wheel. The result is that the upper and lower wishbone pivot axis converging projections form in effect an imaginary leading arm of length R which tends to swing upwards to the rear about the wheel axle. It therefore imparts an upthrust which opposes and cancels the downward pitch of the body.

Similar results can be obtained with the MacPherson strut suspension (Fig. 10.66) where the strut is made to tilt backward from the top and the lower transverse wishbone arm pivot axis tilts upwards to the rear. A line drawn perpendicular to the strut through the top pivot will then intersect a line projecting from the wishbone pivot axis. The distance between the strut to wishbone ball pivot and the meeting point of the two rearward projected lines therefore provides the effective trailing arm length or swing radius R Thus an anti-dive torque T is produced of magnitude FR which opposes the forward transfer of weight when braking.

Fig. 10.62 Leading and trailing arm front wheel drive anti-squat suspension action
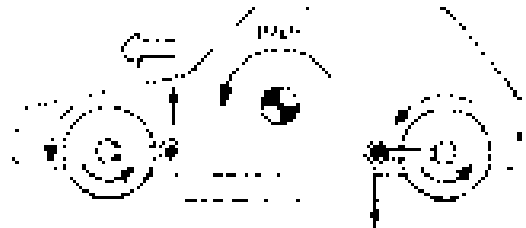
Fig. 10.64 Leading and trailing arm brake anti-drive suspension

Fig. 10.63 Leading and trailing arm rear wheel drive anti-squat suspension action

Fig. 10.65 Transverse double wishbone suspension with longitudinal converging axis geometry
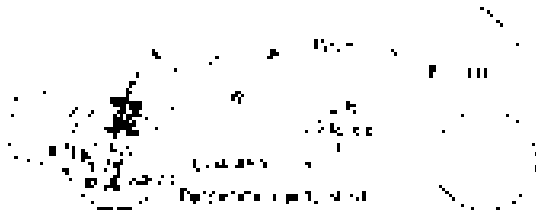
Fig. 10.66 MacPherson strut suspension with longitudinal converging axis geometry

Unfortunately the amount of anti-dive control must be limited since the upward swing of the imaginary trailing arm rotates the steering swivel joints so that the castor angle changes from positive to negative, thus destabilizing the steering firmness and so producing steering reaction and wander. Normally front suspension design restricts the anti-dive control to within 50 to 70% and the rear suspension may provide a 100% cancellation of brake dive.

10.8.4 Front wheel drive independent suspension wheel bearing arrangements (Figs 10.67 and 10.68)

With a front wheel drive independent suspension two major functions must be fulfilled:

1 The wheels must be able to turn about their swivel pins simultaneously as the suspension members deflect between bump and rebound.
2 The transmission of drive torque from the final drive to the wheels must be uninterrupted as the suspension members move between their extremes.

The majority of steered independent suspensions incorporate a wheel hub carrier supported between either;

a) an upper and lower ball and socket joint mounted between a pair of transverse arms (Fig. 10.67),
b) a leg strut mounted on a swivel bearing and a lower ball and socket joint located at the free end of a transverse arm (Fig. 10.68).

In both suspension arrangements the hub carrier has a central bore which may directly or indirectly house the wheel hub bearings. For light and medium loads, roller ball bearings are preferred but for heavy duty applications the taper roller bearing is more suitable.

Traditional wheel bearing assemblies employ two separate bearings; either ball or taper roller
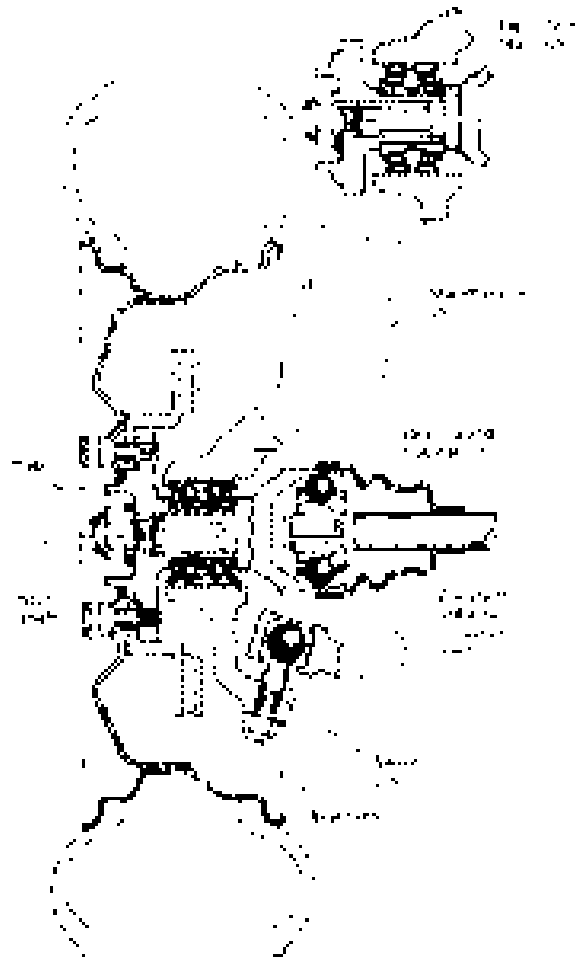
Fig. 10.67 Front wheel drive MacPherson strut suspension with single double row ball or roller wheel bearings

types. The present trend is the use of a single bearing with double row rolling elements, be they ball or taper rollers which are sealed, pre-set and lubricated for life. The preference is because they provide a more compact and cheaper assembly.

These double row rolling element single bearings can be of the following classes:

1 *etachable double row angular contact ball or taper roller bearing type* (Fig. 10.67). There are two separate inner track rings and one wide outer track ring. The contact angle for the balls is 32° to give the greatest distance between pressure centres of the bearing, thus reducing the reactions caused by the tilting action of the wheels. This angle is so chosen that the bearing
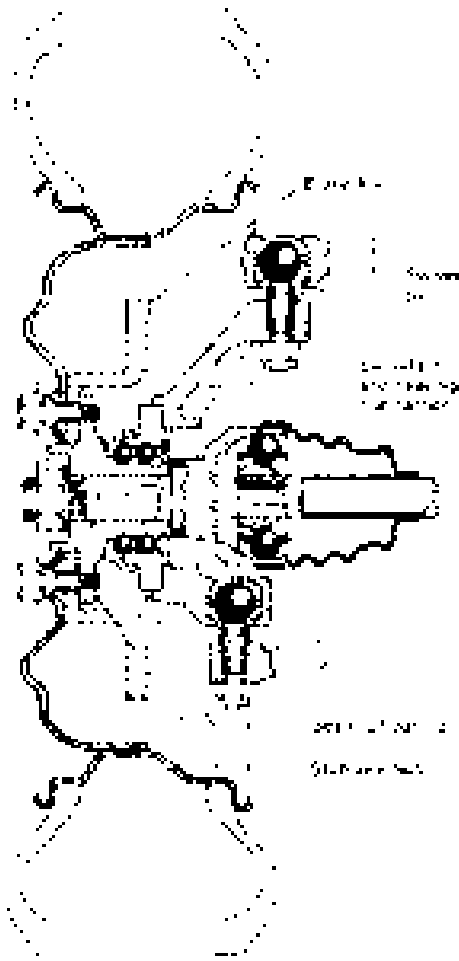
Fig. 10.68 Front wheel drive transverse wishbone suspension with fully integrated double low wheel bearings

has sufficient radial load capacity to withstand the weight imposed on the wheel and also to provide adequate axial load carrying capacity under cornering conditions. The cage that separates the balls is made from Nylon and does not, if damaged, affect the bearing performance. Preloading of the ball or taper roller bearings is set at the factory, therefore no adjustment is required after the bearing is assembled to its hub.

When assembled, the inner track rings are a force fit over the hub sleeve which is internally splined to the constant velocity joint's output stub shaft and the outer track ring is a press fit inside the hub carrier bore.

2 *Fully integrated double row angular contact ball bearing type* (Fig. 10.68). With this arrangement

the inner track ring is extended on the outside with a flange to locate and support the wheel while its middle is bored and splined to accommodate the constant velocity joint splined output shaft. Thus the inner bearing member (track ring) takes over the whole function of the normal drive wheel hub. The outer track ring also supports both rows of balls and it is enlarged in the centre to provide a flange which aligns accurately within the wheel hub carrier's bore. Thus the inner and outer bearing members are integral parts of the wheel hub and bearing housing attached to the hub carrier respectively.

In both bearing arrangements the stub shaft nut is fully tightened to prevent axial movement between the hub and stub shaft and also, in the case of the detachable double row bearing, to secure its position.

10.9 Hydrogen suspension

10.9.1 Hydrogen interconnected suspension
(oulton Dunlop) (Fig. 10.69(a, b and c))
The spring unit is comprised of a nitrogen filled spherical spring chamber welded to a double conical shaped displacement chamber (Fig. 10.69(a, b and c)). A hydraulic damper in the form of a pair of rubber compression blocks separates both spherical spring and displacer chambers, its function being to control the flow of fluid as it passes to and fro between the two chambers. The displacer chamber is sealed at its lower end by a load absorbing nylon reinforced rubber diaphragm which rolls between the conical piston and the tapered displacer chamber skirt as the suspension deflects up and down when the wheels pass over any irregularities on the road surface.

Within the spherical spring chamber is a butyl-rubber diaphragm which separates the sphere into a nitrogen charged (17.5 bars) upper region (the spring media) which is sealed for life, and the lower region which is filled with fluid. Initially fluid is pumped into the displacer chamber until it reaches the nitrogen charging pressure. Then it will compress and lift the separator diaphragm off the bottom of the sphere. Since the gas and fluid pressures on both sides of the diaphragm are equal, the separator diaphragm is not subjected to heavy loads, in fact it only functions as a flexible wall to keep the gas and fluid apart. A water based fluid containing 50% industrial alcohol and a small percentage of anti-corrosion additive is pumped into
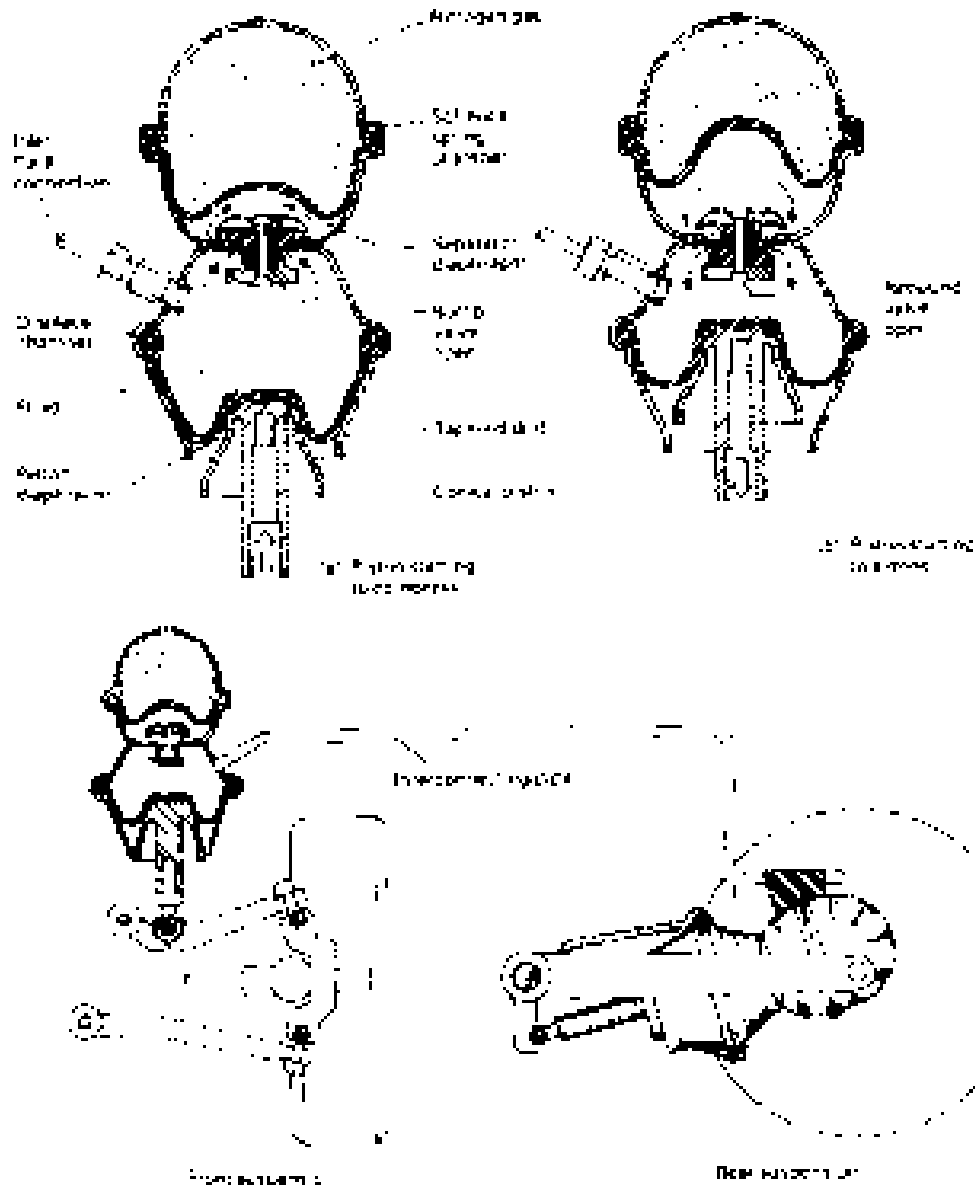
Fig. 10.69 (a–c) Hydrogas interconnected suspension system

the system to a pressure of 23 bars with the car in the unladen state, this being the condition in which the car's body to ground height is checked.

One advantage in using a rolling diaphragm type displacer instead of a piston and cylinder is that a water based fluid can be utilized as opposed to an oil which would not have such stable viscosity characteristics.

aper rate (Fig. 10.69(a and b)) The effective area of piston compressing the fluid is that projected area of the displacer diaphragm which is not supported by the internal tapered skirt of the displacer chamber. Therefore, as the load on the displacer piston increases and the piston is pushed further into the chamber, less of the displacer diaphragm will be supported by the chamber's skirt and more

will form part of the projected effective piston area. The consequence of the diaphragm piston contracting within the displacer chamber is that the load-bearing area of the piston is increased due to the diaphragm rolling away from its supporting tapered chamber skirt. As a result the resistance offered by the fluid against the inward movement of the piston rises. In other words, due to the tapered chamber's skirt, the spring rate (stiffness) increases in proportion to the spring's deflection.

Spring compressing due to bump response (Fig. 10.69(a)) When the tyre of the wheel hits a hump in the road, the whole wheel assembly attached to the suspension rises rapidly. This causes the displacer piston to move further into the displacer chamber. Consequently fluid in the displacer chamber will be displaced and pushed into the spherical spring chamber via the transfer port and bump valve. The rapid transfer of fluid into the spring chamber compresses the separator diaphragm against the nitrogen gas and the resilience of the gas therefore absorbs the impact shock. If there was no elastic media between the body structure and the deflecting suspension, any sudden upward movement would be transmitted directly to the body structure and passengers thus producing a very uncomfortable ride.

In actual fact movement of the fluid from the displacer chamber into the spring chamber takes place in three stages:

1 If the road bumps are very small and the vehicle is moving slowly, sufficient fluid flows through the permanently open transfer hole to equalize the pressure on both sides of this restriction.
2 If the road bumps are more severe the increased pressure build-up in the displacer chamber will be sufficient to lift the flaps on the rubber bump valve off a second pair of bleed holes. Additional fluid can now flow into the spring chamber in a shorter time span.
3 If the roughness of the road surface worsens or the speed at which the vehicle travels increases even more, then there will be a continuous rise in pressure of the fluid trapped in the displacer chamber. As a result of the extreme pressure build-up, the rubber bump valve itself will be progressively lifted from its seat to permit more fluid to enter the spring chamber. Thus in total more fluid is transferred from the displacer chamber to the spring chamber in a given time, but the built-in opposing resistance to the flow of fluid produces a measure of damping which

slows down the violent uplifts caused by the impact of the tyre with obstacles in the road.

Spring extending due to rebound response (Fig. 10.69(b)) After the wheel has passed over a hump in the road the bounce action of the nitrogen gas pushes some of the fluid from the spring chamber back to the displacer chamber causing the displacer piston to extend from the displacer chamber.

The return of fluid from the spring chamber to the displacer chamber takes place in two stages:

1 If the bumps in the road are small or the vehicle is moving very slowly, then only a small amount of fluid needs to be transferred back to the displacer chamber in a given time. The movement of this fluid out of the spring chamber can be coped with adequately by the permanently open transfer hole. This means the damping action takes place as fluid is bypassed through the permanent bleed hole for low speed conditions.
2 If the bumps in the road are larger and the speed of the vehicle is higher, then the highly pressurized fluid in the spring chamber will lift the rubber rebound valve progressively from its seat, thus permitting a greater rate of flow of fluid back into the displacer chamber.

Because the progressive opening of the rubber valve is pressure sensitive, the flow of fluid is restricted and it is this tendency to slow down the fluid movement that produces the retarding effect on the rebound expanding gas.

Comparison of bump and rebound fluid damping control The extension (rebound) of the displacer piston is slightly slower than on contraction (bump) because there is not an intermediate flap valve second stage opening as there is on bump. Thus for small deflections of the displacer piston the permanent bleed transfer hole controls the movement of fluid in both bump and rebound directions. For more rapid displacement of fluid on rebound there is only the rebound compressive rubber block valves which regulate the flow of fluid in the extending direction, this being equivalent to both the flap valve and compressive rubber block valve opening on the contracting (bump) stroke.

Bump and pitch mode (Fig. 10.69(c)) When the front or rear wheel passes over a bump, the contraction of the displacer piston inside the displacer chamber at that wheel causes fluid displacement through the interconnecting pipe to the other wheel spring unit on the same side of the vehicle.

This movement of fluid into the other spring unit's displacer chamber extends its displacer piston within the chamber and thereby lifts the suspension and body up to the same level as that at the car's opposite end. Fluid movement from one suspension spring unit to the other therefore prevents pitch and enables the car to ride at a level attitude.

At moderate speeds the fluid is simply displaced from front to rear spring unit and vice versa, the fluid pressure remaining constant so that the coupled nitrogen gas springs are not further deflected.

Roll or bounce mode (Fig. 10.69(c)) If the body rolls due to cornering or the car bounces as a whole, then both front and rear suspensions are deflected together. The simultaneous fluid displacements increase the fluid pressure and dynamically compress and contract both of the nitrogen springs. Thus with the inward movement of the pistons the projected effective piston areas increase so that a larger fluid area has to be lifted. Consequently both the front and rear spring stiffness on the side of the body furthest away from the turn considerably increase the suspension's resistance to roll.

Similarly, if the body bounces at both ends together, then the spring stiffness rates increase as the displacer pistons approach their inner dead centres so that a much greater resistance against the downward movement of the body occurs if the bounce becomes violent.

10.10 Hydropneumatic automatic height correction suspension (Citroen) (Figs 10.70, 10.71 and 10.72)

The front suspension may be either a MacPherson strut (Fig. 10.70) or a transverse double wishbone arm arrangement (Fig. 10.71(a)), whereas the rear suspension is of the trailing arm type. Front and rear anti-roll bars are incorporated to increase the body roll stiffness and to actuate both front and rear height correction valves.

Spring unit (Fig. 10.71(a)) The suspension spring units (Fig. 10.71(a)) comprise two main parts;

1 a steel spherical canister containing a rubber diaphragm which separates the nitrogen spring media from the displacement fluid;
2 a steel cylinder and piston which relays the suspension's vertical deflection movement to the rubber diaphragm by displacing the fluid.

When the wheel meets a hump in the road, the piston is pushed inwards so that it displaces fluid from the cylinder into the sphere. Consequently the flexible rubber diaphragm squeezes the nitrogen gas into a small space (Fig. 10.71(b)). If the wheel hits a pot hole, the pressurized gas expands and forces fluid from the sphere into the cylinder, thereby making the piston move outward. By this method of changing the volume of fluid entering the sphere, the gas either is compressed or expanded relative to the initial charge pressure so that the resilience of the gas prevents the force of the road shocks from transferring to the body structure.

Pump accumulator and pressure regulator (Fig. 10.70) The initial fluid pressure source comes from a seven piston swashplate engine-driven hydraulic pump which is able to provide a continuous flow of fluid at a predetermined pressure. The pump feeds the spherically shaped accumulator which uses nitrogen as the spring media and a rubber diaphragm to accommodate the volume of stored fluid. The accumulator stores the highly pressurized fluid and can immediately deliver fluid to the system in the event of a sudden demand. It also permits the pump to idle and therefore eliminates repeated cutting in and out.

When the pump is idling the pressure generated is only enough to return the fluid to the reservoir through the pressure regulator. The pressure regulator and accumulator unit control the minimum pressure necessary for the operation of the system and the maximum pressure needed to charge the accumulator and to limit the maximum pressure delivered by the pump (the pump cut-in pressure of 140–150 bar and the cut-out pressure of 165–175 bar).

Height correction valve (Fig. 10.72(a, b, c and d)) Automatic height correction is achieved by varying the volume of incompressible fluid between the sphered diaphragm and the piston. Increased vehicle weight lowers the body, thus causing the suspension arms to deflect and at the same time rotate the anti-roll bar. The angular rotation of the anti-roll bar is a measure of the suspension's vertical deflection relative to the vehicle's normal static height. This movement is relayed to the height correction valve via a torsional control rod clamped to the anti-roll bar at one end and to a control rod lever which is attached to the height correction valve at the other end.

To avoid continuous height correction every time a pair of wheels roll over a hump or dip in the road, a delayed response is introduced to the
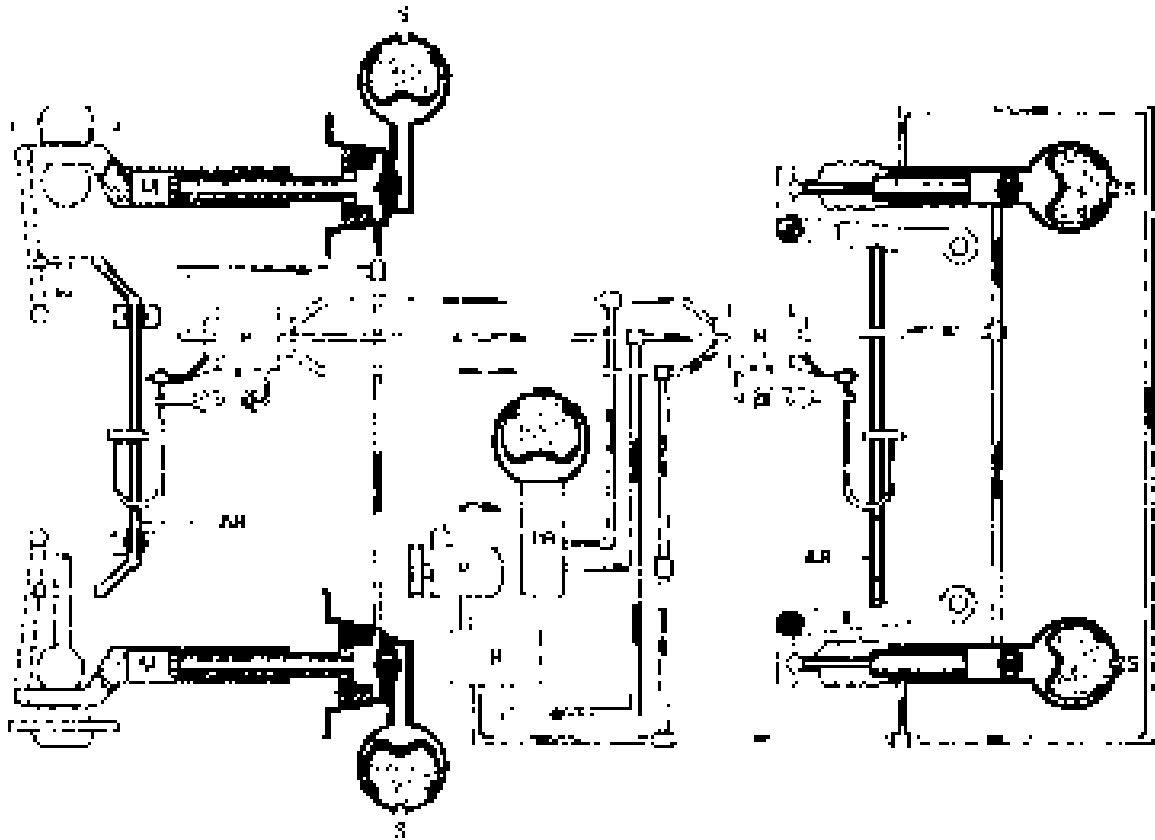
Fig. 10.70 eneral layout of the hydropneumatic automatic height correction suspension

height correction valves so that the spring unit cylinder is not being charged on bump or discharged on rebound. Height correction will therefore be achieved only after a small time pause during which time the suspension will have had time to adjust to a change in the loads imposed on the spring units. Once the spring unit cylinder has been fully recharged, or discharged to bring the suspension height back to the standard setting, the height correction valve is made to respond immediately by either moving from inlet charging to neutral cut-off or from exhaust discharge to neutral cut-off position.

Charging the spring unit (spool valve movement from neutral cut off to inlet open) (Fig. 10.72(a)) An increase in car load causes the lower transverse arm to pivot and the anti-roll bar to rotate. At the same time the control rod twists and tries to tilt the control rod lever, thereby transmitting an axial load to the height correction spool valve. The effect of shifting the spool valve to the left hand side is to move it from the cut-off position to the inlet open position. An increased amount of fluid is now forced between the piston and diaphragm causing the vehicle to rise until the anti-roll bar, which is rotating in the opposite direction, pulls the spool valve back to the neutral cut-off position. The return to the cut-off position is rapid because the spool valve does not offer any resistance in this direction, and the vehicle height will have been brought back to its normal position. To slow down the movement from cut-off to inlet charge positions, the disc valve in the right hand diaphragm chamber is closed. Therefore, the only way the fluid can be transferred from the right to

409
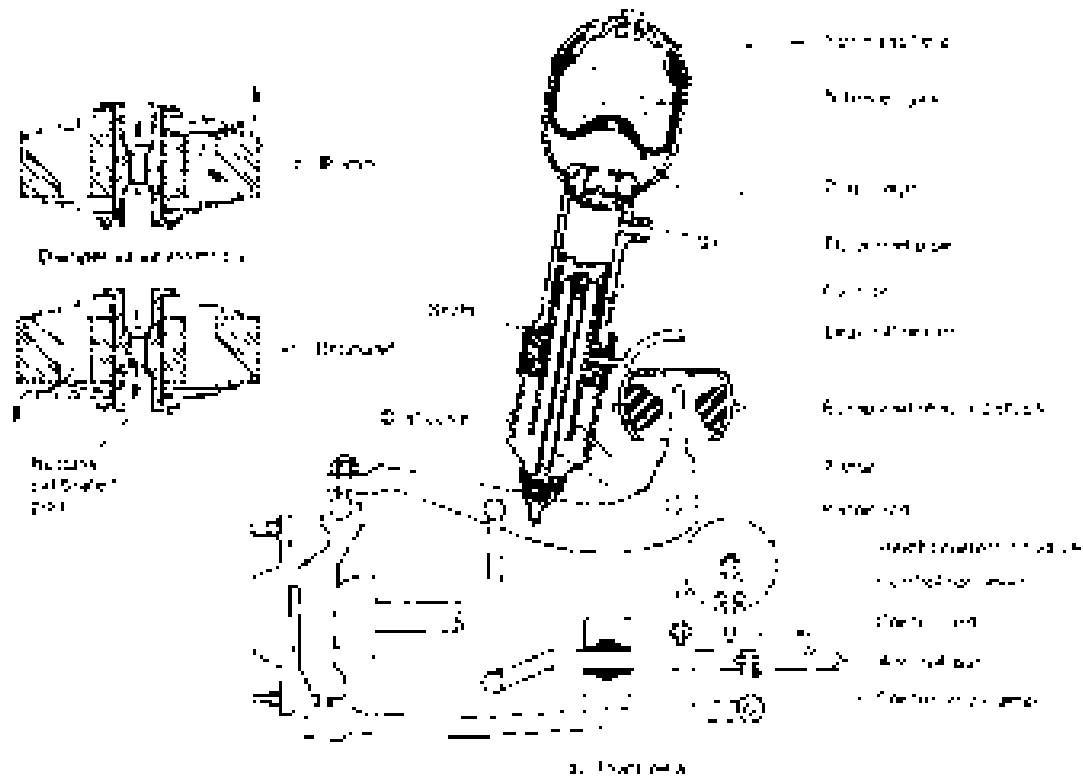
Fig. 10.71 (a–c) Detailed arrangement of hydropneumatic spring unit using a transverse double arm linkage suspension

the left hand chamber is through the restricted passage with the result that the spool valve shift movement is very sluggish.

ischarge the spring unit (spool valve movement from neutral cut off to exhaust open) (Fig. 10.72(b)) Decreasing the car load has the reverse effect to increasing the load. This time the spool valve moves from the neutral cut-off position to the exhaust open position. The excess fluid between the piston and diaphragm is now expelled to the reservoir tank and the suspension spring unit contracts until the body to ground height has been corrected, at which point the spool valve again will be in the neutral cut-off position. Similarly the discharge process is also slowed down so that the valve does not respond to small changes in dynamic loads caused by suspension vibration as the wheels travel over the road surface irregularities.

Spool valve movement from inlet charge to neutral cut off (Fig. 10.72(c)) Once the spring unit cylinder has been fully recharged with fluid, the anti-roll bar will have rotated sufficiently to make the spool

valve alter its direction of slide towards the neutral cut-off position. This return movement of the spool valve to the cut-off position is rapidly speeded up because the left hand disc valve is in the open position so that when the spool valve first starts to change its direction of slide, fluid in the unrestricted passage will force the right hand valve off its seat. As a result, fluid movement from the left hand to the right hand diaphragm chamber takes place through both the restricted and unrestricted passages, speeding up the fluid transfer and accordingly the spool valve movement to the neutral cut-off position. As soon as the spool valve reaches its cut-off position the disc valve in the left hand diaphragm chamber re-seats. This action stops the spool valve overshooting its cut-off position and therefore avoids the valve going through a second recharge and discharge cycle of correction.

Spool valve movement from exhaust discharge to neutral cut off (Fig. 10.72(d)) A rapid closing of the exhaust valve takes place once the fluid in an over-charged cylinder has been permitted to escape back to the reservoir thus restoring the suspension

410
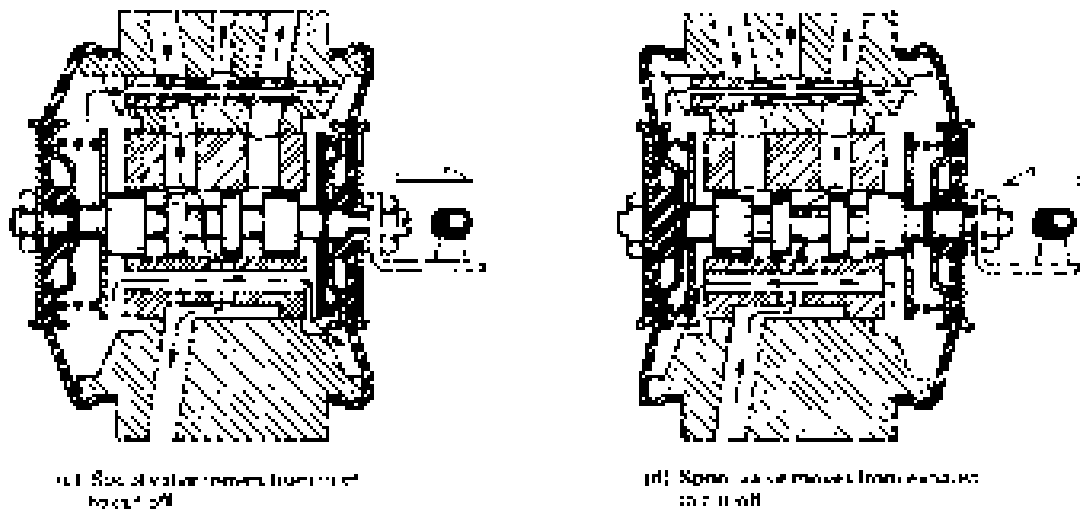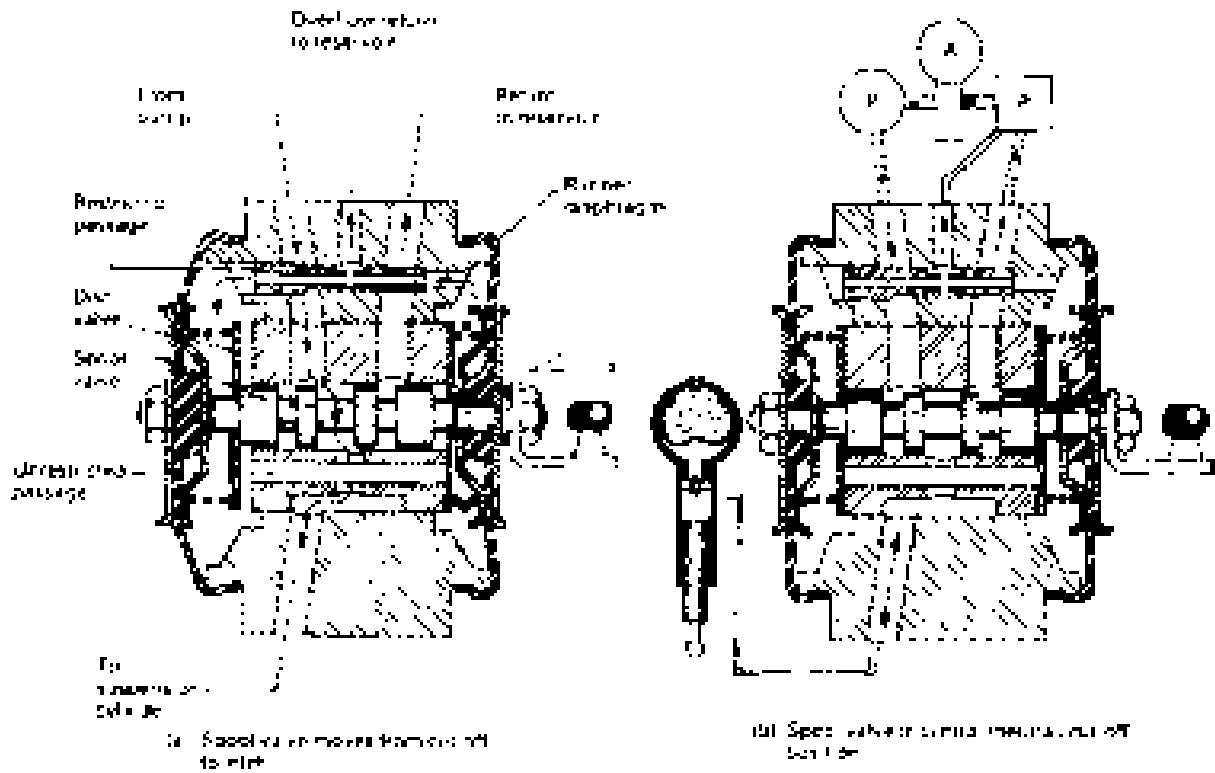
Fig. 10.72 (a–d) Height correction valve action

411

to its standard height. The ability for the spool valve to respond quickly and close off the exhaust valve is due to the right hand disc valve being open. Thus fluid in the unrestricted passage is permitted to push open the right hand disc valve, this allows fluid to readily move through both the restricted and unrestricted passages from the right to left hand diaphragm chamber. Immediately the torsional wind-up of the control rod due to the anti-roll bar rotation causes the spool valve to shift to the neutral cut-off position.

anual height correction A manual control lever is provided inside the car, the lever being connected by actuating rods to the front and rear height correction units. Its purpose is to override the normal operation of the spool valve and to allow the driver to select five different positions:

Normal	this is the standard operating position
High or low	two extreme positions
Two positions	intermediate between normal and high

10.10.1 Hydropneumatic self-levelling spring unit
(Figs 10.73(a and b) and 10.74(a, b and c))
This constant height spring unit consists of two sections;

1 a pneumatic spring and hydraulic damper system,
2 a hydraulic constant level pump system.

An approximately constant frequency of vibration for the sprung mass, irrespective of load, is obtained by having two gas springs, a main gas spring, in which the gas is contained behind a diaphragm, and a correction gas reservoir spring (Fig. 10.73(a, b and c)). The main spring is controlled by displacing fluid from the upper piston chamber to the spring diaphragm chamber and the correction gas spring is operated by the lower piston chamber discharging fluid into the reservoir gas spring chamber.

The whole spring unit resembles a telescopic damper. The cylindrical housing is attached to the sprung body structure whereas the piston and integral rod are anchored to either the unsprung suspension arm or axle.

The housing unit comprises four coaxial cylinders;

1 the central pump plunger cylinder with the lower conical suction valve and an upper one way pump outlet disc valve mounted on the piston,

2 the piston cylinder which controls the gas springs and damper valves,
3 the inner gas spring and reservoir chamber cylinder,
4 the outer gas spring chamber cylinder which is separated from hydraulic fluid by a flexible diaphragm.

The conical suction valve which is mounted in the base of the plunger's cylinder is controlled by a rod located in the hollow plunger. A radial bleed port or slot position about one third of the way down the plunger controls the height of the spring unit when in service.

The damper's bump and rebound disc valves are mounted in the top of the piston cylinder and an emergency relief valve is positioned inside the hollow pump plunger at the top.

The inner gas spring is compressed by hydraulic fluid pressure generated by the retraction of the space beneath the piston.

The effective spring stiffness (rate) is the sum of the stiffnesses of the two gas springs which are interconnected by communication passages. Therefore the stiffness increase of load against deflection follows a steeper curve than for one spring alone.

as spring and damper valve action (Fig. 10.73 (a and b)) There are two inter-related cycles; one is effected by the pressure generated above the piston and the other relates to the pressure developed below the piston.

When, during bump travel (Fig. 10.73(a)), the piston and its rod move upwards, hydraulic fluid passes through the damper bump valve to the outer annular main gas spring chamber and compresses the gas spring. Simultaneously as the load beneath the piston reduces, the inner gas spring and reservoir expand and fluid passes through the transfer port in the wall to fill up the enlarging lower piston chamber cylinder. Thus the deflection of the diaphragm against the gas produces the elastic resilience and the fluid passing through the bump valve slows down the transfer of fluid to the gas spring so that the bump vibration frequency is reduced.

On rebound (Fig. 10.73(b)) fluid is displaced from the outer spring chamber through the damper rebound valve into the upper piston cylinder and at the same time fluid beneath the piston is pushed out of the lower piston chamber into the inner gas spring chamber where it now compresses the inner gas spring.

Likewise fluid which is being displaced from the main gas spring to the upper piston chamber

412
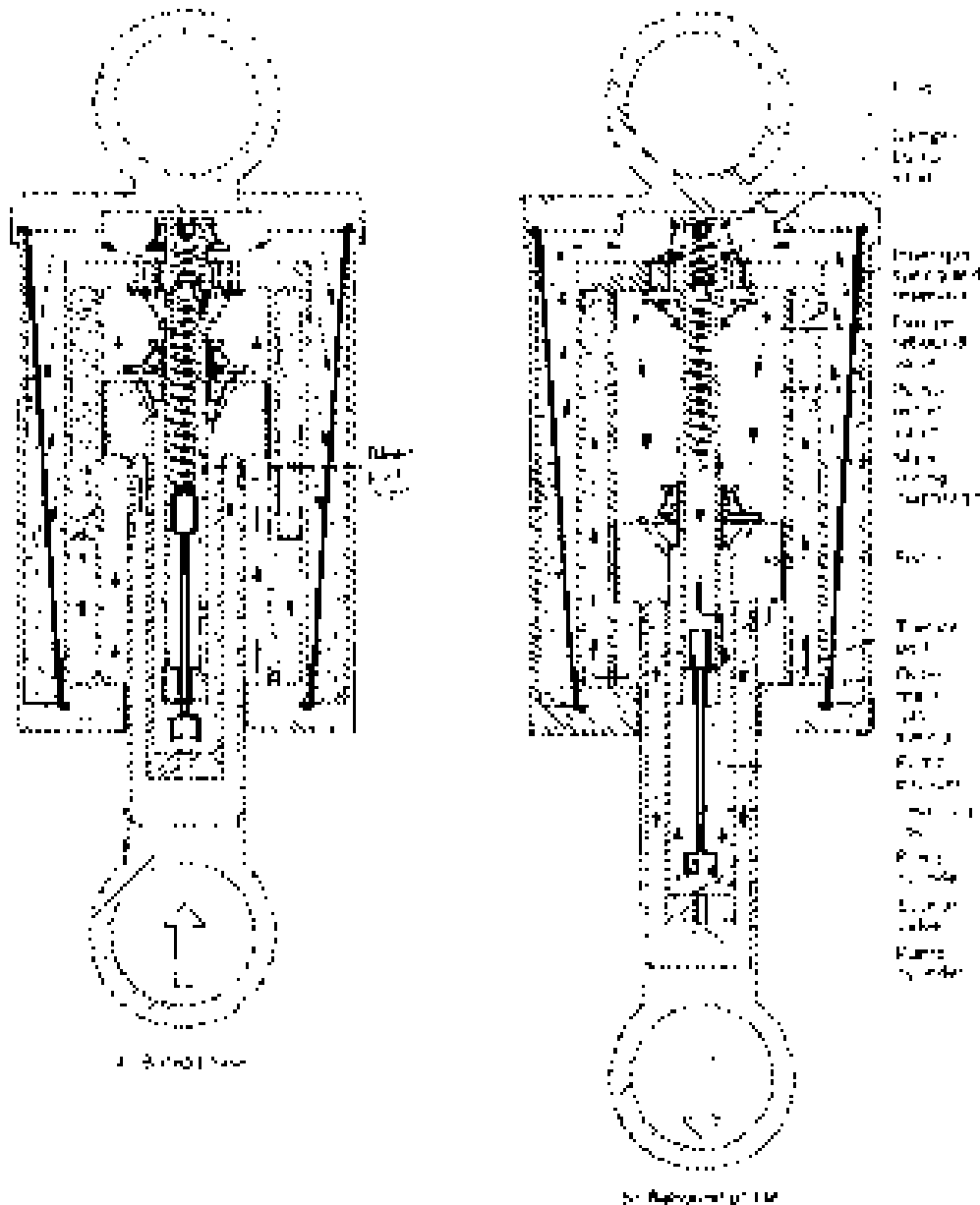
Fig. 10.73 (a and b) Exaggerated diagrams illustrating the self-levelling action of a hydropneumatic suspension unit

experiences an increased resistance due to the rebound valve passage restriction so that the fluid transfer is achieved over a longer period of time.

Pump self-levelling action (Figs 10.74(a, b and c) and 10.73(a and b)) The movement of the piston within its cylinder also causes the pump plunger to be actuated. During bump travel (Figs 10.73(a) and 10.74(a)) the plunger chamber space is reduced,

causing fluid to be compressed and pushed out from below to above the piston via the pump outlet valve. On rebound (Fig. 10.74(c)), the volume beneath the piston is replenished. However, this action only takes place when the piston and rod have moved up in the cylinder beyond the designed operating height.

The conical suction valve, which is mounted in the base of the plunger's cylinder and is controlled

413

Fig. 10.74 (a–c) Self-levelling hydropneumatic suspension

by a rod located in the hollow plunger, and also a radial bleed port or slot, positioned about one third of the way down the plunger, control the height of spring unit when in service.

The damper's bump and rebound disc valves are mounted in the top of the piston cylinder and an emergency relief valve is positioned inside the hollow pump plunger at the top.

The inner gas spring is compressed by hydraulic fluid pressure generated by the retraction of the space beneath the piston.

The pumping action is provided by the head of the plunger's small cross-sectional area pushing down onto the fluid in the pump chamber during the bump travel (Fig. 10.74(a)). This compels the fluid to transfer through the pump outlet valve into the large chamber above the piston. The pressure of the fluid above the piston and that acting against the outer gas spring diaphragm is the pressure necessary to support the vehicle's unsprung mass which bears down on the spring unit. During rebound travel (Fig. 10.74(c)), the fluid volume in the pump chamber increases while the volume beneath the piston decreases. Therefore some of the fluid in the chamber underneath the piston will be forced into the inner gas spring chamber

414

against the trapped gas, whilst the remainder of the excess fluid will be transferred from the lower piston chamber through a passage that leads into an annular chamber that surrounds the pump chamber. The pressurized fluid surrounding the pump chamber will then force open the conical suction valve permitting fluid to enter and fill up the pump chamber as it is expanded during rebound (Fig. 10.74(c)). This sequence of events continues until the piston has moved far enough down the fixed pump plunger to expose the bleed port (or slot) in the side above the top of the piston (Figs 10.74(c) and 10.73(b)).

At this point the hollow plunger provides a connecting passage for the fluid so that it can flow freely between the upper piston chamber and the lower plunger chamber. Therefore, as the piston rod contracts on bump, the high pressure fluid in the plunger chamber will be discharged into the upper piston chamber by not only the pump outlet valve but also by the plunger bleed port (slot) (Fig. 10.74(a)). However, on the expansion stroke some of the pressurized fluid in the upper piston chamber can now return to the plunger chamber and thereby prevents the conical suction valve opening against the pressure generated in the lower piston chamber as its volume decreases. The plunger pumping action still continues while the spring unit height contracts, but on extension of the spring unit (Fig. 10.74(c)) the fluid is replenished not from the lower piston chamber as before but from the upper piston chamber so that the height of the spring unit cannot increase the design spring unit length.

When the spring unit is extended past the design height the underside of the piston increases the pressure on the fluid in the reservoir chamber and at the same time permits fluid to bleed past the conical suction valve into the plunger chamber. If the spring unit becomes fully extended, the suction valve is lifted off its seat, enabling the inner spring chamber to be filled with fluid supplied from the lower piston chamber and the plunger chamber.

10.11 Commercial vehicle a le beam location

An axle beam suspension must provide two degrees of freedom relative to the chassis which are as follows:

1 Vertical deflection of axle due to static load or dynamic bump and rebound so that both wheels can rise and fall together.
2 Transverse axle twist to permit one wheel to rise while the other one falls at the same time as the vehicle travels over uneven ground.

In addition, the suspension must be able to restrain all other axle movements relative to the chassis and the construction should be such that it is capable of supporting the forces and moments that are imposed between the axle and chassis.

Both vertical axle deflection and transverse axle tilt involve some sort of rotational movement of the restraining and supporting suspension members, be they the springs themselves or separate arm members they must be able to swing about some pivot point.

The two basic methods of providing articulation of suspension members is the pivot pin joint and the ball and socket joint. These joints may either be rigid metal, semi-rigid plastic or flexible rubber, their selection and adoption being determined by the vehicle's operating requirements.

To harness the axle so that it is able to transfer accelerating effort from the wheels to the chassis and vice versa, the suspension must have built-in members which can absorb the following forces and moments;

1 vertical forces caused by vehicle laden weight,
2 longitudinal forces caused by tractive and braking effort,
3 transverse forces caused by centrifugal force, side slopes and lateral winds,
4 rotational torque reactions caused by driving and braking efforts.

10.11.1 ulti-leaf spring eye support
(Fig. 10.75(a, b and c))
Axle location by multi-leaf springs relies on the spring eyes having sufficient strength and support to cope with the vehicle's laden weight driving and braking thrust and lateral forces. Springs designed

Fig. 10.75 Spring eye protection

415

for cars and light vans generally need only a single main leaf (Fig. 10.75(a)) wrapped around the bush and shackle pin alone, but for heavy duty conditions it is desirable to have the second leaf wrapped around the main leaf to give it additional support.

If a second leaf were to be wrapped tightly around the main leaf eye, then there could not be any interleaf sliding which is essential for multi-leaf spring flexing to take place. As a compromise for medium duty applications, a partial or half-wrapped second leaf may be used (Fig. 10.75(b)) to support the main leaf of the spring. This arrangement permits a small amount of relative lengthwise movement to occur when the spring deflects between bump and rebound. For heavy duty working conditions, the second leaf may be wrapped loosely in an elongated form around the main lead eye (Fig. 10.75(c)). This allows a degree of relative movement to occur, but at the same time it provides backup for the main leaf eye. If the main leaf should fracture at some point, the second leaf is able to substitute and provide adequate support; it therefore prevents the axle becoming out of line and possibly causing the vehicle to steer out of control.

10.11.2 Transverse and longitudinal spring axle and chassis attachments (Figs 10.76–10.83)

For small amounts of transverse axle twist, rubber bushes supporting the spring eye-pins and shackle plates are adequate to absorb linkage misalignment, and in extreme situations the spring leaves themselves can be made to distort and accommodate axle transverse swivel relative to the chassis frame. In certain situations where the vehicle is expected to operate over rough ground additional measures may have to be taken to cope with very large degrees of axle vertical deflection and transverse axle tilt.

The semi-elliptic spring may be attached to the chassis and to the axle casing in a number of ways to accommodate both longitudinal spring leaf camber (bow) change due to the vehicle's laden weight and transverse axle tilt caused by one or other wheel rising or falling as they follow the contour of the ground.

Spring leaf end joint attachments may be of the following kinds;

a) cross-pin anchorage (Fig. 10.76),
b) pin and fork swivel anchorage (Fig. 10.77),
c) bolt and fork swivel anchorage (Fig. 10.78),
d) pin and ball swivel anchorage (Fig. 10.79),
e) ball and cap swivel anchorage (Fig. 10.80).

Alternatively, the spring leaf attachment to the axle casing in the mid-span region may not be a direct clamping arrangement, but instead may be through some sort of pivoting device to enable a relatively large amount of transverse axle tilt to be

Fig. 10.77 Main spring to chassis pin and fork swivel anchorage

Fig. 10.76 (a and b) Main spring to chassis hinged cross-pin anchorage
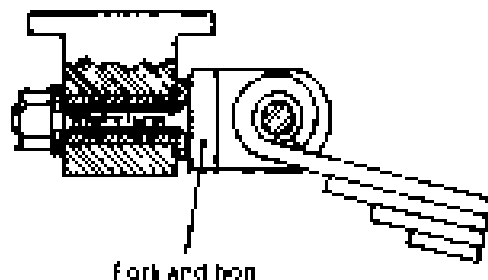
Fig. 10.78 Main spring to chassis bolt and fork swivel anchorage

416
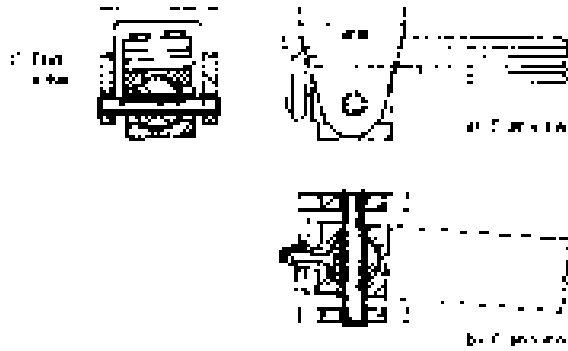
Fig. 10.79 (a–c) Main spring to chassis pin and spherical swivel anchorage
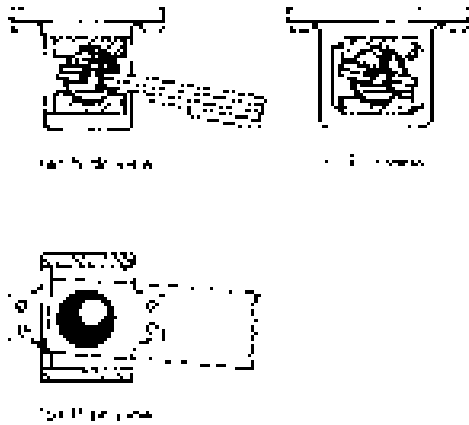
Fig. 10.80 (a–c) Main spring to chassis spherical swivel anchorage
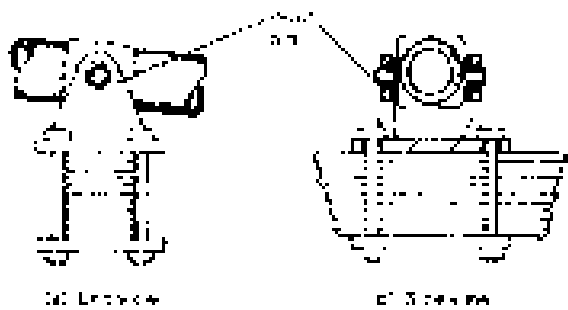
Fig. 10.81(a and b) Axle to spring pivot pin seat mounting

accommodated. Thus transverse axle casing to spring relative movement can be achieved by either a pivot pin (Fig. 10.81) or a spherical axle saddle joint (Fig. 10.82) arrangement. Likewise for reactive balance beam shackle plate attachments the
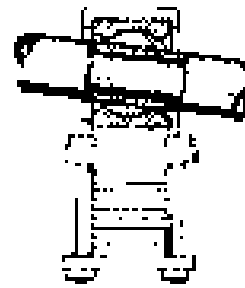
Fig. 10.82 Axle to spring spherical seat mounting
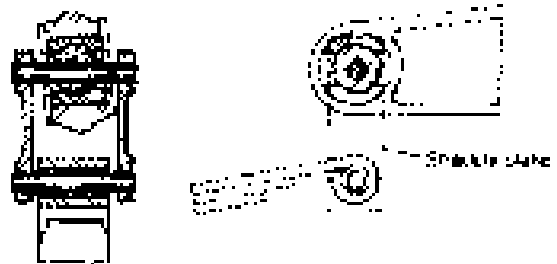
Fig. 10.83 Tandem axle balance beam to shackle plate spherical joint

joints may also be of the spherical ball and cap type joint (Fig. 10.83).

10.12 Variable rate leaf suspension springs

The purpose of the suspension is to protect the body from the shocks caused by the vehicle moving over an uneven road surface. If the axle were bolted directly to the chassis instead of through the media of the springs, the vehicle chassis and body would try to follow a similar road roughness contour and would therefore lift and fall accordingly. With increased speed the wheel passing over a bump would bounce up and leave the road so that the grip between the tyre and ground would be lost. Effectively no tractive effort, braking retardation or steering control could take place under these conditions.

A suspension system is necessary to separate the axle and wheels from the chassis so that when the wheels contact bumps in the road the vertical deflection is absorbed by the elasticity of the spring material, the strain energy absorbed by the springs on impact being given out on rebound but under damped and controlled conditions. The deflection of the springs enables the tyres to remain in contact with the contour of the road under most operating conditions. Consequently the spring insulates the

417

body from shocks, protects the goods being transported and prevents excessively high stresses being imposed on the chassis which would lead to fatigue failure. It also ensures that the driver is cushioned from road vibrations transmitted through the wheels and axle, thereby improving the quality of the ride. The use of springs permits the wheels to follow the road contour and the chassis and body to maintain a steady mean height as the vehicle is driven along the road. This is achieved by the springs continuously extending and contracting between the axle and chassis, thereby dissipating the energy imparted to the wheels and suspension assembly.

A vehicle suspension is designed to permit the springs to deflect from an unladen to laden condition and also to allow further deflection caused by a wheel rapidly rolling over some obstacle or pot hole in the road so that the impact of the unsprung axle and wheel responds to bump and rebound movement. How easily the suspension deflects when loaded statically or dynamically will depend upon the stiffness of the springs (spring rate) which is defined as the load per unit deflection.

$$\text{i e} \quad \text{Spring stiffness or rate } S = \frac{\text{Applied load}}{\text{Deflection}}$$

$$= \frac{}{x} \,(\text{N/m})$$

A low spring stiffness (low spring rate) implies that the spring will gently bounce up and down in its free state which has a low natural frequency of vibration and therefore provides a soft ride. Conversely a high spring stiffness (high spring rate) refers to a spring which has a high natural frequency of vibration which produces a hard uncomfortable ride if it supports only a relatively light load. Front and rear suspensions have natural frequencies of vibration roughly between 60 and 90 cycles per minute. The front suspension usually has a slightly lower frequency than the rear. Typical suspension natural frequencies would be 75/85 cycles per minute for the front and rear respectively. Spring frequencies below 60 cycles per minute promote car sickness whereas frequencies above 90 cycles per minute tend to produce harsh bumpy rides. Increasing the vehicle load or static deflection for a given set of front and rear spring stiffness reduces the ride frequency and softens the ride. Reducing the laden vehicle weight raises the frequency of vibration and the ride hardness.

Vehicle laden weight, static suspension deflection, spring stiffness and ride comfort are all inter-related and produce conflicting characteristics.

For a car there is not a great deal of difference between its unladen and fully laden weight; the main difference being the driver, three passengers, luggage and full fuel tank as opposed to maybe a half full fuel tank and the driver only. Thus if the car weighs 1000 kg and the three passengers, luggage and full fuel tank weigh a further 300 kg, the ratio of laden and unladen weight will be 1300/1000 = 1.3:1. Under these varying conditions, the static suspension deflection can be easily accommodated by soft low spring rates which can limit the static suspension deflection to a maximum of about 50 mm with very little variation in the natural frequency of vibration of the suspension system. For a heavy goods vehicle, if the unladen weight on one of the rear axles is 2000 kg and its fully laden capacity is 10 000 kg, then the ratio of laden to unladen weight would be 10 000/2000 = 5:1. It therefore follows that if the spring stiffness for the axle suspension is designed to give the best ride with the unladen axle, a soft low spring rate would be required. Unfortunately, as the axle becomes fully laden, the suspension would deflect maybe five times the unladen static deflection of, say, 50 mm which would amount to 250 mm. This large change from unladen to fully laden chassis height would cause considerable practical complications and therefore could not be acceptable.

If the suspension spring stiffnesses were to be designed to give the best ride when fully laden, the change in suspension deflection could be reduced to something between 50 and 75 mm when fully laden. The major disadvantage of utilizing high spring rates which give near optimum ride conditions when fully laden would be that when the axle is unladen, the stiffness of the springs would be far too high so that a very hard uncomfortable ride would result, followed by mechanical damage to the various chassis and body structures.

It is obvious that a single spring rate is unsuitable and that a dual or progressive spring rate is essential to cope with large variations in vehicle payload and to restrict the suspension's vertical lift or fall to a manageable amount.

10.12.1 Dual rate helper springs (Fig. 10.84(a))

This arrangement is basically a main semi-elliptic leaf spring with a similar but smaller auxiliary spring located above the main spring. This spring is anchored to the chassis at the front via a shackle pin to the spring hanger so that the driving thrust can be transmitted from the axle and wheel to the chassis. The rear end of the spring only supports
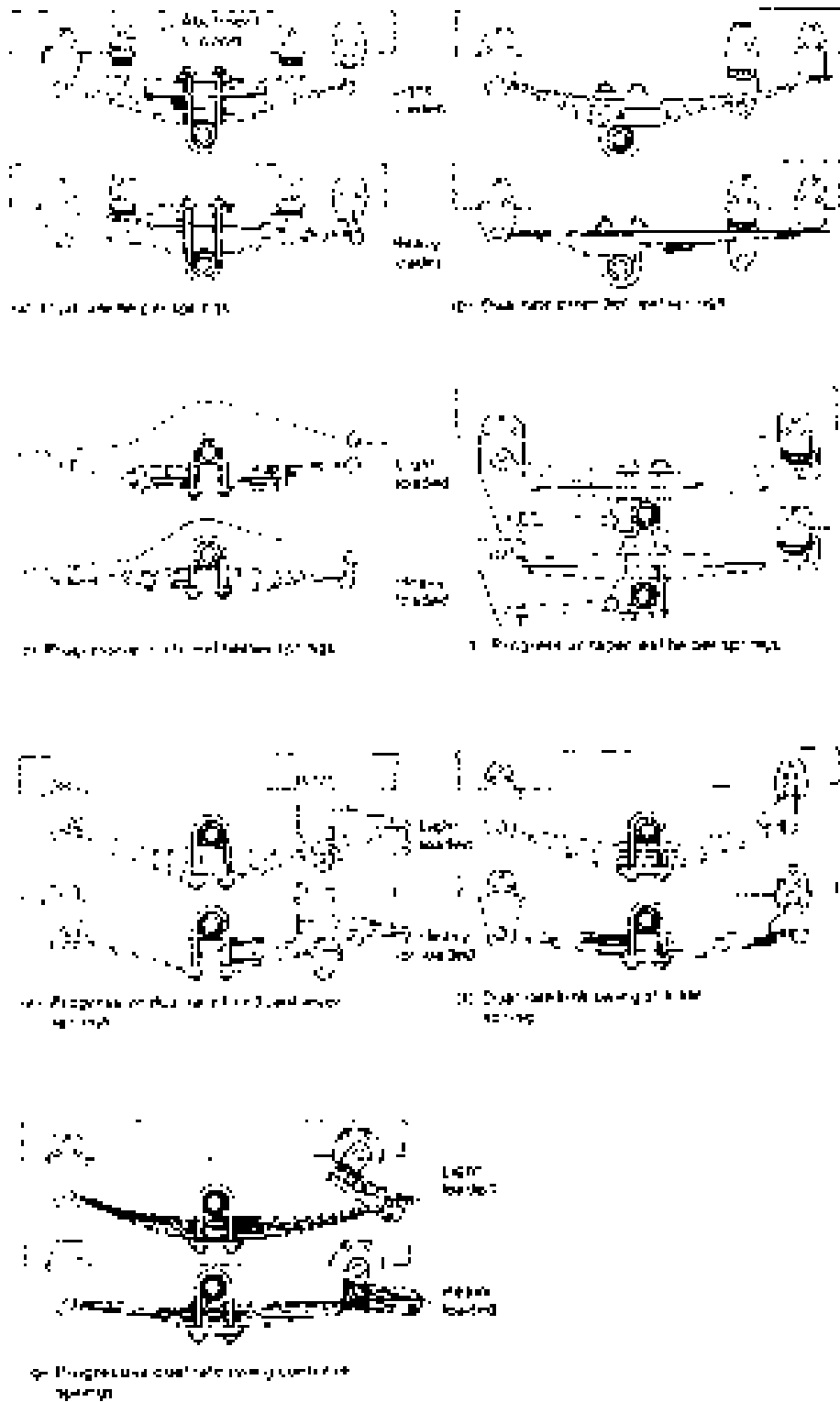
Fig. 10.84 (a–g) Variable rate leaf spring suspension

the downward load and does not constrain the fore and after movement of the spring.

In the unladen state only the main spring supports the vehicle weight and any payload carried (Fig. 10.84(a)) is subjected to a relatively soft ride. Above approximately one third load, the ends of the auxiliary helper spring contact the abutments mounted on the chassis. The vertical downward deflection is now opposed by both sets of springs which considerably increase the total spring rate and also restrict the axle to chassis movement. The method of providing two spring rates, one for lightly laden and a second for near fully laden condition, is adopted by many heavy goods vehicles.

10.12.2 Dual rate extended leaf springs
(Fig. 10.84(b))
With this semi-elliptic leaf spring layout the axle is clamped slightly offset to the mid-position of the spring. The front end of the spring is shackled to the fixed hanger, whereas the rear end when unloaded bears against the outer slipper block. The full span of the spring is effective when operating the vehicle partially loaded. A slight progressive stiffening of the spring occurs with small increases in load, due to the main spring blade rolling on the curved slipper pad from the outermost position towards its innermost position because of the effective spring span shortening. Hence the first deflection stage of the spring provides a very small increase in spring stiffness which is desirable to maintain a soft ride.

Once the vehicle is approximately one third laden, the deflection of the spring brings the main blade into contact with the inner slipper block. This considerably shortens the spring length and the corresponding stiffening of the spring prevents excessive vertical deflection. Further loading of the axle will make the main blade roll round the second slipper block, thereby providing the second stage with a small amount of progressive stiffening. Suspension springing of this type has been successful on heavy on/off road vehicles.

10.12.3 Progressive multi-leaf helper springs
(Fig. 10.84(c))
The spring span is suspended between the fixed hanger and the swinging shackle. The spring consists of a stack of leaves clamped together near the mid-position, with about two thirds of the leaves bowed (cambered) upward so that their tips contact and support the immediate leaf above it. The remainder of the leaves bow downward and so do not assist in supporting the body weight when the

car or van is only partially laden. As the vehicle becomes loaded, the upper spring leaves will deflect and curve down on either side of the axle until their shape matches the first downward set lower leaf. This provides additional upward resistance to the normally upward bowed (curved) leaves so that as more leaves take up the downward bowed shape more of the leaves become active and contribute to the total spring stiffness. This progressive springing has been widely used on cars and vans.

10.12.4 Progressive taper leaf helper springs
(Fig. 10.84(d))
Under light loads a small amount of progressive spring stiffening occurs as the rear end of the main taper leaf rolls from the rearmost to the frontmost position on the curved face of the slipper block, thereby reducing the effective spring length. The progressive action of the lower helper leaf is caused by the normally upward curved main taper leaf flexing and flattening out as heavier loads are imposed on the axle. The consequences are that the main spring lower face contact with the upper face of the helper leaf gradually spreads outwards and therefore provides additional and progressive support to the main taper leaf.

The torque rod is provided to transmit the driving force to the chassis and also forms the cranked arms of an anti-roll bar in some designs.

This progressive spring stiffening arrangement is particularly suitable for tractor unit rear suspension where the rates of loaded to unloaded weight is large.

10.12.5 Progressive dual rate fixed cantilever spring (Fig. 10.84(e))
This interesting layout has the front end of the main leaf spring attached by a shackle pin to the fixed hanger. The main blade rear tip contacts the out end of a quarter-elliptic spring, which is clamped and mounted to the rear spring hanger. When the axle is unloaded the effective spring length consists of both the half- and quarter-elliptic main leaf spans so that the combined spring lengths provides a relative low first phase spring rate.

As the axle is steadily loaded both the half- and quarter-elliptic main leaves deflect and flatten out so that their interface contact area progressively moves forwards until full length contact is obtained. When all the leaves are aligned the effective spring span is much shorter, thereby considerably increasing the operating spring rate. This spring suspension concept has been adopted for the rear spring on some tractor units.

10.12.6 Dual rate kink swing shackle spring
(Fig. 10.84(f))
Support for the semi-elliptic spring is initially achieved in the conventional manner; the front end of the spring is pinned directly to the front spring hanger and indirectly via the swinging shackle plates to the rear spring hanger. The spring shackle plates have a right angled abutment kink formed on the spring side of the plates.

In the unladen state the cambered (bowed) spring leaves flex as the wheel rolls over humps and dips, causing the span of the spring to continuously extend and contract. Thus the swinging shackle plates will accommodate this movement. As the axle becomes laden, the cambered spring leaves straighten out until eventually the kink abutment on the shackle plates contact the upper face of the main blade slightly in from the spring eye. Any further load increase will kink the main leaf, thereby shortening the effective spring span and resulting in the stiffening of the spring to restrict excessive vertical deflection. A kink swing shackle which provides two stages of spring stiffness is suitable for vans and light commercial vehicles.

10.12.7 Progressive dual rate swing contilever springs (Fig. 10.84(g))
This dual rate spring has a quarter-elliptic spring pack clamped to the spring shackle plates. In the unloaded condition the half-elliptic main leaf and the auxiliary main leaf tips contact each other. With a rise in axle load, the main half-elliptic leaf loses its positive camber and flattens out. At the same time the spring shackle plates swing outward. This results in both main spring leaves tending to roll together thereby progressively shortening the effective spring leaf span. Instead of providing a sudden reduction in spring span, a progressive shortening and stiffening of the spring occurs. Vans and light commercial vehicles have incorporated this unusual design of dual rate springing in the past, but the complicated combined swing shackle plate and spring makes this a rather expensive way of extending the spring rate from unladen to fully laden conditions.

10.13   andem and tri-a le bogies
A heavy goods vehicle is normally laden so that about two thirds or more of the total load is carried by the rear axle. Therefore the concentration of weight over a narrow portion of the chassis and on one axle, even between twin wheels, can be excessive.

In addition to the mechanical stresses imposed on the vehicle's suspension system, the subsoil stress distribution on the road for a single axle (Fig. 10.85(a)) is considerably greater than that for a tandem axle bogie (Fig. 10.85(b)) for similar payloads. Legislation in this country does not normally permit axle loads greater than ten tonne per axle. This weight limit prevents rapid deterioration of the road surface and at the same time spreads the majority of load widely along the chassis between two or even three rear axles.

The introduction of more than two axles per vehicle poses a major difficulty in keeping all the wheels in touch with the ground at the same time, particularly when driving over rough terrains (Fig. 10.86). This problem has been solved largely by having the suspensions of both rear axles interconnected so that if one axle rises relative to the chassis the other axle will automatically be lowered and wheel to road contact between axles will be fully maintained.

If twin rear axles are used it is with conventional half-elliptic springs supported by fixed front spring hangers and swinging rear spring shackle plates. If they are all mounted separately onto the chassis, when moving over a hump or dip in the road the front or rear axle will be lifted clear of the ground (Fig. 10.87) so that traction is lost for that particular axle and its wheels. The consequences of one or the other pairs of wheels losing contact with the road surface are that road-holding ability will be greatly reduced, large loads will suddenly be imposed on a single axle and an abnormally high amount of tyre scruffing will take place.
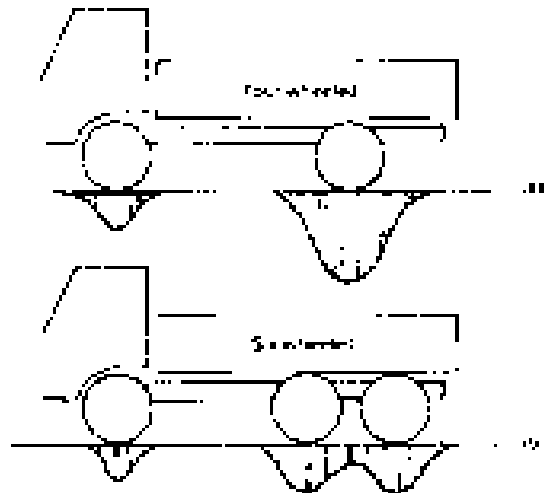
Fig. 10.85 (a and b)   Road stress distribution in subsoil underneath road wheels

421
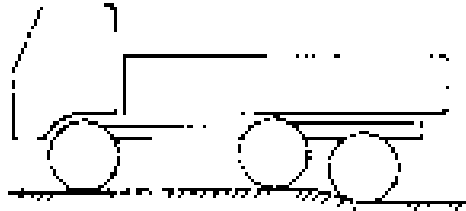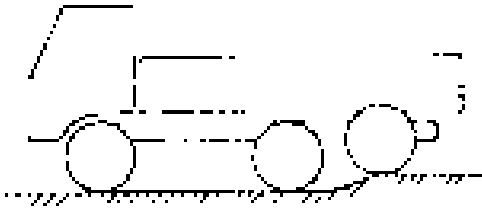
Fig. 10.86 Illustrating the need for tandem axle articulation
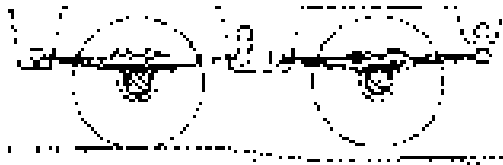
Fig. 10.87   ncompensated twin axle suspension

To share out the vehicle's laden weight between the rear tandem axles when travelling over irregular road surfaces, two basic suspension arrangements have been developed:

1 pivoting reactive or non-reactive balance beam which interconnects adjacent first and second semi-elliptic springs via their shackle plates,
2 a central pivoting single (sometimes double) vertical semi-elliptic spring which has an axle clamped to it at either end.

10.13.1 Equali ation of vehicle laden weight between axles (Figs 10.88 and 10.89)

Consider a reactive balance beam tandem axle bogie rolling over a hump or dip in the road (Fig. 10.88). The balance beam will tilt so that the rear end of the first axle is lifted upwards and the front end of the second axle will be forced downward. Consequently both pairs of axle wheels will be compelled to contact the ground and equally share out the static laden weight imposed on the whole axle bogie.

The tilting of the balance beam will lift the first axle a vertical distance $h/2$, which is half the hump

Fig. 10.88 Payload distribution with reactive balance beam and swing shackles
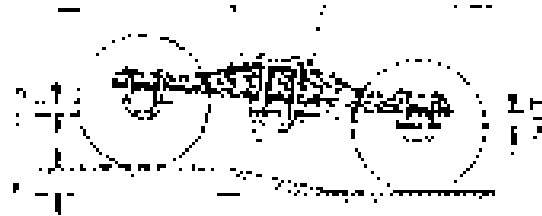
Fig. 10.89 Payload distribution with single inverted semi-elliptic spring

or dip's vertical height. The second axle will fall a similar distance $h/2$. The net result is that the chassis with the tandem axle bogie will only alter its height relative to ground by half the amount of a single axle suspension layout (Fig. 10.88). Thus the single axle suspension will lift or lower the chassis the same amount as the axle is raised or lowered from some level datum, whereas the tandem axle bogie only changes the chassis height relative to the ground by half the hump lift or dip drop.

In contrast to the halving of the vertical lift or fall movement of the chassis with tandem axles, there are two vertical movements with a tandem axle as opposed to one for a single axle each time the vehicle travels over a bump. Thus the frequency of the chassis vertical lift or fall with tandem axles will be twice that for a single axle arrangement.

Similar results will be achieved if a central pivoting inverted transverse spring tandem axle bogie rides over a hump or dip in the road (Fig. 10.89). Initially the first axle will be raised the same distances as the hump height h but the central pivot will only lift half the amount $h/2$. Conversely if the first axle goes into a dip, the second axle will be above the first axle by the height of the dip, but the chassis will only be lowered by half this vertical movement $h/2$. Again the frequency of lift and fall of the chassis as the tandem axles move over the irregularities in the road will be double the frequency compared to a single axle suspension.

422
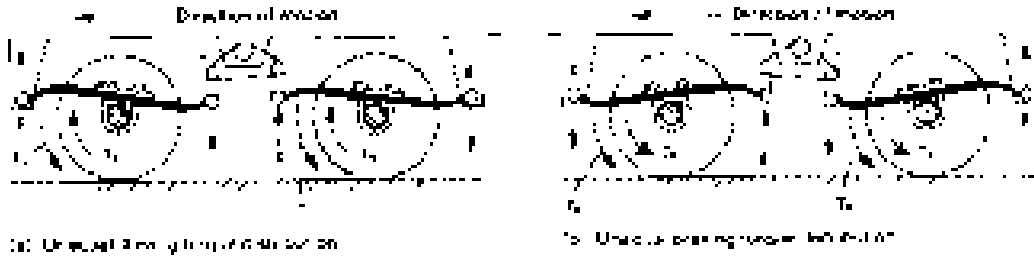
Fig. 10.90 (a and b) Reactive balance beam tandem axle suspension

10.13.2 Reactive balance beam tandem axle bogie suspension (Fig. 10.90(a and b))

Suspension arrangements of this type distribute the laden weight equally between the two axles due to the swing action of the balance beam (Fig. 10.90 (a and b)). The balance beam tilts according to the reaction load under each axle so that, within the chassis to ground height variation limitations, it constantly adjusts the relative lift or fall of each axle to suit the contour of the road.

Unfortunately the driving and braking torques produce unequal reaction through the spring linkage. Therefore under these conditions the vehicle's load will not be evenly distributed between axles.

Consider the situation when tractive effort is applied at the wheels when driving away from a standstill (Fig. 10.90(a)). Under these conditions the driving axle torque T_D produces an equal but opposite torque reaction T_R which tends to make the axle casing rotate in the opposite direction to that of the axle shaft and wheel. Subsequently the front spring ends of both axles tend to be lifted by force F and the rear spring ends are pulled downwards by force F Hence the overall reaction at each spring to chassis anchor point causes the balance beam to tilt anticlockwise and so lift the chassis away from the first axle, whereas the second axle is drawn towards the chassis. This results in the contact reaction between wheel and ground for the first axle to be far greater than for the second axle. In fact the second axle may even lose complete contact with the road.
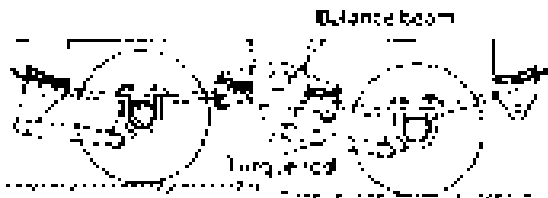
Fig. 10.91 Reactive balance beam with slipper contact blocks and torque arms tandem axle suspension
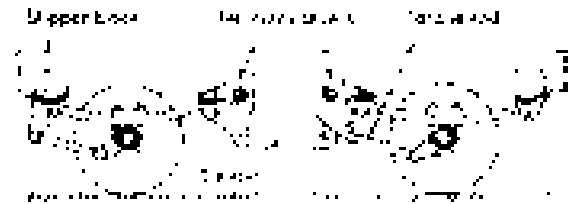
Fig. 10.92 Tandem wide spread reactive bell crank lever taper leaf spring

Conversely if the brakes are applied (Fig. 10.90(b)), the retarding but still rotating wheels will tend to drag the drum or disc brake assembly round with the axle casing T_R. The rotation of the axle casing in the same direction of rotation as the wheels means that the front spring ends of both axles will be pulled downward by force F The corresponding rear spring ends will be lifted upward by the reaction force F Thus in contrast to the driving torque directional reaction, the braking torque T_B will tilt the balance beam clockwise so that the second axle and wheel will tend to move away from the chassis, thereby coming firmly into contact with the road surface. The first axle and wheel will move further towards the chassis so that very little grip between the tyre and road occurs. In practice the upward lift of the first wheel and axle will cause the tyres to move in a series of hops and rebounds which will result in heavily loading the second axle, reducing the overall braking effectiveness and causing the first axle tyres to be subjected to excessive scuffing.

A reactive balance beam tandem axle bogie suspension using tapered leaf springs and torque arms to transmit the driving and braking forces and torques is shown in Fig. 10.91. With this layout driving and braking torque reactions will cause similar unequal load distribution.

To enable a wide spread axle to be used on trailers, the conventional reactive balance beam interconnecting spring linkage has been modified

so that laden vehicle weight can still be shared equally between axles. Thus instead of the central balance beam (Fig. 10.90) there are now two bell crank levers pivoting back to back on chassis spring hangers with a central tie rod (Fig. 10.92).

In operation, if the front wheel rolls over an obstacle its supporting spring will deflect and apply an upward thrust against the bell crank lever slipper. Accordingly, a clockwise turning moment will be applied to the pivoting lever. This movement is then conveyed to the rear bell crank lever via the tie rod, also making it rotate clockwise. Consequently the rear front end of the spring will be lowered, thus permitting the rear wheels to keep firmly in contact with the road while the chassis remains approximately horizontal.

10.13.3 Non-reactive bell crank lever and rod tandem axle bogie suspension
(Fig. 10.93(a and b))
To overcome the unequal load distribution which occurs with the reactive balance beam suspension when either driving or braking, a non-reactive bell crank lever and rod linkage has been developed which automatically feeds similar directional reaction forces to both axle rear spring end supports (Fig. 10.93(a and b)).

Both axle spring end reactions are made to balance each other by a pair of bell crank levers mounted back to back on the side of the chassis via pivot pins. Each axle rear spring end is attached by a shackle plate to the horizontal bell crank lever ends while the vertical bell crank lever ends are interconnected by a horizontally positioned rod.

When the vehicle is being driven (Fig. 10.93(a)) both axle casings react by trying to rotate in the opposite direction to that of the wheels so that the axle springs at their rear ends are pulled downward. The immediate response is that both bell crank levers will tend to twist in the opposite direction to each other, but this is resisted by the connecting rod which is put into compression. Thus the rear

end of each axle spring remains at the same height relative to the chassis and both axles will equally share the vehicle's laden weight.

Applying the brakes (Fig. 10.93(b)) causes the axle casings to rotate in the same direction as the wheels so that both axle springs at their rear ends will tend to lift. Both rear spring ends are attached to the horizontal ends of the bell crank levers. Therefore they will attempt to rotate in the opposite direction to each other, but any actual movement is prevented by the interconnected rod which will be subjected to a tensile force. Therefore equal braking torques are applied to each axle and equal turning moments are imposed on each bell crank lever which neutralizes any brake reaction in the suspension linkage. Since there is no interference with the suspension height adjustment during braking, the load distribution will be equalized between axles, which will greatly improve brake performance.

10.13.4 nverted semi-elliptic spring centrally pivoted tandem axle bogie suspension
(Figs 10.94, 10.95 and 10.96)
This type of tandem axle suspension has either one or two semi-elliptic springs mounted on central pivots which form part of the chassis side members. The single springs may be low (Fig. 10.94) or high (Fig. 10.95) mounted. To absorb driving and braking torque reaction, horizontally positioned torque arms are linked between the extended chassis side members and the axle casing. If progressive slipper spring ends are used (Fig. 10.95), double torque arms are inclined so that all driving and braking torque reactions are transmitted through these arms and only the vehicle's laden vertical load is carried by the springs themselves.

Articulation of the axles is achieved by the inverted springs tilting on their pivots so that one axle will be raised while the other one is lowered when negotiating a hump or dip in the road. As the axles move up and down relative to the central pivots, the torque arms will also pivot on their
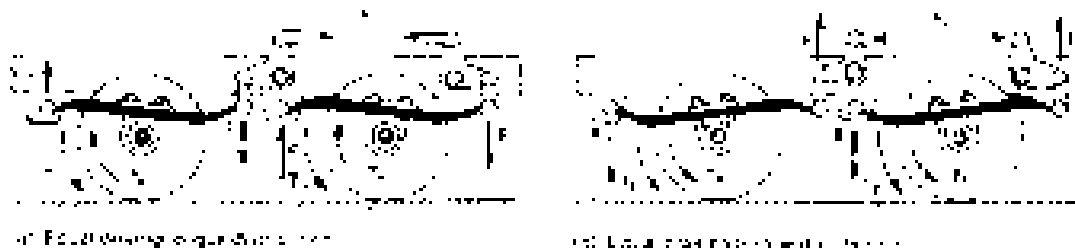
Fig. 10.93 (a and b)   Non-reaction bell crank lever and rod
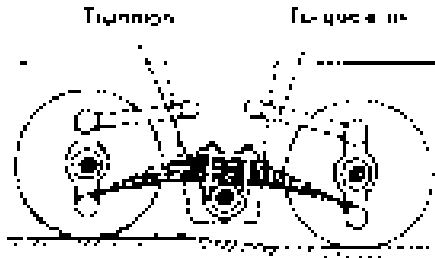
Fig. 10.94 Low mounted single inverted semi-elliptic spring with upper torque rods
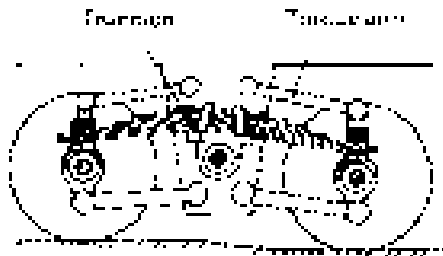
Fig. 10.95 High mounted single inverted semi-elliptic spring with lower torque rods

rubber end joints. Therefore the axle casing vertical arms will remain approximately upright at all times.

Any driving or braking reaction torque is transmitted through both the springs and torque arms to the central spring pivot and torque rod joint pins mounted on the reinforced and extended chassis side members. Very little interference is experienced with the load distribution between the two axles when the vehicle is being accelerated or retarded.

For heavy duty cross-country applications the double inverted semi-elliptic spring suspension is particularly suitable (Fig. 10.96). The double inverted spring suspension and the central spring

pivots, enable the springs to swivel a large amount (up to a 500 mm height difference between opposite axles) about their pivots when both pairs of axle wheels roll continuously over very uneven ground. This arrangement tolerates a great deal more longitudinal axle articulation than the single inverted spring and torque arm suspension.

Large amounts of transverse (cross) articulation are made possible by attaching the upper and lower spring ends to a common gimbal bracket which is loosely mounted over the axle casing (Fig. 10.96). The gimbal brackets themselves are supported on horizontal pivot pins anchored rigidly to the casing. This allows the axle to tilt transversely relative to the bracket's springs and chassis without causing any spring twist or excessive stress concentrations between flexing components.

10.13.5 Alternative tandem axle bogie arrangements

Leading and trailing arms with inverted semi-elliptic spring suspension (Fig. 10.97) An interesting tandem axle arrangement which has been used for recovery vehicles and tractor units where the laden to unladen ratio is high is the inverted semi-elliptic spring with leading and trailing arm (Fig. 10.97). The spring and arms pivot on a central chassis member; the arm forms a right angle with its horizontal portion providing the swing arm, while the vertical upper portion is shaped to form a curved slipper block bearing against the end of the horizontal semi-elliptic leaf spring.

The upper faces of the horizontal swing arm are also curved and are in contact with a centrally mounted 'V' -shaped member which becomes effective only when the tandem axle bogie is about half laden. Initially in the unladen state, both swing arms are supported only by the full spring length; this therefore provides a relatively low spring stiffness. As the axles become loaded, the leading and
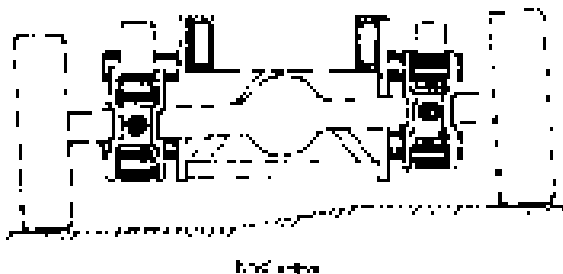
Fig. 10.96 Double inverted semi-elliptic spring
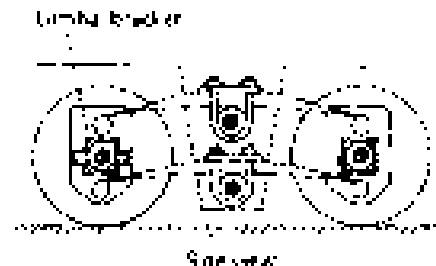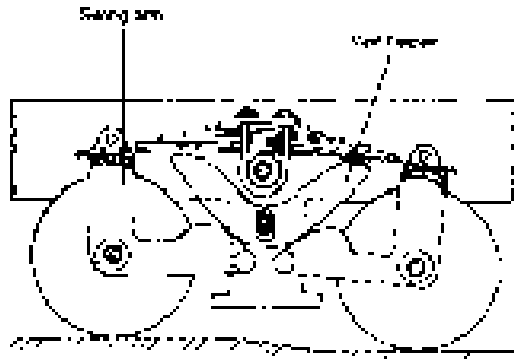
Fig. 10.97 Leading and trailing arms with inverted semi-elliptic spring

trailing arms pivot and swing upward, thereby steadily pushing the central 'V' helper member into contact with the main spring leaf over a much shorter blade span. The rolling contact movement between the upper and lower faces of the swing arms and the central 'V' helper member produce a progressive stiffening of the main spring under laden conditions.

Hendrickson long equali ation balance beam with single semi-elliptic springs (Fig. 10.98) This tandem suspension arrangement uses a low mounted, centrally pivoted long balance beam spanning the distance between axles and high mounted leading and trailing torque rods (Fig. 10.98). A semi-elliptic spring supports the vehicle's payload. It is anchored at the front end to a spring hanger and at the rear bears against either the outer or both inner and outer curved slipper hangers. The balance beam is attached to the spring by the 'U' bolts via its pivot mount.

The spring provides support for the vehicle's weight and transmits the accelerating or decelerat-ing thrusts between the axles and chassis. The balance beam divides the vehicle's laden weight between the axles and in conjunction with the torque rods absorbs the driving and braking torque reaction. The two stage spring stiffness is controlled by the effective spring span, which in the unladen condition spans the full spring length to the outer slipper block and in the laden state is shortened as the spring deflects, so that it now touches the inner slipper block spring hanger. For some cross-country applications the outer slipper block hanger is not incorporated so that there is only a slight progressive stiffening due to the spring blade to curved slipper block rolling action as the spring deflects with increasing load. With this four point chassis frame mounting and rigid balance beam, both the springs and the chassis are protected against concentrated stress which therefore makes this layout suitable for on/off rigid six or eight wheel rigid tracks.

Pivot beam with single semi-elliptic spring (Figs 10.99 and 10.100) This kind of suspension has a single semi-elliptic spring attached at the front end directly to a spring hanger and at the rear to a pivoting beam which carries the trailing axle (Fig. 10.99).

With a conventional semi-elliptic spring suspension, the fixed and swing shackles both share half ( ) of the reaction force imposed on the chassis caused by an axle load
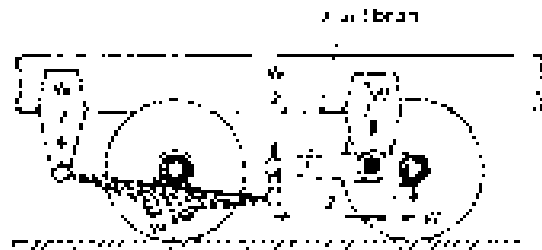
Fig. 10.99 Pivot beam with single semi-elliptic spring
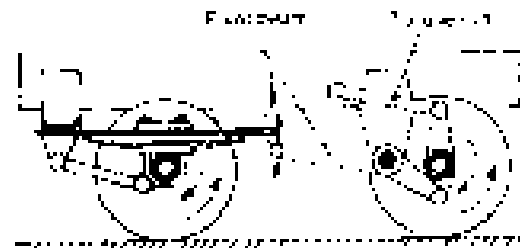
Fig. 10.100 Pivot beam with semi-elliptic spring and torque rod

Fig. 10.98 Hendrickson long equalization balance beam with single semi-elliptic spring

426

With the pivoting balance beam coupled to the tail-end of the spring, half the leading axle load ( ) reacting at the swing shackle is used to balance the load supported by the trailing axle. For the chassis laden weight to be shared equally between axles, the length of beam from the pivot to the shackle plate must be twice the trailing distance from the pivot to the axle. This means that if the load reaction at each axle is then with the leading axle clamped to the centre of the spring span and with a pivot beam length ratio of 2:1 the upward reaction force on the front spring hanger will be and that acting through the pivot 1 giving a total upward reaction force of + 1 = 2 . In other words, the downward force at the front of the pivot beam caused by the trailing axle supported by the pivot is balanced by the upward force at the rear end of the spring caused by the load on the leading axle. Thus if the front wheel lifts as it rolls over a bump, the trailing end of the spring rises twice as much as the axle. It attempts to push the trailing axle down so that its wheels are in hard contact with the ground.

With the second axle mounted between the lower trailing arm and the upper torque rod (Fig. 10.100), most of the driving and braking torque reaction is neutralized. Only when accelerating with a single drive axle is there some weight transfer from the non-drive axle (second) to the drive axle (first).

By arranging the first axle to be underslung (Fig. 10.100) instead of overslung (Fig. 10.99), a wider spring base projected to the ground will result in greater roll resistance.

railing arm with progressive quarter-elliptic spring (Fig. 10.101) Each axle is carried on a trailing arm; the arms on one side are interconnected by a spring in such a way that the upward reaction at one wheel increases the downward load on the other (Fig. 10.101). The inverted quarter-elliptic spring is clamped to the rear trailing arm.
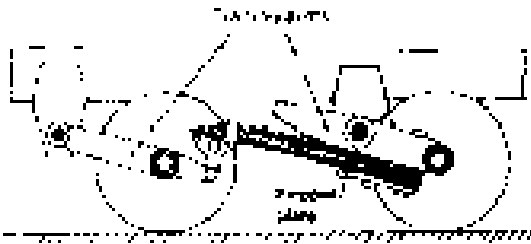
Fig. 10.101 Trailing arm with progressive quarter-elliptic spring

Its leading end is shackled to a bracket on the front trailing arm. Both trailing arms are welded fabricated steel members of box-section. The attachment of the quarter-elliptic springs to the rear trailing arms is so arranged that as the spring deflects on bump a greater length of spring comes into contact with the curved surface of the arm, thereby reducing the effective spring length with a corresponding increase in stiffness. On rebound, the keeper plate beneath the spring is extended forward and curved downward so that there is some progressive stiffening of the spring also on rebound. With this effective spring length control, the trailer will ride softly and easily when unladen and yet the suspension will be able to give adequate upward support when the trailer is fully laden.

ri-a le semi-trailer suspension (Fig. 10.102(a and b)) Tri-axle bogies are used exclusively on trailers. Therefore all these axles are dead and only laden weight distribution and braking torque reaction need to be considered.

The reactive balance beam interlinking between springs is arranged in such a way that an upward reaction at one wheel increases the downward load on the other, so that each of the three axles supports one third of the laden load (Fig. 10.102(a)).

The load distribution between axles is not quite so simple when the vehicle is being braked, owing to torque reaction making the axle casings rotate in the opposite sense to that of the road wheels. Consequently the foremost end of each spring tends to pull downwards while the rearmost spring ends push upwards. Accordingly the balance beams will react and therefore tilt clockwise. The net change in axle height relative to the chassis is that as the first axle is raised slightly so that tyre to road contact is reduced, the second axle experiences very little height change since the spring front end is made to dip while the rear end is lifted, and the third axle is forced downwards which increases the axle load and the tyre to road contact grip. This uneven axle load distribution under braking conditions is however acceptable since it does not appear to greatly affect the braking efficiency or to cause excessive tyre wear.

One problem with tri-axle trailers is that it is difficult and even impossible to achieve true rolling for all wheels when moving on a curved track due to the large wheel span of the three parallel axles, thus these layouts can suffer from excessive tyre scrub. This difficulty can be partially remedied by using only single wheels on the foremost and rearmost axles with the conventional twin wheels on
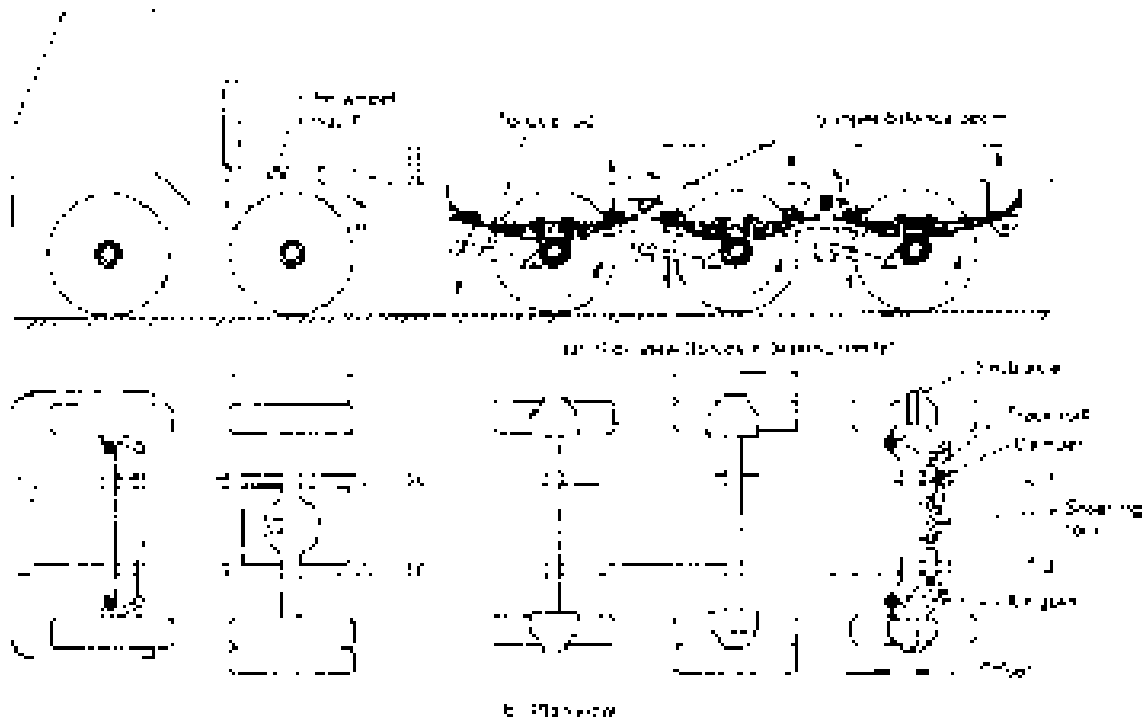
Fig. 10.102 (a and b) Tri-axle semi-trailer with self-steer axle

the middle axle (Fig. 10.102(b)). An alternative and more effective method is to convert the third axle into a self-steer one. Self-steer axles, when incorporated as part of the rearmost axle, not only considerably reduce tyre scrub but also minimize trailer cut-in because of the extent that the rear end is kicked out when cornering. Not only do self-steer axles improve tri-axle wheel tracking but they are also justified for tandem axle use.

Self-steer a le (Fig. 10.102(b))   The self-steer axle has a conventional axle beam with kingpin bosses swept forward to that of the stub axle centre line to provide the offset positive castor trail (Fig. 10.102(b)). Consequently the cornering side thrust on the tyre walls causes the wheels to turn the offset kingpins into line with the vehicle's directional steered path being followed. Excessive movement of either wheel about its kingpin is counteracted by the opposite wheel through the interconnecting track rod, while the trail distance between the kingpin and stub axle provides an automatic self-righting action when the vehicle comes out of a turn.

Possible oscillation on the stub axles is absorbed by a pair of heavy-duty dampers which become very effective at speed, particularly if the wheels are out of balance or misaligned.

Since the positive castor trail is only suitable for moving in the forward direction, when the vehicle reverses the wheels would tend to twitch and swing in all directions. Therefore, when the vehicle is being reversed, the stub axles are locked by a pin in the straight ahead position, this operation being controlled by the driver in the cab. The vehicle therefore behaves as if all the rear wheels are attached to rigid axles.

10.14 Rubber spring suspension

10.14.1 Rubber springs mounted on balance beam with stabili ing torque rods (Fig. 10.103)
Suspension rubber springs are made from alternatively bonded layers of rubber blocks and steel reinforcement plates sandwiched between inclined mounting plates so that the rubber is subjected to a combination of both shear and compressive forces. The rubber springs are mounted between the chassis spring cradle and a centrally pivoted wedge-shaped load transfer member (Fig. 10.103). The load between the two axles is equalized by a box-sectioned balance beam which is centrally mounted by a pivot to the load transfer member. To eliminate brake torque reaction, upper 'A'

428
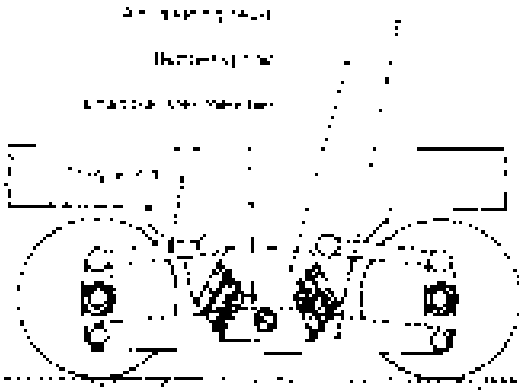
Fig. 10.103 Rubber spring mounted on balance beam with leading and trailing torque arms

brackets or torque arms are linked between the axles and chassis. With a pair of inclined rubber springs positioned on both sides of the chassis, loading of the axles produces a progressive rising spring rate due to the stress imposed into the rubber, changing from shear to compression as the laden weight rises.

The axles are permitted to articulate to take up any variation in road surface unevenness independently of the amount the laden weight of the vehicle has caused the rubber springs to deflect.

All pivot joints are rubber bushed to eliminate lubrication.

These rubber spring suspensions can operate with a large amount of axle articulation and are suitable for non-drive tandem trailers, rigid trucks with tandem drive axles and bulk carrier tankers.

10.14.2 Rubber spring mounted on leading and trailing arms interlinked by balance beam
(Fig. 10.104(a, b and c))
This tandem axle suspension is comprised of leading and trailing swing arms pivoting at their inner ends on the downward extending chassis frame with their outer ends clamped to the axle casings (Fig. 10.104(a)). The front and rear rubber springs are sandwiched between swing arm rigid mounting plates and a centrally pivoting balance beam. When in position these springs are at an inclined angle and are therefore subjected to a combination of compression and shear force.

When the swing arms articulate the spring mounting plate faces swivel and move in arcs. Thus the nature of the spring loading changes from a mainly shear action with very little compressive loading when the axles are unladen (Fig. 10.104(b)) to much greater compressive loading

and very little shear as the axles become fully laden (Fig. 10.104(c)). Since the rubber springs are about 14 times stiffer in compression than shear, the springs become progressively harder as the swing arms deflect with increasing laden weight. If the first axle is deflected upward as it moves over a bump, the increased compressive load acting on the spring will tilt the balance beam so that an equal increase in load will be transferred to the second axle.

Because the axles are mounted at the ends of the swing arms and the springs are positioned nearer to the pivot centres, axle movement will be greater than spring deflection. Therefore the overall suspension spring stiffness is considerably reduced for the ratio of axle and spring plate distance from the swing arm pivot centre which accordingly lowers the bounce frequency by 30%. Both leading and trailing swing arms absorb the braking torque reaction so that load distribution between axles will be approximately equal.

10.14.3 illetts (velvet ride) leading and trailing arm torsional rubber spring suspension
(Fig. 10.105(a and b))
The tandem suspension consists of leading and trailing swing arms. These arms are mounted back to back with their outer ends attached to the first and second drive axles whereas the swivel ends are supported on central trunnion pivot tubes which are mounted on a frame cross-member on either side of the chassis (Fig. 10.105(a)).

Torque arms attached to the suspension cross-member and to brackets in the centres of each axle casing assist the swing arms to transfer driving and braking torque reaction back to the chassis. These stabilizing torque arms also maintain the axles at the correct angular position. Good drive shaft geometry during articulation is obtained by the torque arms maintaining the axles at their correct angular position. Panhard rods (transverse tracking arms) between the frame side-members and the axle casings provide positive axle control and wheel tracking alignment laterally.

The spring consists of inner and outer annular shaped rubber members which are subjected to both torsional and vertical static deflection (Fig. 10.105(b)). The inner rubber member is bonded on the inside to the pivot tube which is supported by the suspension cross-member and on the outside to a steel half shell.

The outer rubber member is bonded on the inside to a median ring and on the outside to two half shells. The inside of the median ring is profiled
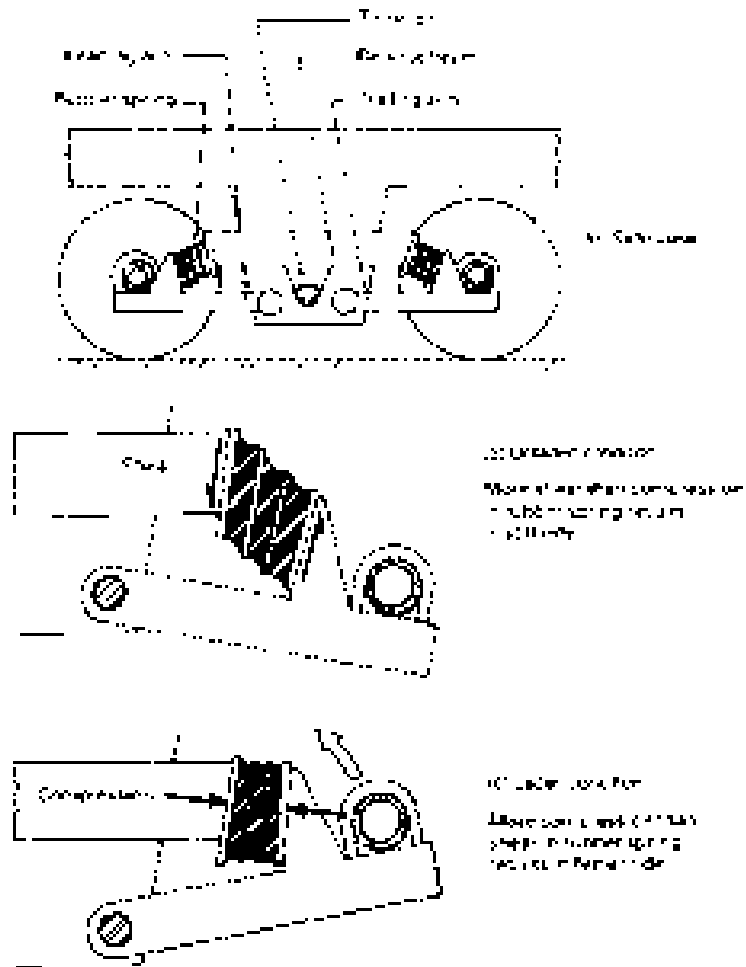
Fig. 10.104 (a–c) Rubber springs mounted leading and trailing arms interlinked by rocking beam
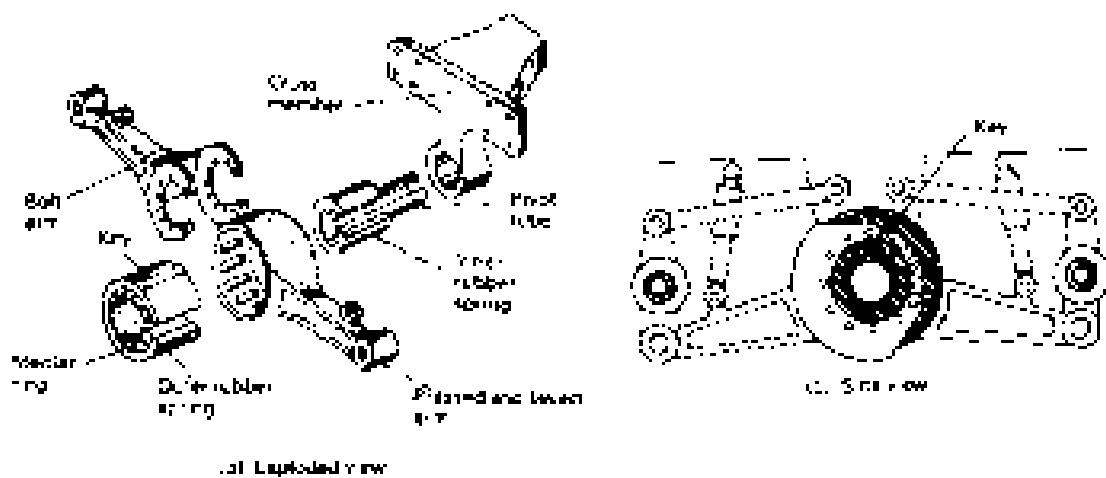
Fig. 10.105 (a and b) Willetts (velvet) leading and trailing arm torsional rubber spring suspension

430

to the same shape as the inner rubber member and half shell thus preventing inter rotation between the inner and outer rubber members. ey abutments are formed on the circumference of each outer half shell. These keys are used to locate (index) the outer rubber spring members relative to the trailing pressed (keyed) swing arm (Fig. 10.105(b)).

When assembled, the outer rubber member fits over the inner rubber member and half shell, whereas the trailing swing arm spring aperture is a press fit over the outer pair of half shells which are bonded to the outer rubber member. The leading swing arm side plates fit on either side of the median ring and aligned bolt holes enable the two members to be bolted together (Fig. 10.105(a and b)).

Load equalization between axles is achieved by torsional wind-up of the rubber spring members. Thus any vertical deflection of one or other swing arm as the wheels roll over any bumps on the road causes a torque to be applied to the rubber members. Accordingly an equal torque reaction will be transferred through the media of the rubber to the other swing arm and axle. As a result, each axle will support an equal share of the laden weight. Therefore contact and grip between wheels of both axles will be maintained at all times.

The characteristic of this springing is a very low stiffness in the unladen state which therefore provides a soft ride. A progressive spring stiffening and hardness of ride occurs as the swing arms are made to deflect against an increase in laden weight. An overall cushioned and smoothness of ride results.

An additional feature of this suspension geometry is that when weight is transferred during cornering from the inside to the outside of the vehicle, the deflection of the swing arms spreads the outer pair of wheels and draws the inner pair of wheels closer together. As a result smaller turning circles can be achieved without excessive tyre scrub.

10.15 Air suspensions for commercial vehicles

A rigid six wheel truck equipped with pairs of air springs per axle is shown in Fig. 10.106. The front suspension has an air spring mounted between the underside of each chassis side-member and the transverse axle beam, and the rear tandem suspension has the air springs mounted between each trailing arm and the underside of the chassis (Figs 10.107 and 10.108).

Air from the engine compressor passes through both the unloader valve and the pressure regulator valve to the reservoir tank. Air is also delivered to the brake system reservoir (not shown). Once the compressed air has reached some pre-determined upper pressure limit, usually between 8 and 8.25 bar, the unloader valve exhausts any further air delivery from the pump directly to the atmosphere, thereby permitting the compressor to 'run light'. Immediately the air supply to the reservoir has dropped to a lower limit of 7.25 bar, the unloader valve will automatically close its exhaust valve so that air is now transferred straight to the reservoir to replenish the air consumed. Because the level of air pressure demanded by the brakes is greater than that for the suspension system, a pressure regulator valve is incorporated between the unloader valve and suspension reservoir valve, its function being to reduce the delivery pressure for the suspension to approximately 5.5 bar.

Air now flows from the suspension reservoir through a filter and junction towards both the front and rear suspensions by way of a single
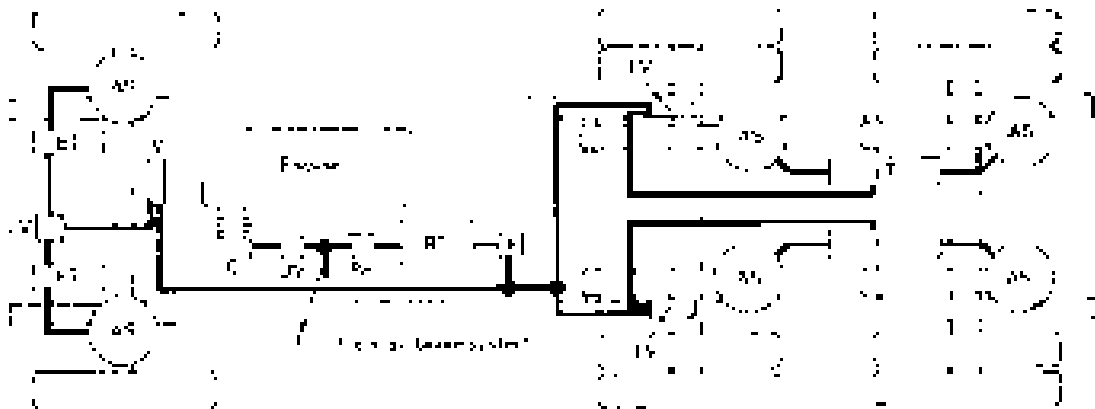
Fig. 10.106 Air spring suspension plan view layout

Fig. 10.107 Tandem trailing arm rolling diaphragm air sprung suspension

Fig. 10.108 Tandem trailing arm bellows spring suspension with rubber anti-roll blocks

central levelling valve at the front (Fig. 10.109) and a pair of levelling valves on each side of the first tandem axle. These levelling valves are bolted to the chassis, but they are actuated by an arm and link rod attached to the axles. It is the levelling valves' function to sense any change in the chassis to axle height and to increase or decrease the air pressure supply passing to the air springs, thereby raising or reducing the chassis height respectively. The air pressure actually reaching the springs may vary from 5.5 bar fully laden down to 2.5 bar when the vehicle is empty.

To improve the quality of ride, extra volume tanks can be installed in conjunction with the air springs to increase the volume of air in the system. This minimizes changes in overall pressure and reduces the spring rate (spring stiffness), thus enabling the air springs to provide their optimum frequency of spring bounce.

An additional feature at the front end of the suspension is an isolating valve which acts both as a junction to split the air delivery to the left and right hand air springs and to permit air to pass immediately to both air springs if there is a demand for more compressed air. This valve also slows down the transfer of air from the outer spring to the inner spring when the body rolls while the vehicle is cornering.

10.15.1 *Levelling valve* (Figs 10.109 and 10.110(a and b))

A pre-determined time delay before air is allowed to flow to or from the air spring is built into the valve unit. This ensures that the valves are not operated by axle bump or rebound movement as the vehicle rides over rough road surfaces, or by increased loads caused by the roll of the body on prolonged bends or on highly cambered roads.

The valve unit consists of two parts; a hydraulic damper and the air control valve (Fig. 10.110(a and b)). Both the damper and the valves are actuated by the horizontal operating lever attached to the axle via a vertical link rod. The operating lever pivots on a cam spindle mounted in the top of the valve assembly housing. The swing movement of the operating lever is relayed to the actuating arm through a pair of parallel positioned leaf springs fixed rigidly against the top and bottom faces of the flat cam, which forms an integral part of the spindle.

When the operating lever is raised or lowered, the parallel leaf springs attached to the lever casing pivot about the cam spindle. This causes both leaf springs to deflect outwards and at the same time
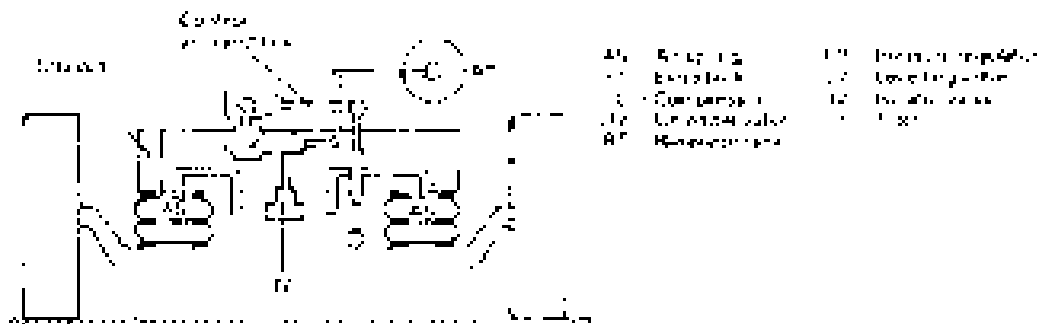
Fig. 10.109 Air spring suspension front view layout

432
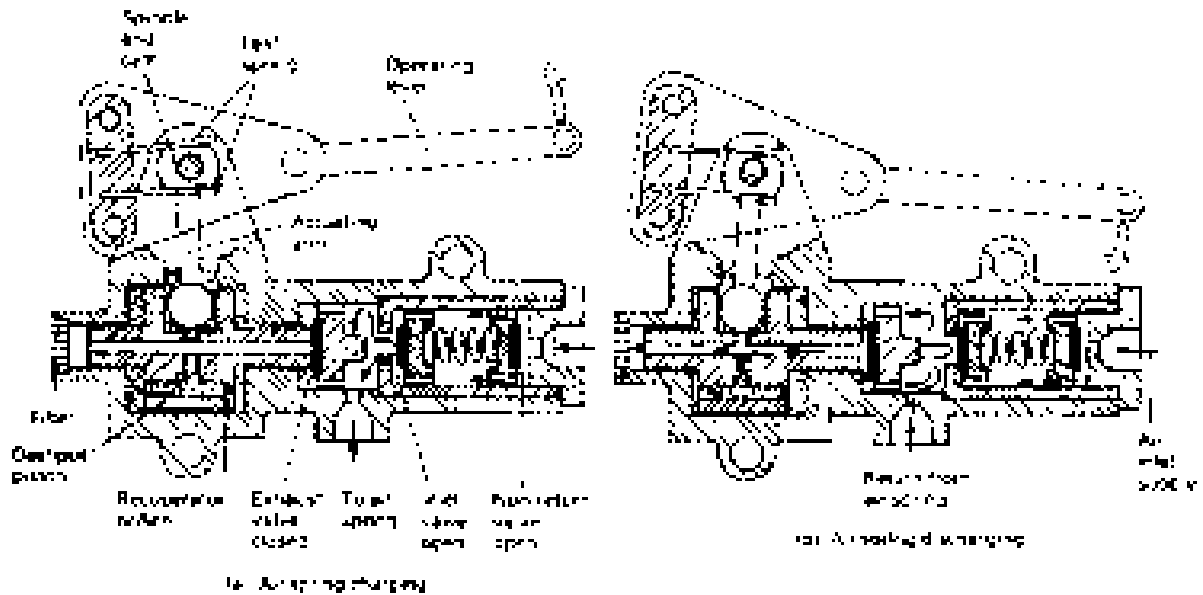
Fig. 10.110 (a and b) Levelling air control valve

applies a twisting movement to the cam spindle. It therefore tends to tilt the attached actuating arm and accordingly the dashpot piston will move either to the right or left against the fluid resistance. There will be a small time delay before the fluid has had time to escape from the compressed fluid side of the piston to the opposite side via the clearance between the piston and cylinder wall, after which the piston will move over progressively. A delay of 8 to 12 seconds on the adjustment of air pressure has been found suitable, making the levelling valve inoperative under normal road surface driving conditions.

Vehicle being loaded (Fig. 10.110(a)) If the operating lever is swung upward, due to an increase in laden weight, the piston will move to the right, causing the tubular extension of the piston to close the exhaust valve and the exhaust valve stem to push open the inlet valve. Air will then flow past the non-return valve through the centre of the inlet valve to the respective air springs. Delivery of air will continue until the predetermined chassis-to-axle height is reached, at which point the lever arm will have swung down to move the piston to the left sufficiently to close the inlet valve. In this phase, the springs neither receive nor lose air. It is therefore the normal operating position for the levelling valve and springs.

Vehicle being unloaded (Fig.10.110(b)) If the vehicle is partially unloaded, the chassis will rise relative to the axle, causing the operating arm to swing downward. Consequently, the piston will move to the left so that the exhaust valve will now reach the end of the cylinder. Further piston movement to the left will pull the tubular extension of the piston away from its rubber seat thus opening the exhaust valve. Excessive air will now escape through the centre of the piston to the atmosphere until the correct vehicle height has been established. At this point the operating lever will begin to move the piston in the opposite direction, closing the exhaust valve. This cycle of events will be repeated as the vehicle's laden weight changes. A non-return valve is incorporated on the inlet side to prevent air loss from the spring until under maximum loading or if the air supply from the reservoir should fail.

10.15.2 solating valve (Fig. 10.111(a and b))
An isolating valve is necessary when cornering to prevent air being pumped from the spring under compression to that under expansion, which could considerably reduce body roll resistance.

The valve consists of a T-piece pipe air supply junction with a central cylinder and plunger valve (Fig. 10.111(a and b)).

When the air springs are being charged, compressed air enters the inlet part of the valve from
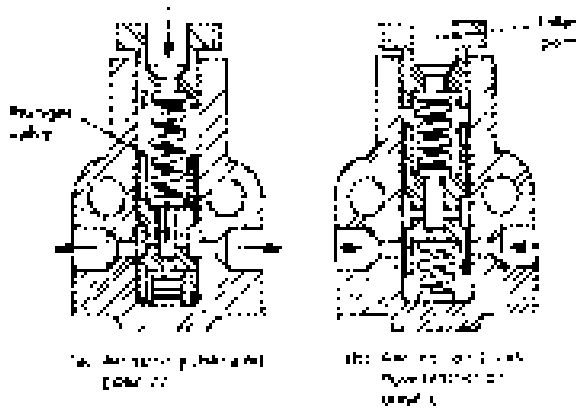
Fig. 10.111 (a and b) Isolator valve

the levelling valve and pushes the shuttle valve towards the end of its stroke against the spring situated between the plunger and cylinder blank end (Fig. 10.111(a)). Air will pass through the centre of the valve and come out radially where the annular groove around the valve aligns with the left and right hand output ports which are connected by pipe to the air springs.

Once the levelling valve has shut off the air supply to the air springs, the shuttle valve springs are free to force the shuttle valve some way back towards the inlet port. In this position the shuttle skirt seals both left and right hand outlet ports (Fig. 10.111(b)) preventing the highly pressurized outer spring from transferring its air charge to the expanded inner spring (which is subjected to much lower pressure under body roll conditions).

The shuttle valve is a loose fit in its cylinder to permit a slow leakage of air from one spring to the other should one spring be inflated more rapidly than the other, due possibly to uneven loading of the vehicle.

10.15.3 *Air spring bags* (Figs 10.112 and 10.113)

Air spring bags may be of the two or three convoluted bellows (Fig. 10.112) or rolling lobe (diaphragm) type (Fig. 10.113), each having distinct characteristics. In general, the bellows air spring
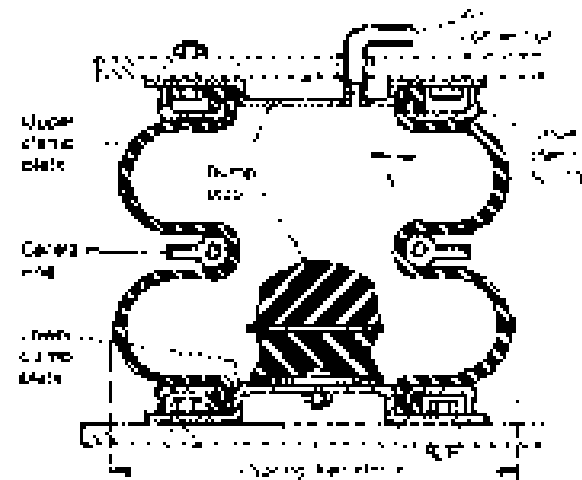
Fig. 10.112 Involute bellow spring
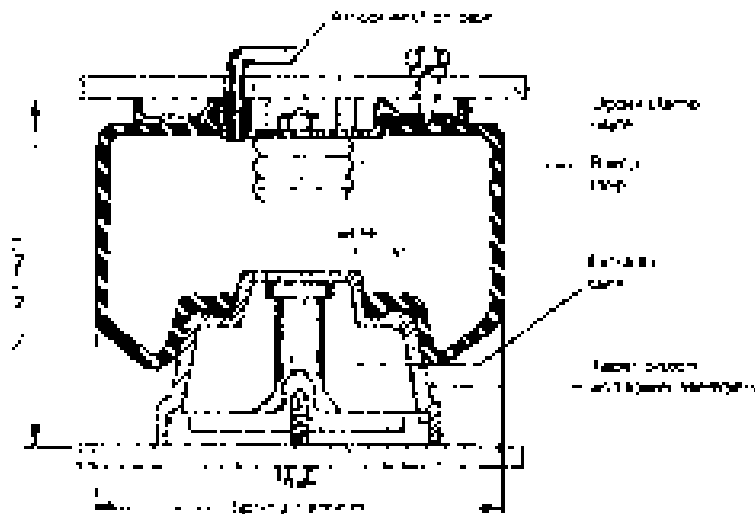
Fig. 10.113 Rolling diaphragm spring

434

(Fig. 10.112) is a compact flexible air container which may be loaded to relatively high load pressures. Its effective cross-sectional area changes with spring height   reducing with increase in static height and increasing with a reduction in static height. This is due to the squeezing together of the convolutes so that they spread further out. For large changes in static spring height, the three convolute bellows type is necessary, but for moderate suspension deflection the twin convolute bellow is capable of coping with the degree of expansion and contraction demanded.

With the rolling diaphragm or lobe spring (Fig. 10.113) a relatively higher installation space must be allowed at lower static pressures. Progressive spring stiffening can be achieved by tapering the skirt of the base member so that the effective working cross-sectional area of the rolling lobe increases as the spring approaches its maximum bump position.

The normal range of natural spring frequency for a simply supported mass when fully laden and acting in the direct mode is 90–150 cycles per minute (cpm) for the bellows spring and for the rolling lobe type 60–90 cpm. The higher natural frequency for the bellow spring compared to the rolling lobe type is due mainly to the more rigid construction of the convolute spring walls, as opposed to the easily collapsible rolling lobe.

As a precaution against the failure of the supply of air pressure for the springs, a rubber limit stop of the progressive type is assembled inside each air spring, and compression of the rubber begins when about 50 mm bump travel of the suspension occurs.

The springs are made from tough, nylon-reinforced Neoprene rubber for low and normal operating temperature conditions but Butyl rubber is sometimes preferred for high operating temperature environments.

An air spring bag is composed of a flexible cylindrical wall made from reinforced rubber enclosed by rigid metal end-members. The external wall profile of the air spring bag may be plain or bellow shaped. These flexible spring bags normally consist of two or more layers of rubber coated rayon or nylon cord laid in a cross-ply fashion with an outside layer of abrasion-resistant rubber and sometimes an additional internal layer of impermeable rubber to minimize the loss of air.

In the case of the bellow type springs, the air bags (Fig. 10.112) are located by an upper and lower clamp ring which wedges their rubber moulded edges against the clamp plate tapered spigots. The rolling lobe bag (Fig. 10.113) relies only upon the necks of the spring fitting tightly over the tapered and recessed rigid end-members. Both types of spring bags have flat annular upper and lower regions which, when exposed to the compressed air, force the pliable rubber against the end-members, thereby producing a self-sealing action.

10.15.4 Anti-roll rubber blocks (Fig. 10.108)
A conventional anti-roll bar can be incorporated between the trailing arms to increase the body roll stiffness of the suspension or alternatively built-in anti-roll rubber blocks can be adopted (Fig. 10.108). During equal bump or rebound travel of each wheel the trailing arms swing about their front pivots. However, when the vehicle is cornering, roll causes one arm to rise and the other to fall relative to the chassis frame. Articulation will occur at the rear end of the trailing arm where it is pivoted to the lower spring base and axle member. Under these conditions, the trailing arm assembly adjacent to the outer wheel puts the rubber blocks into compression, whereas in the other trailing arm, a tensile load is applied to the bolt beneath the rubber block. As a result, the total roll stiffness will be increased. The stiffness of these rubber blocks can be varied by adjusting the initial rubber compressive preload.

10.15.5 Air spring characteristics (Figs 10.114, 10.115, 10.116 and 10.117)
The bounce frequency of a spring decreases as the sprung weight increases and increases as this weight is reduced. This factor plays an important part in the quality of ride which can be obtained on a heavy goods or passenger vehicle where there could be a fully laden to unladen weight ratio of up to 5:1.

An inherent disadvantage of leaf, coil and solid rubber springs is that the bounce frequency of vibration increases considerably as the sprung spring mass is reduced (Fig. 10.114). Therefore, if a heavy goods vehicle is designed to give the best ride frequency, say 60 cycles per minute fully laden, then as this load is removed, the suspension's bounce frequency could rise to something like 300 cycles per minute when steel or solid rubber springs are used, which would produce a very harsh, uncomfortable ride. Air springs, on the other hand, can operate over a very narrow bounce frequency range with considerable changes in vehicle laden weight, say 60–110 cycles per minute for a rolling lobe air spring (Fig. 10.114). Consequently the quality of ride with air springs is maintained over a wide range of operating conditions.
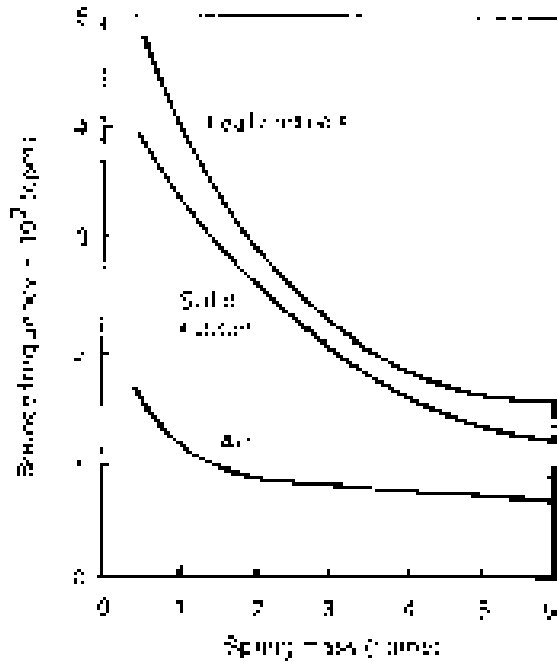
Fig. 10.114 Effects and comparison of payload on spring frequency for various types of spring media
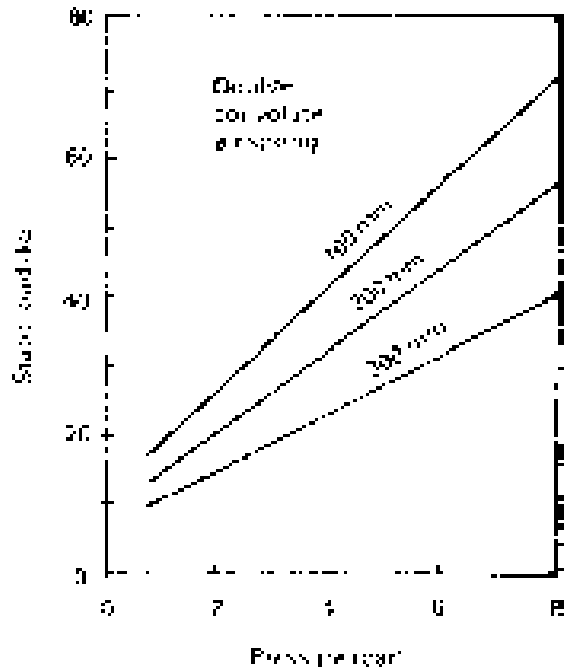
Fig. 10.116 Effects of static payload on spring air pressure for various spring static heights
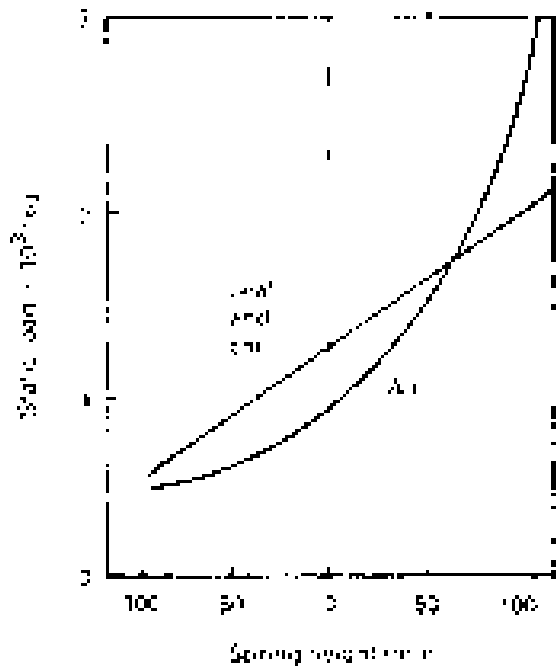
Fig. 10.115 Effects of static load on spring height
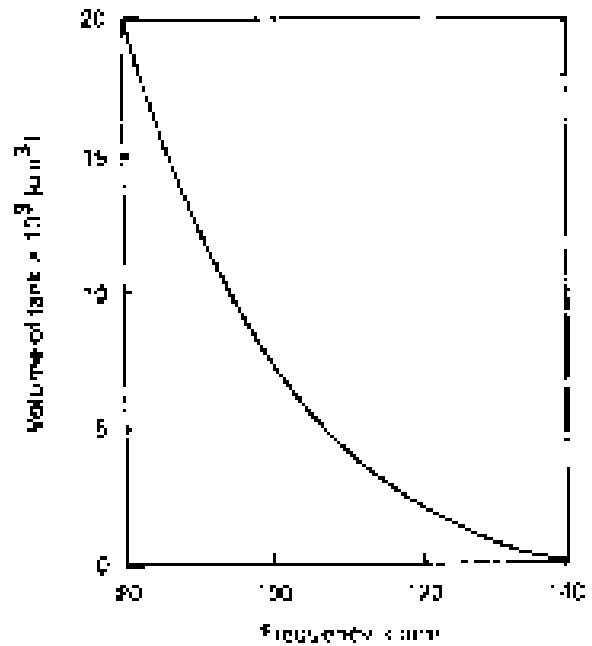
Fig. 10.117 Relationship of extra air tank volume and spring frequency

436

Steel springs provide a direct rise in vertical deflection as the spring mass increases, that is, they have a constant spring rate (stiffness) whereas air springs have a rising spring stiffness with increasing load due to their effective working area enlarging as the spring deflects (Fig. 10.115). This stiffening characteristic matches far better the increased resistance necessary to oppose the spring deflection as it approaches the maximum bump position.

To support and maintain the spring mass at constant spring height, the internal spring air pressure must be increased directly with any rise in laden weight. These characteristics are shown in Fig. 10.116 for three different set optimum spring heights.

The spring vibrating frequency will be changed by varying the total volume of air in both extra tank and spring bag (Fig. 10.117). The extra air tank capacity, if installed, is chosen to provide the optimum ride frequency for the vehicle when operating between the unladen and fully laden conditions.

10.16 Lift axle tandem or tri-axle suspension
(Figs 10.118, 10.119 and 10.120)
Vehicles with tandem or tri-axles which carry a variety of loads ranging from compact and heavy

to bulky but light may under-utilize the load carrying capacity of each axle, particularly an empty return journey over a relatively large proportion of the vehicle's operating time.

When a vehicle carries a full load, a multi-axle suspension is essential to meet the safety regulations, but the other aspects are improved road vibration isolation from the chassis, better road holding and adequate ride comfort.

If a conventional multi-axle suspension is operated below half its maximum load carrying capacity, the quality of road holding and ride deteriorates, suspension parts wear rapidly, and increased wheel bounce causes a rise in tyre scrub and subsequent tyre tread wear.

Conversely, reducing the number of axles and wheels in contact with the road when the payload is decreased extends tyre life, reduces rolling forward resistance of the vehicle and therefore improves fuel consumption.

10.16.1 Balance beam lift axle suspension arrangement (Figs 10.118 and 10.119)
A convenient type of tandem suspension which can be adapted so that one of the axles can be simply and rapidly raised or lowered to the ground

Fig. 10.118 (a and b)   Hydraulically operated lift axle suspension with direct acting ram

Fig. 10.119 (a and b)   Hydraulically operated lift axle suspension with bell-crank lever and ram

Fig. 10.120 (a and b) Pneumatically operated lift axle suspension

without having to make major structural changes is the *semi elliptic spring and balance beam combin ation* (Figs 10.118 and 10.119). Raising the rear-most of the two axles from the ground is achieved by tilting the balance beam anticlockwise so that the forward part of the balance beam appears to push down the rear end of the semi-elliptic spring. In effect, what really happens is the balance beam pivot mounting and chassis are lifted relative to the forward axle and wheels. Actuation of the balance beam tilt is obtained by a power cylinder and ram, anchored to the chassis at the cylinder end, whilst the ram-rod is connected either to a tilt lever, which is attached indirectly to the balance beam pivot, or to a bell crank lever, which relays motion to the extended forward half of the balance beam.

Balance beam suspension with tilt lever a le lift (Fig. 10.118(a and b)) With the tilt lever axle lift arrangement, applying the lift control lever introduces fluid under pressure to the power cylinder, causing the ram-rod to extend. This forces the tilt lever to pivot about its centre of rotation so that it bears down on the left hand side of the beam. Consequently the balance beam is made to take up an inclined position (Fig. 10.118(b)) which is sufficient to clear the rear road wheels off the ground. When the axle is lowered by releasing the hydraulic pressure in the power cylinder, the tilt lever returns to its upright position (Fig. 10.118(a)) and does not then interfere with the articulation of the balance beam as the axles deflect as the wheels ride over the irregularities of the road surface.

Balance beam suspension with bell-crank lever a le lift (Fig. 10.119(a and b)) An alternative lift axle arrangement uses a bell-crank lever to transmit the ram-rod force and movement to the

extended front end of the balance beam. When hydraulic pressure is directed to the power cylinder, the bell-crank lever is compelled to twist about its pivot, causing the roller to push down and so roll along the face of the extended balance beam until the rear axle is fully raised (Fig. 10.119(b)). Removing the fluid pressure permits the weight of the chassis to equalize the height of both axles again and to return the ram-rod to its innermost position (Fig. 10.119(a)). Under these conditions the bell-crank lever roller is lifted clear of the face of the balance beam. This prevents the oscillating motion of the balance beam being relayed back to the ram in its cylinder.

10.16.2 Pneumatically operated lift axle suspension (Fig. 10.120(a and b))

A popular lift axle arrangement which is used in conjunction with a trailing arm air spring suspension utilizes a separate single air bellow situated at chassis level in between the chassis side-members. A yoke beam supported by the lift air bellows spans the left and right hand suspension trailing arms, and to prevent the bellows tilting as they lift, a pair of pivoting guide arms are attached to the lift yoke on either side. To raise the axle wheels above ground level, the manual air control valve is moved to the raised position; this causes compressed air to exhaust from the suspension air springs and at the same time allows pressurized air to enter the lift bellows. As the air pressure in the lift bellows increases, the bellows expand upward, and in doing so, raise both trailing arm axle and wheels until they are well above ground level (Fig. 10.120(b)). Moving the air control valve to 'release' position reverses the process. Air will then be exhausted from the lift bellows while the air springs will be charged with compressed air so that the axle takes its full share of payload (Fig. 10.120(a)).

438

An additional feature of this type of suspension is an overload protection where, if the tandem suspension is operating with one axle lifted and receives loads in excess of the designed capacity, the second axle will automatically lower to compensate.

10.17 Active suspension

An ideal suspension system should be able to perform numerous functions that are listed below:

1 To absorb the bumps and rebounds imposed on the suspension from the road.
2 To control the degree of body roll when cornering.
3 To maintain the body height and to keep it on an even keel between light and full load conditions.
4 To prevent body dive and squat when the car is rapidly accelerated or is braked.
5 To provide a comfortable ride over rough roads yet maintain suspension firmness for good steering response.
6 To isolate small and large round irregularities from the body at both low and high vehicle speeds.

These demands on a conventional suspension are only partially achieved as to satisfy one or more of the listed requirements may be contrary to the fulfilment of some of the other desired suspension properties. For example, providing a soft springing for light loads will excessively reduce the body height when the vehicle is fully laden, or conversely, stiffening the springing to cope with heavy loads will produce a harsh suspension under light load conditions. Accordingly, most conventional suspensions may only satisfy the essential requirements and will compromise on some of the possibly less important considerations. An active suspension will have built into its design means to satisfy all of the listed demands; however, even then it may not be possible due to the limitations of a design and cost to meet and overcome all of the inherent problems experienced with vehicle suspension. Thus it would be justified to classify most suspensions which have some form of height levelling and anti-body roll features as only semi-active suspensions.

For an active suspension to operate effectively various sensors are installed around the vehicle to monitor changing driving conditions; the electrical signals provided by these sensors are continuously fed to the input of an electronic control unit microprocessor. The microprocessor evaluates and processes the data supplied by the sensors on the changing speed, loads, and driving conditions imposed on the suspension system. On the basis of these data and with the aid of a programmed map memory, calculations are made as to what adjustments should be made to the suspension variables. These instructions are then converted into electrical output signals and are then directed to the various levelling and stiffening solenoid control valves. The purpose of these control valves is to deliver or exit fluid to or from the various parts of a hydraulic controlled self-levelling suspension system.

10.17.1 Description and application of sensors

A list of sensors which can be used are given below; however, a limited combination of these sensors may only be installed depending on the sophistication of the suspension system adopted:

1 body height sensor
2 steering wheel sensor
3 longitudinal acceleration sensor
4 lateral acceleration sensor
5 brake pressure sensor
6 brake pedal sensor
7 acceleration pedal sensor
8 load sensor
9 vehicle speed sensor
10 mode selector

Height sensor (Fig. 10.121) The linear variable differential sensor is often used to monitor vertical height movement as there is no contact between moving parts; it therefore eliminates any problems likely to occur due to wear. It is basically a transformer having a central primary winding and two
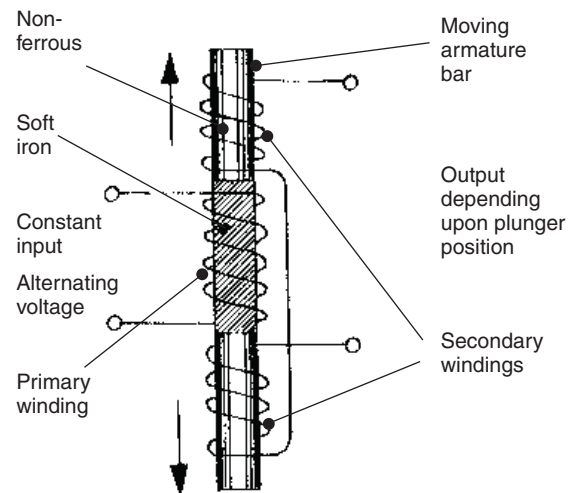
Fig. 10.121 Height sensor (linear variable differential type)

439

secondary windings connected in series in opposition to each other. An alternative input supply voltage is applied to the primary winding; this produces a magnetic flux which cuts through the secondary winding thereby inducing an alternative voltage into the secondary winding. The difference between the voltage generated in each secondary winding therefore becomes the output signal voltage. With the non-ferromagnetic/soft iron armature bar in the central position each secondary winding will generate an identical output voltage so that the resultant output voltage becomes zero. However when the armature (attached to the lower suspension arm) moves up or down as the body height changes the misalignment of the soft iron/non-ferro-magnetic armature causes the output voltage to increase in one winding and decrease in the other, the difference in voltage increasing in direct proportion to the armature displacement. This alternative voltage is then converted to a direct voltage before entering the electronic-control unit.

Steering sensor (Fig. 10.122)   This sensor monitors the angular position of the steering wheel and the rate of change of the steering angle. The sensor comprises a slit disc attached to the steering column and rotates with the steering wheel and a fixed 'U' shaped detector block containing on one side three phototransistors and on the other side three corresponding light-emitting diodes. The disc rotates with the steering column and wheel and at the same time the disc moves between the light-emitting diode and the phototransistor block overhang. When the column is turned the rotating slotted disc alternatively exposes and blocks the light-emitting beams directed towards the phototransistors; this interruption of the light beams generates a train of logic pulses which are then processed by the microprocessor to detect the steering angle and the rate of turn. To distinguish which way the steering wheel is turned a left and right hand phototransistor is included, and a third phototransistor is located between the other two to establish the neutral straight ahead position. The difference in time between light beam interruptions enables the microprocessor to calculate the angular velocity of the driving wheel at any one instance in time. In some active suspension systems, when the angular velocity exceeds a pre-fixed threshold the electronic-control unit switches the suspension to a firm ride mode.

Acceleration sensor (Fig. 10.123)   A pendulum strain gauge type acceleration sensor is commonly used for monitoring body acceleration in both longitudinal and lateral directions. It is comprised of a leaf spring rigidly supported at one end with a mass attached at its free end. A thin film strain gauge wired in the form of a wheatstone bridge circuit is bonded to the leaf spring on one side, two of the four resistors are passive whereas the other two are active. As the vehicle is accelerated the pendulum due to the inertia of the mass will reluctantly hold back thus causing the spring to deflect. The pair of active resistor arms therefore become strained (stretch) and hence alter their resistance, thus producing an imbalance to the wheatstone bridge circuit resulting in an output voltage proportional to the magnitude of the acceleration. When using this type of sensor for monitoring
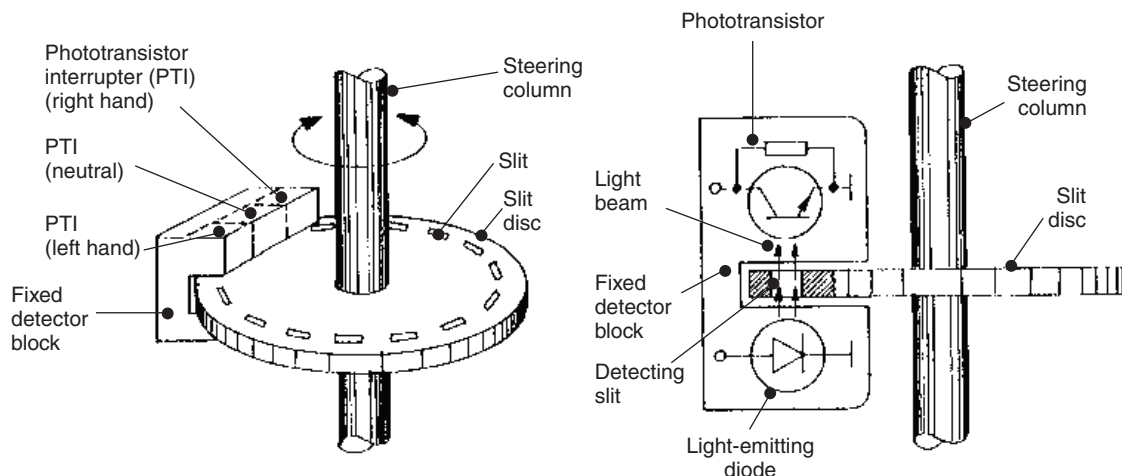
Fig. 10.122   Steering sensor (photo interrupter type)

440
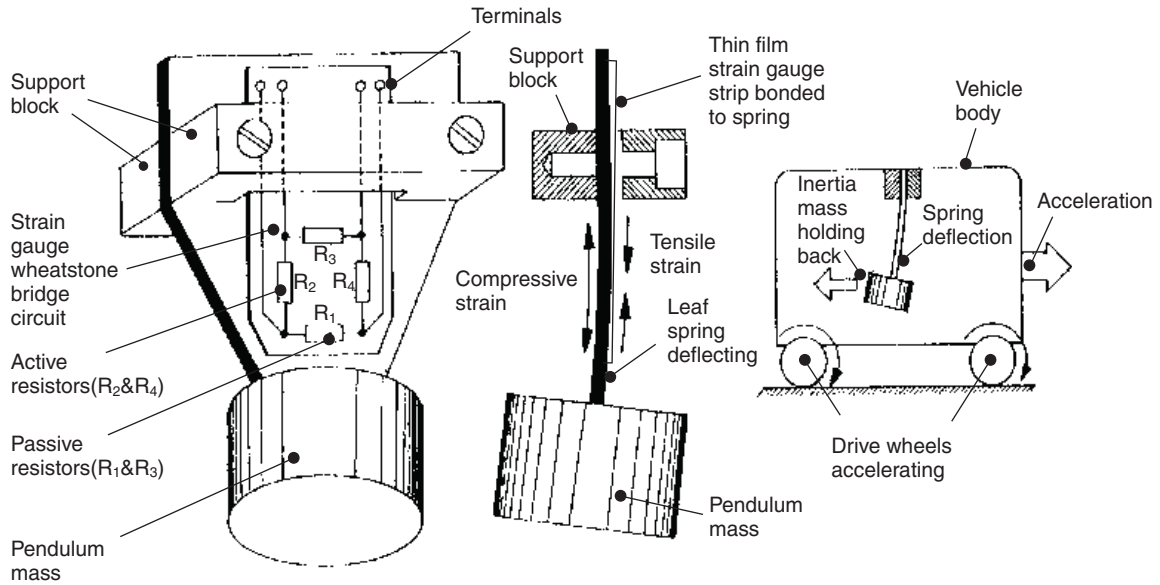
Fig. 10.123 Acceleration sensor (pendulum strain gauge type)

lateral acceleration, it should be installed either near the front or rear to enable it to sense the swing of the body when the car is cornering it, there is also a measure in the degree of body yaw.

Brake pedal/pressure sensor These sensors are used to indicate the driver's intentions to brake heavily by either monitoring the brake pedal movement or in the form of a pressure switch tapped in to the hydraulic brake circuit. With the pressure switch method the switch is set to open at some predetermined brake-line pressure (typically about 35 bar); this causes the input voltage to the electronic-control unit to rise. Once 5 volts is reached (usual setting) the electronic-control unit switches the suspension to 'firm' ride mode. When the braking pressure drops below 35 bar the pressure switch closes again; this grounds the input to the electronic-control unit and causes its output voltage to the solenoid control valves also to collapse, and at this point the suspension reverts to 'soft' ride mode.

Acceleration pedal sensor These sensors can be of the simple rotary potentiometer attached to the throttle linkage indicating the throttle opening position. A large downward movement or a sudden release of the accelerator pedal signals to the electronic-control unit that the driver intends to rapidly accelerate or decelerate, respectively. When accelerating hard the rapid change in the

potentiometer resistance and hence input voltage signals the electronic-control unit to switch the suspension to firm ride mode.

Load sensor Load sensors are positioned on top of the strut actuator cylinder; its purpose is to monitor the body load acting down on each strut actuator.

Vehicle speed sensor Vehicle speed can be monitored by the speedometer or at the transmission end by an inductive pick-up or Hall effect detector which produces a series of pulses whose frequency is proportional to vehicle speed. Once the vehicle speed exceeds some predetermined value the electronic-control unit automatically switches the suspension to 'firm' ride mode. As vehicle speed decreases, a point will be reached when the input to the electronic-control unit switches the suspension back to 'soft' ride mode.

Mode selector This dashboard mounted control switches the suspension system via the electronic-control unit to either a comfort (soft) ride mode for normal driving conditions or to a sports (firm) ride mode. However, if the vehicle experiences severe driving conditions while in the comfort ride mode, the electronic-control unit overrides the mode selector and automatically switches the suspension to sports (firm) ride mode.

441

10.17.2 *Active hydro/coil spring suspension system* (Fig. 10.124(a–h))

A typical fully active hydraulic self-levelling suspension system utilizing strut actuators consisting of a cylinder, piston and ram-shaft installed between each spring and its body support. A hydraulic pump driven from the engine supplies high pressure fluid to the accumulator and to the individual strut actuators via a pressure regulating valve and a levelling valve. The purpose of the accumulator is to store fluid at maximum pressure so that it can instantly be discharged to the various strut actuator cylinders when commanded; this would not be possible without an excessive time delay since the pump could not generate and discharge sufficient quantity of high pressure fluid in the time-span required to maintain the car on a level keel. When the engine is running, high pressure fluid is supplied to each strut actuator to bring the body level up to its design height via the levelling control valve. Note some systems may have more than one body height setting.

Unequal weight distribution levelling control (Fig 10.124(a and b)) Uneven weight distribution is automatically compensated for by the levelling control

(a) Heavy load without height level control

(b) Heavy load with height level control

(c) Cornering without anti-roll control

(d) Cornering with anti-roll control
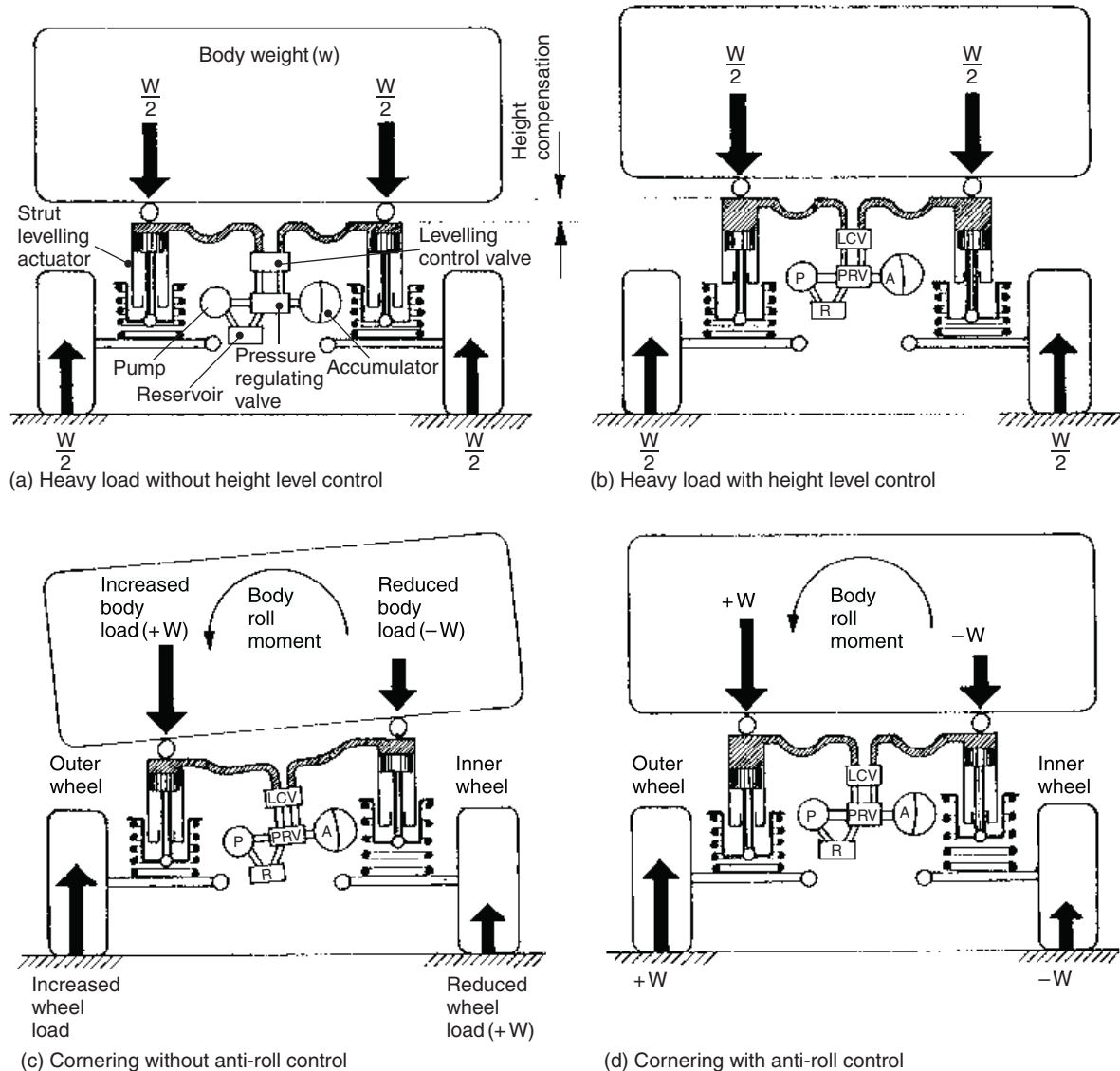
Fig. 10.124 (a–g) Active self–levelling hydraulic/coil spring suspension

(e) Heavy braking without anti-dive control

(f) Heavy braking with anti-dive control

(g) Heavy acceleration without anti-squat control

(h) Heavy acceleration with anti-squat control
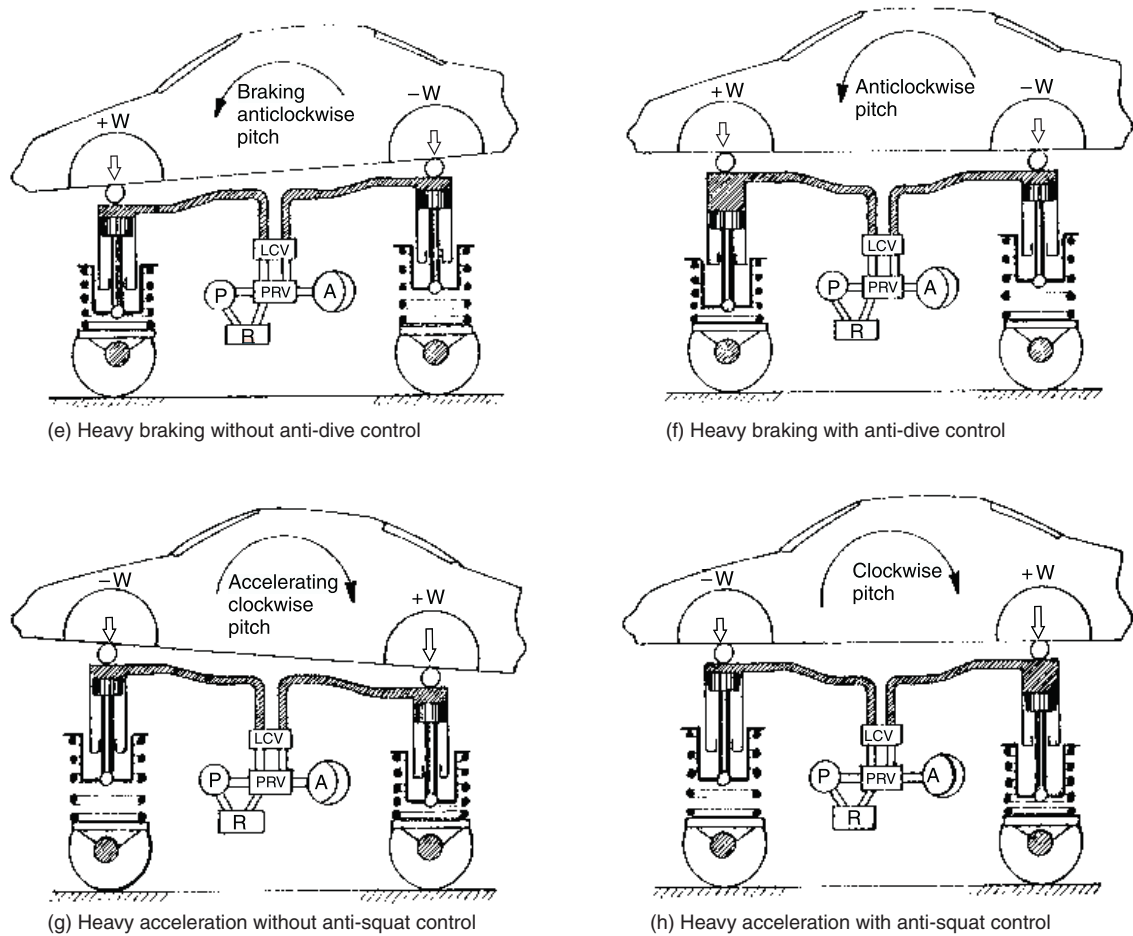
Fig. 10.124 *contd*

valve which allows fluid to enter or exit from the individual strut actuators. Figure 10.124(a) shows the body height if there was no level control or if the self-levelling height is set to a low level, whereas Fig. 10.124(b) shows the suspension level raised against the compressed road springs to compensate for the heavy load. Longitudinal levelling also takes place between the front and rear suspension strut actuators when, for instance, all the passenger seats are occupied and the boot is full of luggage.

Body roll control (Fig. 10.124(c and d))   When the car is negotiating a corner the body tends to tilt so that the inner and outer wheel loads are reduced and increased, respectively. This lateral load transfer shown in Fig. 10.124(c) compresses the outer springs and expands the inner springs thus causing the body to roll and to become uncomfortable for the driver and passengers. To compensate for the

weight transfer, fluid is pumped or released into the outer strut actuators via the levelling control valve until it has lifted the body on the outside to the same height as the inside (see Fig. 10.124(d)). Usually a small angular roll is deliberately allowed to provide the driver with a sense of caution.

Anti-dive control (Fig. 10.124(e and f))   If the car is braked rapidly there is a tendency for the body to pitch forwards, that is, the front of the body temperately dives downwards and the rear lifts (see Fig. 10.124(e)); the dive experienced is due to the longitudinal weight transfer since the body mass wants to continue moving forwards but the road wheels and the unsprung suspension mass are being retarded by the action of the braking force. To overcome this inherent deficiency in the suspension design which occurs usually when soft springs are used, fluid is rapidly transferred into both of the

front strut actuator cylinders (see Fig. 10.124(f)), thereby correcting the front to rear tilt of the body over the braking period and then releasing the excess fluid from the front actuators when normal driving resumes.

Anti-squat control (Fig. 10.124(g and h))   If a car is accelerated rapidly, particularly when pulling away from a standstill, there is a proneness for the body due to its inertia to hold back whereas the propelled wheels and unsprung part of suspension tend to move ahead of the interlinked body. This results in the body tilting backwards so that it squats heavily on the rear axles and wheels (see Fig. 10.124(g)). To correct this ungainly stance when the car is being accelerated, fluid is quickly displaced from the accumulator and pump through the open levelling control valve into the rear strut actuator cylinders; this levels the body longitudinally (see Fig. 10.124(h)). Once the acceleration sensor detects a reduction in acceleration, the electronic-control unit signals the levelling control valve to return the excess fluid trapped in the rear actuator cylinders back to the reservoir so that under steady driving conditions the body remains parallel to the road.

10.17.3 Semi-active controlled hydro/gas suspension (Fig. 10.125(a and b))

Most of the self-levelling suspension layouts are only designed to achieve semi-active suspension control as there is a cost factor and it might not be justified to produce a near perfect suspension since there is always some inherent deficiencies due to other factors built into the body/suspension/transmission design. One version of a semi-active suspension is shown in Fig. 10.125(a, b) here, instead of using steel coil or leaf springs a hydro/gas spring is employed. These spring units basically consist of two hemispherical chambers separated by a flexible diaphragm; the outer sealed chamber is filled with pressurized nitrogen gas which acts as a spring media whereas the underside chamber is connected through the fluid to the height adjusting strut actuators and the supply pump and accumulator. There are three springs for both front and rear pair of suspensions. The system is designed to give two spring rates, these are 'sports' (stiff springing) and 'comfort' (light springing) and there are two damping modes, firm and soft.

Wheel deflection is absorbed through the height actuator cylinder and piston ram so that as a wheel goes over a bump or pot hole, the suspension swing-arm will tilt correspondingly up or down. An upward movement of a swing-arm and ram will displace fluid into the chamber underneath the diaphragm thereby causing it to compress the nitrogen gas. If however the wheel goes into a pot hole the downward movement of the swing-arm and ram causes the effective cylinder space to increase, this reduces the fluid pressure and permits the hydrogen gas to expand by pushing down the diaphragm, and fluid will therefore be displaced back into the height actuator cylinder.

Height and levelling control (Fig. 10.125(a and b)) With the engine running fluid is pumped into the accumulator and into all four height actuators and

(a) Soft ride interlinked three spring suspension mode

(b) Firm ride independent two spring suspension mode
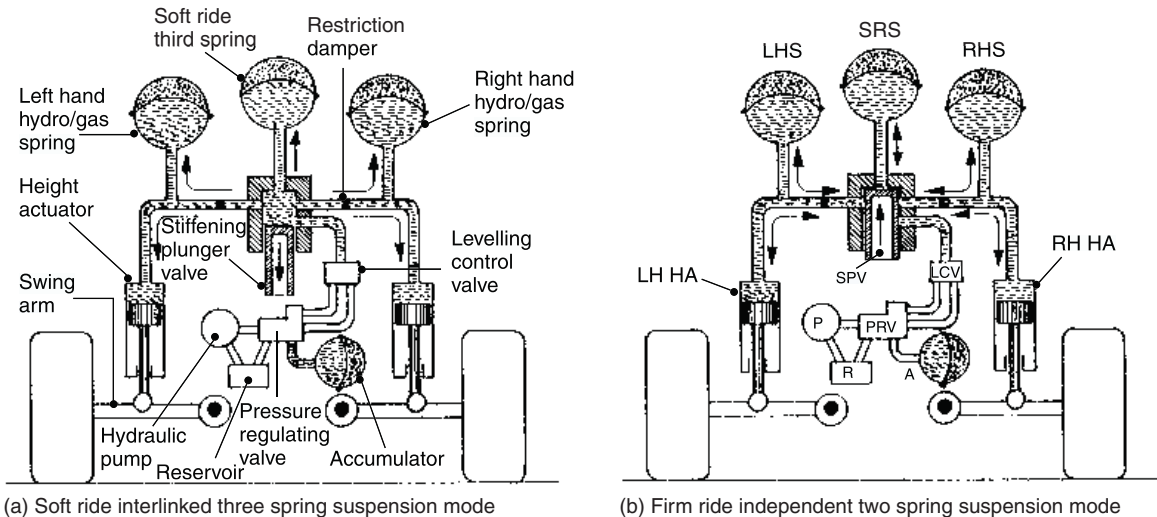
Fig. 10.125 (a and b)   Semi-active hydro/gas suspension

into each lower spring unit chamber until the pre-determined body height setting is reached, at which point the levelling control valve blocks any further supply of fluid to the system. If the load changes due to more or less passengers and luggage, the levelling control valve will automatically permit fluid to enter or leave the spring/actuator system to maintain the optimum programmed body height setting.

Comfort (light springing) ride (Fig. 10.125(a))   For a comfortable ride under normal driving road conditions the spring loaded plunger is pulled outwards (downward). Both height actuators are interconnected with the fluid supply and the soft-ride third spring; the latter increases the volume of compressible gas by 50%, thus reducing the spring stiffness, thereby providing greater ride comfort. When the body rolls during cornering fluid is transferred from the outer to the inner height actuator cylinders but this is slowed down to some extent by the restrictor dampers.

Sports (stiff springing) ride (Fig. 10.125(b))   If the car is to be driven fast or is moving sharply around a bend the suspension system can be switched to sports (firm) ride by way of pushing the stiffening plunger inwards (upwards), see Fig. 10.125(b). This has the effect of blocking the third spring fluid movement from both adjacent right and left hand wheel springs. All three springs are now isolated from each other so that the vertical deflection of each wheel strut actuator is confined entirely to its own gas filled spring. Accordingly the spring rate for vertical piston ram movement is stiffened for each wheel suspension, and this also provides a degree of body roll control.

10.17.4 Semi-active hydro/gas electronic controlled suspension system (Fig. 10.126)
The system shown in Fig. 10.126 illustrates a front double wishbone suspension and a rear semi-trailing swing-arm suspension, each wheel having its own levelling strut actuator and hydro/gas spring unit. A third soft ride hydro/gas unit is shared between the front pair of suspensions and similarly a third unit is installed for the rear suspension. An electronic-control unit microprocessor is incorporated in the system which takes in signals from the various sensors; this information is then processed and converted to electrical instructions to the various solenoid control valves. Note that some of the sensors shown in Fig. 10.126 may not always be included in a semi-active suspension system,

the actual sensors chosen will depend upon the degree of control sophistication to be built into the suspension design. The sensors shown are listed as follows: height sensors, load sensors, steering sensor, longitudinal and lateral acceleration sensors, acceleration pedal sensor, brake pressure and pedal sensors and vehicle speed sensor. For instance a simple semi-active suspension can operate effectively with just four sensors such as height sensors, steering sensor, brake pressure sensor and vehicle speed sensor. The electronic-control unit output directs the energizing and de-energizing of the front and rear solenoid controlled levelling valves and the front and rear solenoid stiffening valves. There is a mode selector which enables the driver to put the suspension in either sports (firm) ride or comfort (soft) ride; however, when switched to comfort ride, if driving conditions become harsh the suspension automatically reverts to sports (stiff springing) ride, but it will eventually change back to comfort ride when normal driving conditions prevail.

Comfort (soft) ride mode (Fig. 10.126)   When switched to comfort ride mode the electronic-control unit will supply current to energize both the front and rear solenoid controlled delivery valves incorporated in both of the levelling control valve units and also to supply current to energize the front and rear stiffening solenoid valve units (note Fig. 10.126 shows the front levelling control valve and the front stiffening valves open to fluid delivery). Fluid is now permitted to flow to each wheel strut actuator cylinder and its corresponding hydro/gas spring unit and in addition to the both front and rear soft ride third spring units until the preset body height level is reached. The appropriate sensors now signal the electronic-control unit to switch off the power supply to all the solenoid valves; the correct quantity of fluid is therefore contained in the four actuator cylinders for the conditions prevailing at the time. Should the body load be reduced such as when a passenger gets out, the electronic-control unit signals both the solenoid controlled return valves installed in the front and rear levelling control valve units to open (note Fig. 10.126 shows the rear levelling control return valve open for the exit of fluid), fluid will now return to the reservoir until the body height sensors signal that the correct body height has settled again to the manufacturer's setting, and at this point the solenoid return valves close. This cycle of events for delivery and releasing fluid to the spring and levelling system is continuous. When driving conditions change
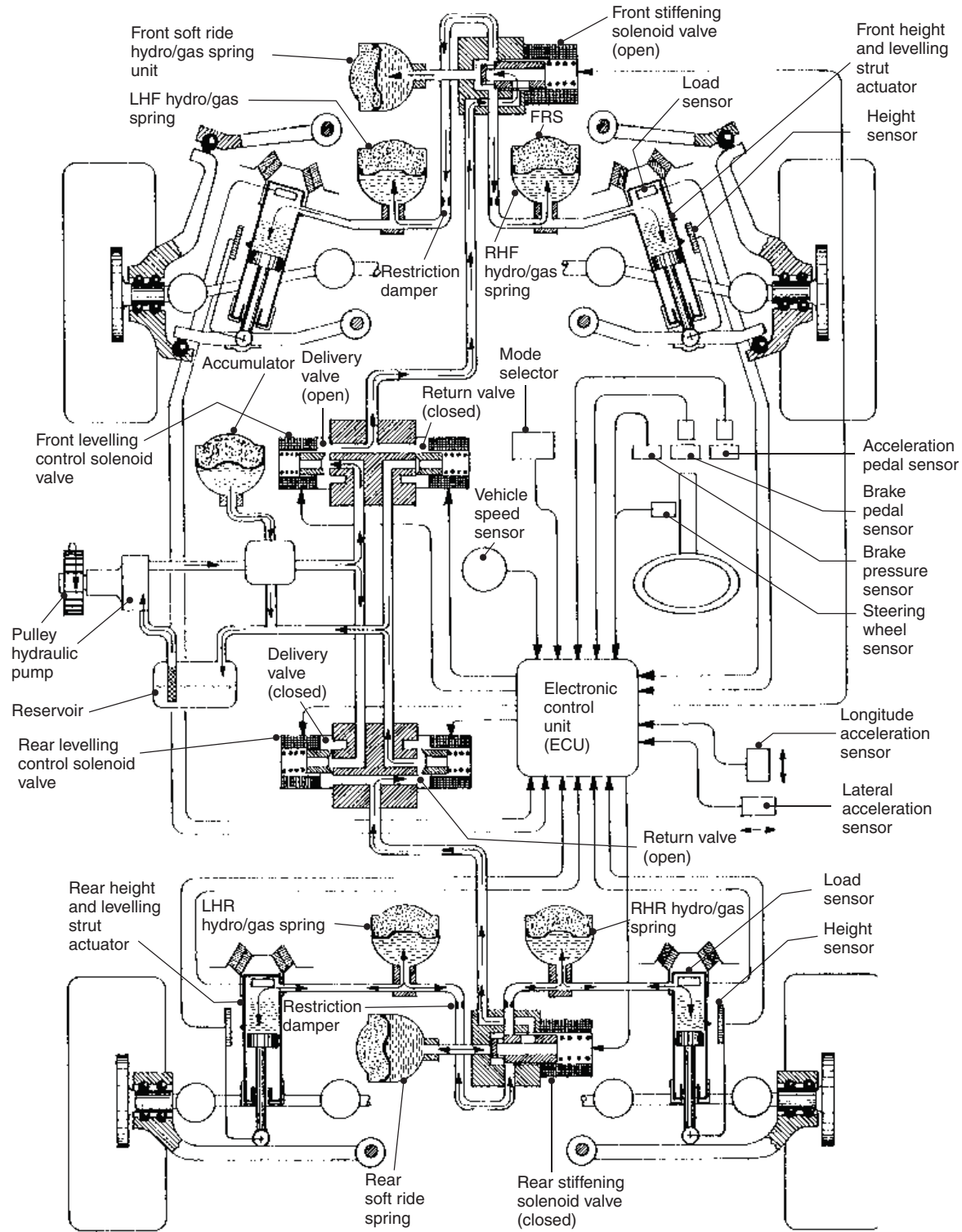
Fig. 10.126 Semi-active hydro/gas electronic controlled suspension

446

such as during rapid acceleration, hard braking, fast cornering or high speed driving, the front and rear solenoid controlled stiffening valves are signalled to de-energize; this permits the stiffening plunger valves to close, thereby isolating the front and rear interconnected three spring units from each other, thus automatically changing the suspension mode to sports (firm) ride (note Fig. 10.126 shows the rear stiffening solenoid valve closed as for sports (stiff springing) ride mode).

Sports (firm) ride mode (Fig. 10.126) If the driver switches the suspension to sports (stiff springing) ride mode the electronic-control unit will instruct the front and rear stiffening valve unit to de-energize and push in the plunger valve. As a result each wheel spring and strut actuator becomes isolated from the spring opposite it and also from the third stiffening spring unit. As a result the reduction in diaphragm area exposed to the nitrogen gas (now 50% less) increases the stiffness rate of each spring/actuator unit; it therefore provides a firm ride for good steering response, and the prevention of fluid moving between the right and left hand strut actuators increases the suspension's resistance to body roll.

10.18 lectronic controlled pneumatic (air) suspension for on and off road use

A pneumatic (air) controlled suspension system provides a variable spring rate so that a constant suspension frequency is obtained between light and heavy load conditions. Additional telescopic dampers are also installed to improve the ride quality and comfort. A driver's height control is also provided to enable the body height to be adjusted for specific purposes such as loading the boot or cargo space, towing a trailer, driving over rough terrain, or muddy ground, or travelling though flooded areas.

10.18.1 Description of system (Fig. 10.127)

The basic air suspension system consists of four rolling lobe air springs mounted between the chassis and each of the suspension lower swing arms. These springs work within an air pressure ranging between 6 and 10 bar, thus for low height settings and light loads the pressure needed may be down to 6 bar but for higher height settings and heavy loads the pressure could rise to just under 10 bar. Compressed air is supplied via a single cylinder compressor driven by an electric motor and a reservoir tank is provided to store compressed air for instant use. A unloader valve is provided to safe-

guard the compressor and system from overload. There are four ride height solenoid valves, one for each air spring unit and an inlet and outlet solenoid valve which controls the air supply to and from the system. In addition there are inlet and outlet exhaust solenoid valves which control the release of excess air to the atmosphere. An air silica gel drier dries the wet freshly compressed air before it passes though the various valves and enters the reservoir tank and air spring units.

There are various sensors and switches which provide essential information for the ECU to process and to make corrective decisions to the levelling of the vehicle's body, these are as follows:

Compressor pressure switch A compressor pressure switch monitors the reservoir tank air pressure and provides a signal to the electronic-control unit (ECU) to switch the compressor's electric motor on when the pressure drops below 7.5 bar and to switch it off as the pressure reaches 10 bar.

Height sensors The air spring unit height is monitored by individual height sensors which provide voltage signals to the ECU, the ECU then computes this informs to determine when the pre-determined spring unit height has been reached, and when to switch 'on' or 'off' the ride height solenoid valves. Each wheel suspension is constantly moving up and down so that the different heights measured over a period of 12 sec are recorded to enable the ECU to calculate if more or less air is needed to fill any of the rolling lobe springs.

ngine speed sensor The engine speed sensor takes its readings from the alternator; it is used to indicate if the engine is running as the ECU will switch 'off' the compressor's electric motor if the engine is not running or if the engine speed drops below 500 rev/min to prevent the electric motor draining the battery.

Vehicle speed sensor The vehicle speed sensor takes its readings from the speedometer, to monitor road speed. When the vehicle's speed exceeds 80 km/h the ECU lowers the ride height by about 20 mm to reduce aerodynamic drag and to improve the road holding stability.

Hand brake switch The hand brake switch is used to inform the ECU that the vehicle has stopped and is stationary and only then can the lower access height setting mode be actuated for safety reasons if required.
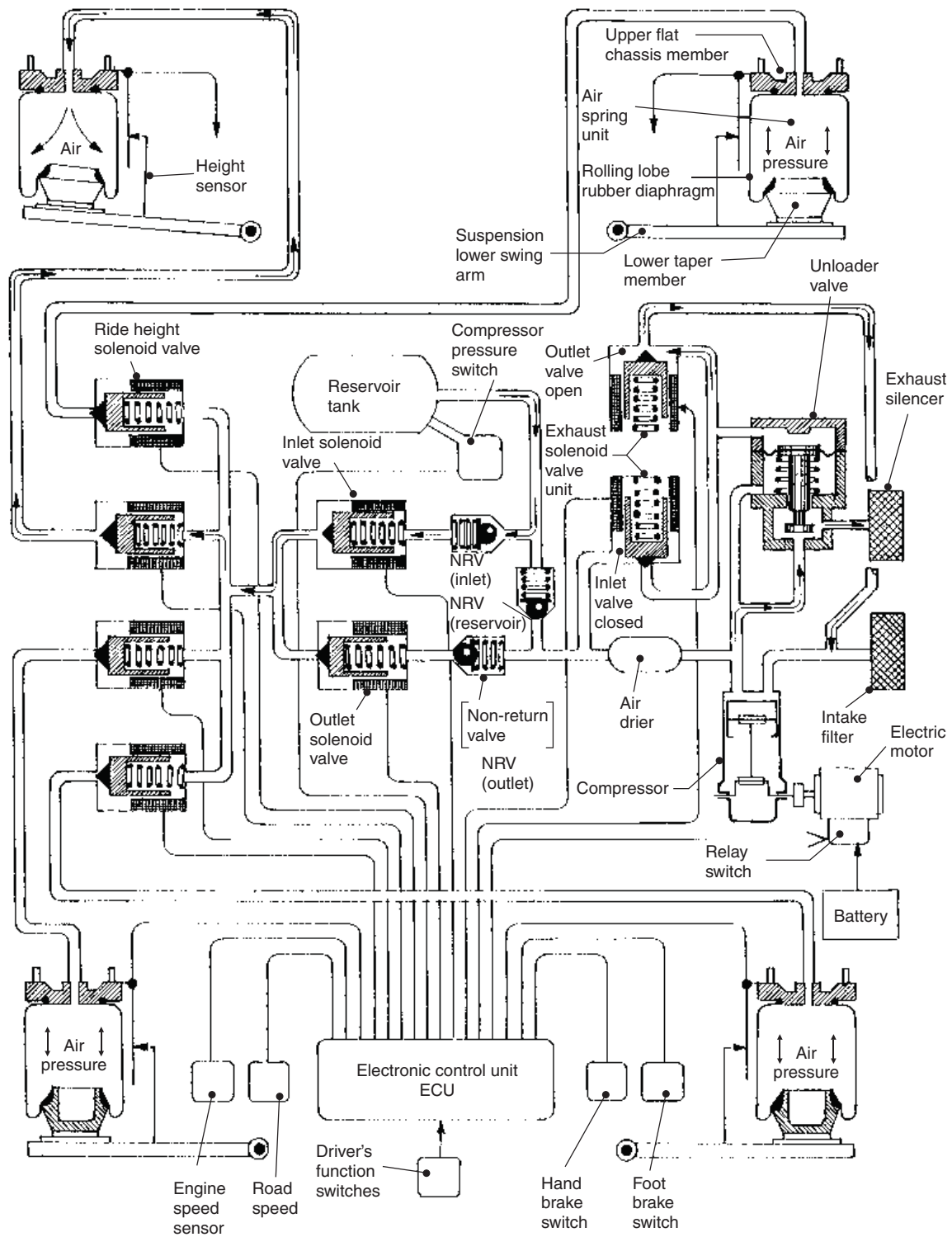
Fig. 10.127 Electronic controlled pneumatic (air) suspension

448

river s function switches The driver's function switches are dash board mounted and are used to select the different height programmes in the ECU when in manual mode. These are standard ride height used for trailer towing, high profile mode for off-road driving and a low profile mode for load accessing.

10.18.2 Operating conditions (Fig. 10.127)

Compressor charging   When the ignition switch is on and the engine is running the ECU signals the electric-motor relay switch to close and to energize the electric motor; the compressor will then rotate and begin to charge the reservoir tank via the reservoir non-return valve (NRV). Under these conditions both the inlet and the outlet solenoid valves are de-energized, thereby closing off the ride height solenoid valves and air spring units from the rest of the system. The exhaust outlet solenoid valve is de-energized and is closed, and the inlet solenoid valve is energized and therefore opens, thus permitting air pressure to force down the unloader valve diaphragm against the spring tension, hence closing the unloader valve and thus preventing air discharging into the atmosphere (opposite to that shown in Fig. 10.127). As the air pressure in the reservoir approaches 10 bar the compressor pressure switch signals the ECU to switch 'off' the compressor's electric motor.

Raising and lowering height levels   If the rolling lobe air springs are in the low profile access mode or if there has been a loss or expulsion of air previously from the springs, then each height sensor will signal to the ECU to energize and open the inlet solenoid valve and each of the ride height solenoid valves, hence permitting the reservoir tank to discharge additional air into the four air spring bags. Once the air spring units have risen to

their setting mode the ECU de-energizes and closes the inlet and ride height solenoid valves thereby retaining the air mass within the spring units. Should the air spring height be too high for a particular ride height setting due to a reduction in load or that a different height profile mode has been selected, then the ECU will energize and open the four ride height valves, the outlet solenoid valve, the inlet and outlet exhaust solenoid valves. Air will now be released from the air spring bags, where it then escapes to the compressors intake and intake filters and via the open unloader valve into the atmosphere; this will continue until the air springs are restored to their pre-programmed height at which point the ECU will cut 'off' the current supply to close all of these valves.

ndividual spring levelling   If one corner of the vehicle dips more than the others (see left hand top corner Fig. 10.127), possibly due to uneven passenger or load distribution, the relevant height sensors detect this in the from of a voltage change. This is therefore passed on to the ECU and accordingly energizes and opens the inlet solenoid valve and the appropriate ride height inlet solenoid valve; once again pressurized air will be delivered from the reservoir tank via the inlet NRV to the particular spring to compensate for the body sag until the body is level again.

Unloader valve action   When the compressor is charging the system the exhaust inlet and outlet valves are open and closed respectively, if the discharge pressure from the compressor should exceed its maximum safe limit of around 10 bar, then air pressure acting with the spring underneath the diaphragm lifts it up against the resisting downward acting air pressure thus enabling the valve to open and to unload the system.

11 Brake system

11.1 Braking fundamentals

11.1.1 The energy of motion and work done in braking (Fig. 11.1)

A moving vehicle possesses kinetic energy whose value depends on the weight and speed of the vehicle. The engine provides this energy in order to accelerate the vehicle from a standstill to given speed, but this energy must be partially or totally dissipated when the vehicle is slowed down or brought to a standstill. Therefore it is the function of the brake to convert the kinetic energy possessed by the vehicle at any one time into heat energy by means of friction (Fig. 11.1).

The equation for kinetic energy, that is the energy of motion, may be given by

$$E_k = \tfrac{1}{2} m v^2$$

where
E_k = kinetic energy of vehicle (J)
m = mass of vehicle (kg)
v = speed of vehicle (m/s)

The work done in bringing the vehicle to rest is given by

$$E_w = Fs$$

where
E_w = work done (J)
F = average braking force (N)
s = distance travelled (m)

When braking a moving vehicle to a standstill, the work done by the brake drums must equal the initial kinetic energy possessed by the vehicle so that

$$E_w = E_k,$$
$$Fs = \tfrac{1}{2} m v^2$$

Average brake force $F = \dfrac{m v^2}{2s} \,(\text{N})$

Fig. 11.1 Illustration of braking conditions

Example (Fig. 11.1)   A car of mass 800 kg is travelling at 36 km/h. Determine the following:

a) the kinetic energy it possesses,
b) the average braking force to bring it to rest in 20 metres.

a) $v = \dfrac{36 \times 1000}{60 \times 60} = 10\,\text{m/s}$

   kinetic energy $= \tfrac{1}{2} m v^2 = \tfrac{1}{2} \times 800 \times 10^2$
   $= 40\,\text{kJ}$

b) Work done to stop car = change in vehicle's kinetic energy

$$Fs = \tfrac{1}{2} m v^2$$
$$20F = 40\,000$$
$$F = \frac{40\,000}{20} = 2000\,\text{N}$$
$$= 2\,\text{kN}$$

11.1.2 Brake stopping distance and efficiency

Braking implies producing a force which opposes the motion of the vehicle's wheels, thereby reducing the vehicle speed or bringing it to a halt. The force or resistance applied to stop a vehicle or reduce its speed is known as the braking force.

The braking efficiency of a vehicle is defined as the braking force produced as a percentage of the total weight of the vehicle, that is

$$\text{Braking efficiency} = \frac{\text{Braking force}}{\text{Weight of vehicle}} \times 100$$

When the braking force is equal to the whole weight of the vehicle being braked, the braking efficiency is denoted as 100%. The braking efficiency is generally less than 100% because of insufficient road adhesion, the vehicle is on a down gradient or the brake system is ineffective.

The brake efficiency is similar to the coefficient of friction which is the ratio of the frictional force to the normal load between the rubbing surfaces.

i e   Coefficient of friction $= \dfrac{\text{Friction force}}{\text{Normal load}}$

that is $\mu = \dfrac{F}{N}$

i e $\mu = \dfrac{F}{N} =$

where   coefficient of friction
                 brake efficiency
F friction force
N normal load.

Thus a braking efficiency of 100% is equal to a coefficient of friction of one.

i e $(100\%) = \dfrac{F}{N} = \quad = 1$

11.1.3 Determination of brake stopping distance
(Fig. 11.1)
A rough estimate of the performance of a vehicle's brakes can be made by applying one of the equations of motion assuming the brakes are 100%.

i e $^2 = \quad ^2 + 2gs$
where   initial braking speed (m/s)
              final speed (m/s)
g deceleration due to gravity (9.81 m/s^2)
s stopping distance (m)

If the final speed of the vehicle is zero (i.e. V = 0)

then $0 = \quad + 2gs$

and $s = \dfrac{^2}{2g} = \dfrac{^2}{2 \times 9\,81} \hat{=} \dfrac{^2}{20}$

To convert km/h to m/s;

$$U(m/s) = \frac{1000}{60 \times 60} \quad = 0\,28 \quad (km/h)$$

$$s = \frac{(0\,28\quad)^2}{2g} = \frac{(0\,28)^2 \quad ^2}{2 \times 9\,81} = 0\,004 \quad ^2(m)$$

xample Calculate the minimum stopping distance for a vehicle travelling at 60 km/h.

Stopping distance $s = 0\,004 \quad ^2(m)$
$$= 0\,004 \times 60^2 = 14\,4\,m$$

11.1.4 Determination of brake efficiency
(Fig. 11.1)
The brake efficiency can be derived from the kinetic energy equation and the work done in bringing the vehicle to a standstill.

Let F = braking force (N),
        = coefficient of friction,
        = vehicle weight (N),
        = initial braking speed (m/s),
m = vehicle mass (kg),
s = stopping distance (m),
        = brake efficiency.

Then equating work and kinetic energy,

$$Fs = \quad m \quad ^2$$

but $m = \dfrac{}{g}$

$$Fs = \frac{^2}{2g}$$

$$s = \frac{^2}{2Fg}$$

but $\dfrac{F}{} = \quad =$

Thus $s = \dfrac{^2}{2g}$

$$= \frac{^2}{2gs} \times 100$$

but   (m/s) $= 0\,28$ (km/h)

Hence $= \dfrac{(0\,28\quad)^2}{2 \times 9\,81 s} \times 100 = 0\,4\dfrac{^2}{s}\%$

xample Determine the braking efficiency of a vehicle if the brakes bring the vehicle to rest from 60 km/h in a distance of 20 metres.

$$= \frac{0\,4 \quad ^2}{s} = \frac{0\,4 \times 60^2}{20}$$
$$= 72\%$$

A table of vehicle stopping distances for various vehicle speeds and brake efficiencies is shown in Table 11.1.

11.1.5 Adhesion factor
The stopping distance of a wheel is greatly influenced by the interaction of the rotating tyre tread

ab e 11.1 Stopping distances for various vehicle speeds and brake efficiencies

Vehicle speed	Stopping distance for various braking efficiencies (m)					
km/h	00%	90%	80%	70%	0%	50%
0	0.4	0.4	0.5	0.	0.7	0.8
20	.	.8	2.0	2.3	2.7	3.2
30	3.	4.0	4.5	5.	.0	7.2
40	.4	7.	8.0	9.	0.7	2.8
50	0.0	.	2.5	4.3	.7	20.0
0	4.4	.0	8.0	80.	24.0	28.8
70	9.	2 .8	24.5	28.0	32.7	39.2
80	25.	28.4	32.0	3 .	42.7	5 .2
90	32.4	3 .0	40.5	4 .3	54.0	4.8
00	40.0	44.4	50.0	57.	.7	80.0

ab e 11.2 Adhesion factors for various road surfaces

No.	Material	Condition	Adhesion factor
	Concrete, coarse asphalt	dry	0.8
2	Tarmac, gritted bitumen	dry	0.
3	Concrete, coarse asphalt	wet	0.5
4	Tarmac	wet	0.4
5	ritted bitumen tarmac	wet	0.3
	ritted bitumen tarmac	greasy	0.25
7	ritted bitumen, snow compressed	greasy	0.2
8	ritted bitumen, snow compressed	dry	0. 5
9	Ice	wet	0.

and the road surface. The relationship between the decelerating force and the vertical load on a wheel is known as the *adhesion factor* ($_a$). This is very similar to the coefficient of friction ( ) which occurs when one surface slides over the other, but in the case of a correctly braked wheel, it should always rotate right up to the point of stopping to obtain the greatest retarding resistance.

Typical adhesion factors for various road surfaces are given in Table 11.2.

11.2 Brake shoe and pad fundamentals

11.2.1 Brake shoe self-energi ation (Fig. 11.2)
The drum type brake has two internal semicircular shoes lined with friction material which matches up to the internal rubbing face of the drum. The shoes are mounted on a back plate, sometimes known as

a torque plate, between a pivot anchor or wedge type abutment at the lower shoe ends, and at the upper shoe top end by either a cam or hydraulic piston type expander. For simplicity the expander in Fig. 11.2 is represented by two opposing arrows and the shoe linings by two small segmental blocks in the mid-region of the shoes.

When the drum is rotating clockwise, and the upper tips of the shoes are pushed apart by the expander force F_e, a normal inward reaction force N will be provided by the drum which resists any shoe expansion.

As a result of the drum sliding over the shoe linings, a tangential frictional force $F_t = N$ will be generated between each pair of rubbing surfaces.

The friction force or drag on the right hand shoe (Fig. 11.2) tends to move in the same direction as its shoe tip force F_e producing it and accordingly helps to drag the shoe onto the drum, thereby effectively raising the shoe tip force above that of the original expander force. The increase in shoe tip force above that of the input expander force is described as *positive servo* and shoes which provide this self-energizing or servo action are known as *leading shoes*

i e $\quad F_L = F_e + F_t$

where $\quad F_L$ = leading shoe tip resultant force

Likewise considering the left hand shoe (Fig. 11.2) the frictional force or drag F_t tends to oppose and cancel out some of the shoe tip force F_e producing it. This causes the effective shoe tip force to be less than the expander input force. The resultant reduction in shoe tip force below that of the initial
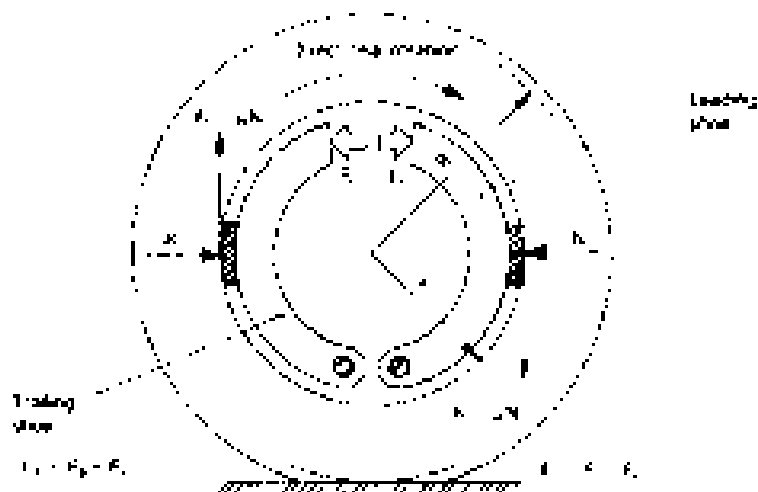
Fig. 11.2 Drum and shoe layout

input tip force is described as *negative servo* and shoe arrangements which have this de-energizing property are known as *trailing shoes*

i e $\qquad F_T = F_e - F_t$

where $\quad F_T$ = trailing shoe tip resultant force

The magnitude of the self-energizing action is greatly influenced by the rubbing surface temperature, dampness, wetness, coefficient of friction and speed of drum rotation.

Changing the direction of rotation of the drum causes the original leading and trailing shoes to reverse their energizing properties, so that the leading and trailing shoes now become trailing and leading shoes respectively.

The shoe arrangement shown in Fig. 11.2 is described as a *leading trailing shoe drum brake*

Slightly more self-energizing is obtained if the shoe lining is heavily loaded at the outer ends as opposed to heavy mid-shoe loading.

11.2.2 Retarding wheel and brake drum torques
(Fig. 11.2)
The maximum retarding wheel torque is limited by wheel slip and is given by

$$T_w = \mu_a W R \text{ (Nm)}$$

where $\qquad T_w$ = wheel retarding torque (Nm)
$\qquad \mu_a$ = adhesion factor
$\qquad W$ = vertical load on wheel (N)
$\qquad R$ = wheel rolling radius (m)

Likewise the torque produced at this brake drum caused by the frictional force between the lining and drum necessary to bring the wheel to a standstill is given by

$$T_B = \mu N r \text{ (Nm)}$$

where $\qquad T_B$ = brake drum torque (Nm)
$\qquad \mu$ = coefficient of friction between lining and drum
$\qquad N$ = radial force between lining and drum (N)
$\qquad r$ = drum radius (m)

Both wheel and drum torques must be equal up to the point of wheel slip but they act in the opposite direction to each other. Therefore they may be equated.

i e $\qquad T_B = T_w$

$\qquad \mu N r = \mu_a W R$

Force between lining and drum $N = \dfrac{\mu_a W R}{\mu r}$ (N)

xample A road wheel has a rolling radius of 0.2 m and supports a load of 5000 N and has an adhesion factor of 0.8 on a particular road surface. If the drum radius is 0.1 m and the coefficient of friction between the lining and drum is 0.4, determine the radial force between the lining and drum.

$$N = \frac{\mu_a W R}{\mu r}$$

$$= \frac{0\,8 \times 5000 \times 0\,2}{0\,4 \times 0\,1}$$

$$= 20\,000 \text{ N or } 20 \text{ kN}$$

11.2.3 Shoe and brake factors (Fig. 11.2)
If the brake is designed so that a low operating force generates a high braking effort, it is said to have a high self-energizing or servo action. This desirable property is obtained at the expense of stability because any frictional changes disproportionately affect torque output. A brake with little self-energization, while requiring a higher operating force in relation to brake effort, is more stable in operation and is less affected by frictional changes.

The multiplication of effort or self-energizing property for each shoe is known as the *shoe factor*

The shoe factor S is defined as the ratio of the tangential drum drag at the shoe periphery F_t to the force applied by the expander at the shoe tip F_e.

i e $\quad$ Shoe factor $= \dfrac{\text{Tangential drum force}}{\text{Shoe tip force}}$

$$S = \frac{F_t}{F_e}$$

The combination of different shoe arrangements such as leading and trailing shoes, two leading shoes, two trailing shoes etc. produces a brake factor B which is the sum of the individual shoe factors.

$\qquad$ Brake factor = Sum of shoe factors

i e $\qquad B = (S_L + S_T), 2S_L, 2S_T$ and

$\qquad (S_p + S_s)$

11.2.4 Drum shoe arrangements (Fig. 11.3(a–c))

Leading and trailing shoe brakes (Fig. 11.3(a)) If a single cylinder twin piston expander (double acting) is mounted between two shoe tips and the opposite shoe tips react against a fixed abutment, then the leading shoe is forced against the drum in the forward rotation direction, whilst the trailing shoe works against the rotation direction producing
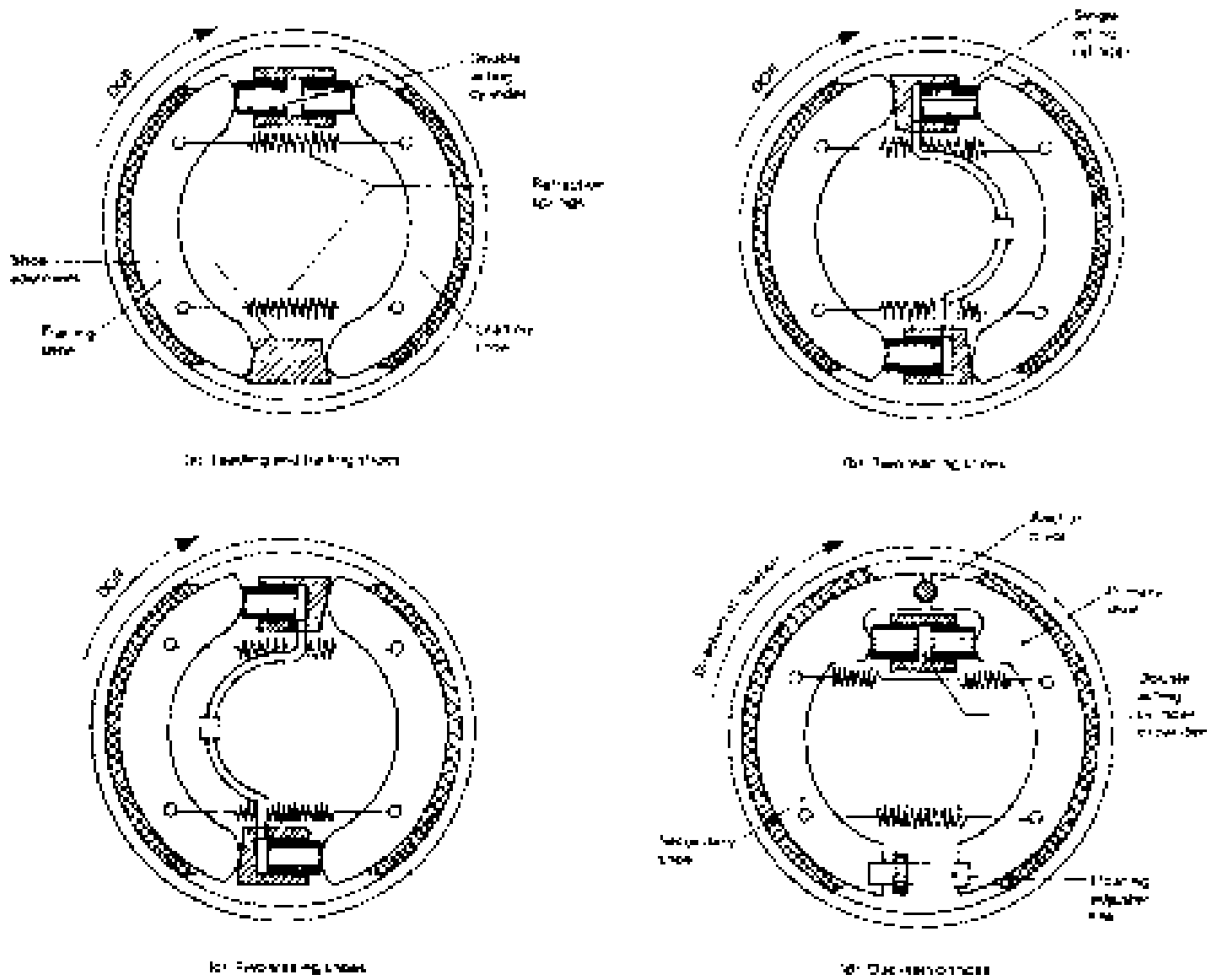
Fig. 11.3 (a–d) Various brake shoe arrangements

much less frictional drag. Such an arrangement provides a braking effect which is equal in both forward and reverse motion. Rear wheel brakes incorporating some sort of hand brake mechanism are generally of the lead and trailing shoe type.

wo leading shoe brakes (Fig. 11.3(b)) By arranging a pair of single piston cylinders (single acting) diametrically opposite each other with their pistons pointing in the direction of drum rotation, then when hydraulic pressure is applied, the drum to lining frictional drag force pulls the shoes in the same direction as the shoe tip piston forces, thus causing both shoes to become self-energizing. Such a layout is known as a *two leading shoe drum type brake* In reverse, the braking force is reduced due to the drag force opposing the piston tip forces; both shoes in effect then have a trailing action. Two leading shoe brakes are possibly still the most popular light commercial type front wheel brake.

wo trailing shoe brakes (Fig. 11.3(c)) If now two separate single acting cylinders are mounted between the upper and lower shoe tips so that both pistons counteract the rotational forward direction of the drum, then the resultant lining drag force will be far less for each shoe, that is, there is a negative servo condition.

Brakes with this layout are therefore referred to as *two trailing shoe brakes* This arrangement is suitable for application where lining stability is important and a servo assisted booster is able to compensate for the low resultant drag force relative to a given input shoe tip force. A disadvantage of a two trailing shoe brake is for the same brake effect as a two leading shoe brake; much higher hydraulic line pressures have to be applied.

uo-servo shoe brakes (Fig. 11.3(d)) A double acting cylinder expander is bolted to the back plate and the pistons transmit thrust to each adjacent shoe, whereas the opposite shoe tip ends are joined together by a floating adjustment link. On application of the brake pedal with the vehicle being driven forward, the pistons move both shoes into contact with the revolving drum. The shoe subjected to the piston thrust which acts in the same direction as the drum rotation is called the *primary shoe* and this shoe, when pulled around with the drum, transfers a considerable force to the adjacent shoe tip via the floating adjustment link. This second shoe is known as the *secondary shoe* and its

initial movement with the drum pushes it hard against the anchor pin, this being permitted by the pistons themselves floating within the cylinder to accommodate any centralization which might become necessary. Under these conditions a compounding of both the primary circumferential drag force and that produced by the secondary shoe itself takes place so that a tremendous wedge or self-wrapping effect takes place far in excess of that produced by the two expander pistons alone. These brakes operate equally in the forward or reverse direction. Duo-servo shoe brakes give exceptionally good performance but are very sensitive to changes in shoe lining properties caused by heat and wetness.

Because the secondary shoe performs more work and therefore wears quicker than the primary shoe, lining life is equalized as far as possible by fitting a thick secondary shoe and a relatively thin primary shoe.

11.2.5 *The principle of the disc brake*
(Fig. 11.4(a, b and c))
The disc brake basically consists of a rotating circular plate disc attached to and rotated by the wheel hub and a bridge member, known as the caliper, which straddles the disc and is mounted on the suspension carrier, stub axle or axle casing (Fig. 11.4(b)). The caliper contains a pair of pistons and friction pads which, when the brakes are applied, clamp the rotating disc, causing it to reduce speed in accordance to the hydraulic pressure behind each piston generated by the pedal effort.

The normal clamping thrust N on each side of the disc (Fig. 11.4(b and c)) acting through the pistons multiplied by the coefficient of friction generated between the disc and pad interfaces produces a frictional force $F = N$ on both sides of the disc. If the resultant frictional force acts through the centre of the friction pad then the mean distance between the centre of pad pressure and the centre of the disc will be

$$\frac{R_2 - R_1}{2} = R$$

Accordingly, the frictional braking torque (Fig. 11.4(a)) will be dependent upon twice the frictional force (both sides) and the distance the pad is located from the disc centre of rotation. That is,

$$\text{Braking torque} = 2 N\left(\frac{R_2 - R_1}{2}\right)(\text{Nm})$$

$$\text{i e} \qquad T_B \qquad = 2 NR \ (\text{Nm})$$
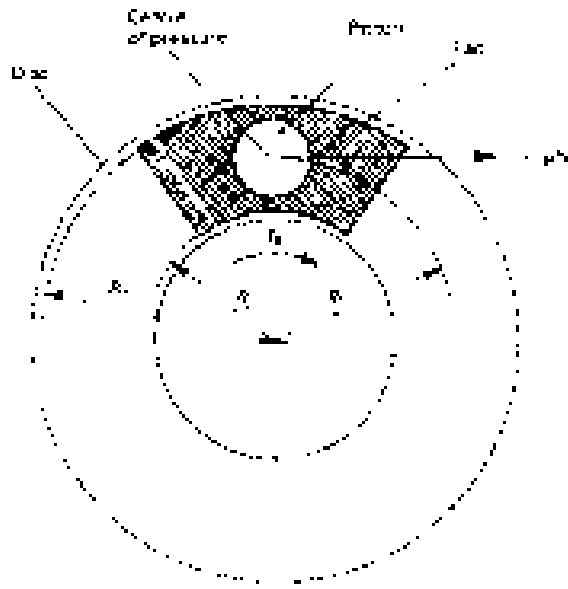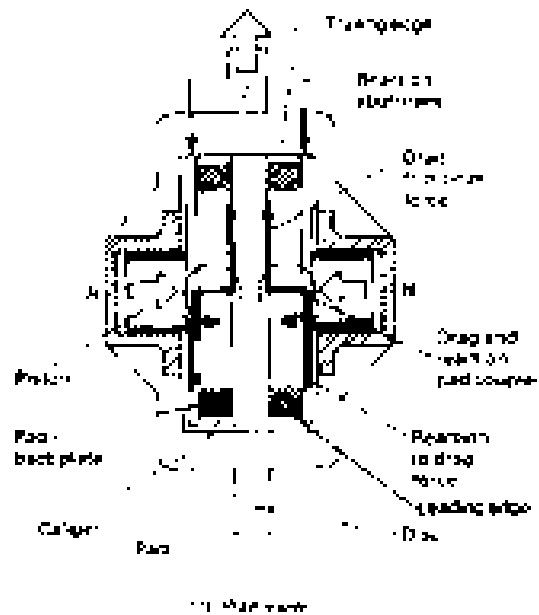
Fig. 11.4 (a–c) Disc and pad layout

Example If the distance between the pad's centre of pressure and the centre of disc rotation is 0.12 m and the coefficient of friction between the rubbing faces is 0.35, determine the clamping force required to produce a braking torque of 84 Nm.

$$T_B = 2 \mu NR$$

$$\text{Clamping force } N = \frac{T_B}{2 \mu R}$$

$$= \frac{84}{2 \times 0.35 \times 0.12}$$

$$= 1000 \, N$$
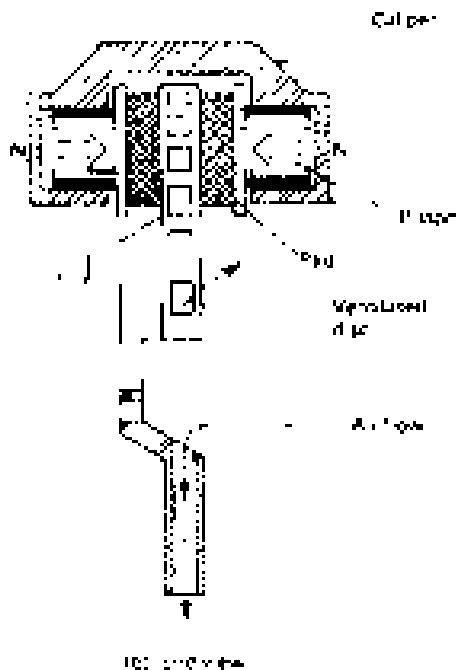
11.2.6 Disc brake pad alignment (Fig. 11.4)

When the pads are initially applied they are loaded against the disc with uniform pressure, but a small tilt tendency between the leading and trailing pad edges caused by frictional pad drag occurs. In addition the rate of wear from the inner to the outer pad edges is not uniform. The bedding-in conditions of the pads will therefore be examined in the two parts as follows:

1 Due to the thickness of the pad there is a small offset between the pad/disc interface and the pad's back plate reaction abutment within the caliper (Fig. 11.4(c)). Consequently, a couple is

456

produced which tends to tilt the pad into contact with the disc at its leading edge harder compared to the trailing edge. This in effect provides a very small self-energizing servo action, with the result that the wear rate at the leading edge is higher than that at the trailing edge.

2 The circular distance covered by the disc in one revolution as it sweeps across the pad face increases proportionately from the inner to the outer pad's edges (Fig. 11.4(a)). Accordingly the rubbing speed, and therefore the work done, increases from the inner to the outer pad edges, with the result that the pad temperature and wear per unit area rises as the radial distance from the disc centre becomes greater.

11.2.7 Disc brake cooling (Fig. 11.4)

The cooling of the brake disc and its pads is achieved mostly by air convection, although some of the heat is conducted away by the wheel hub. The rubbing surface between the rotating disc and the stationary pads is exposed to the vehicle's frontal airstream and directed air circulation in excess of that obtained between the drum and shoe linings. Therefore the disc brake is considerably more stable than the drum brake under continued brake application. The high conformity of the pad and disc and the uniform pressure enable the disc to withstand higher temperatures compared to the drum brake before thermal stress and distortion become pronounced. Because there is far less distortion with discs compared to drums, the disc can operate at higher temperatures. A further feature of the disc is it expands towards the pads, unlike the drum which expands away from the shoe linings. Therefore, when hot, the disc brake reduces its pedal movement whereas the drum brake increases its pedal movement.

Cast ventilated discs considerably improve the cooling capacity of the rotating disc (Fig. 11.4(b)). These cast iron discs are in the form of two annular plates ribbed together by radial vanes which also act as heat sinks. Cooling is effected by centrifugal force pushing air through the radial passages formed by the vanes from the inner entrance to the outer exit. The ventilated disc provides considerably more exposed surface area, producing something like a 70% increase in convection heat dissipation compared to a solid disc of similar weight. Ventilated discs reduce the friction pad temperature to about two-thirds that of a solid disc under normal operating conditions. Pad life is considerably increased with lower operating temperatures, but there is very little effect on the fric-

tional properties of the pad material. Ventilated wheels have very little influence on the disc cooling rate at low speeds. At very high speeds a pressure difference is set up between the inside and outside of the wheel which forces air to flow through the vents towards the disc and pads which can amount to a 10% improvement in the disc's cooling rate. The exposure of the disc and pads to water and dirt considerably increases pad wear.

The removal of dust shields will increase the cooling rate of the disc and pad assembly but it also exposes the disc and pads to particles of mud, dust and grit which adhere to the disc. This will cause a reduction in the frictional properties of the rubbing pairs. If there has to be a choice of a lower working temperature at the expense of contaminating the disc and pads or a higher working temperature, the priority would normally be in favour of protecting the rubbing surfaces from the atmospheric dust and from the road surface spray.

11.2.8 A comparison of shoe factors and shoe stability (Fig. 11.5)

A comparison of different brake shoe arrangements and the disc brake can be made on a basis of shoe factor, S or output torque compared against the variation of rubbing coefficient of frictions (Fig. 11.5). The coefficient of friction for most linings and pads ranges between 0.35 and 0.45, and it can be seen that within the normal coefficient of friction working range the order of smallest to greatest shoe factor is roughly as follows in Table 11.3.

This comparison shows that the torque output (shoe factor) for a single or two trailing shoes is only approximately one-third of the single or two leading shoe brake, and that the combination of a leading and trailing shoe is about twice that of the two trailing shoe, or roughly two-thirds of the two leading shoe arrangement (Fig. 11.5). The disc and pad's performance is very similar to the two trailing shoe layout, but with higher coefficients of friction the disc brake shoe factor rises at a faster rate than that of the two trailing shoe brake. Overall, the duo-servo shoe layout has a superior shoe factor relative to all other arrangements, amounting to roughly five times that of the two trailing shoes and just under twice that of the two leading shoe brake.

Conversely, the lining or pad stability, that is, the ability of the shoes or pads to maintain approximately the same shoe factor if there is a small change in the coefficient, due possibly to wetness or an increase in the friction material temperature, alters in the reverse order as shown in Table 11.3.
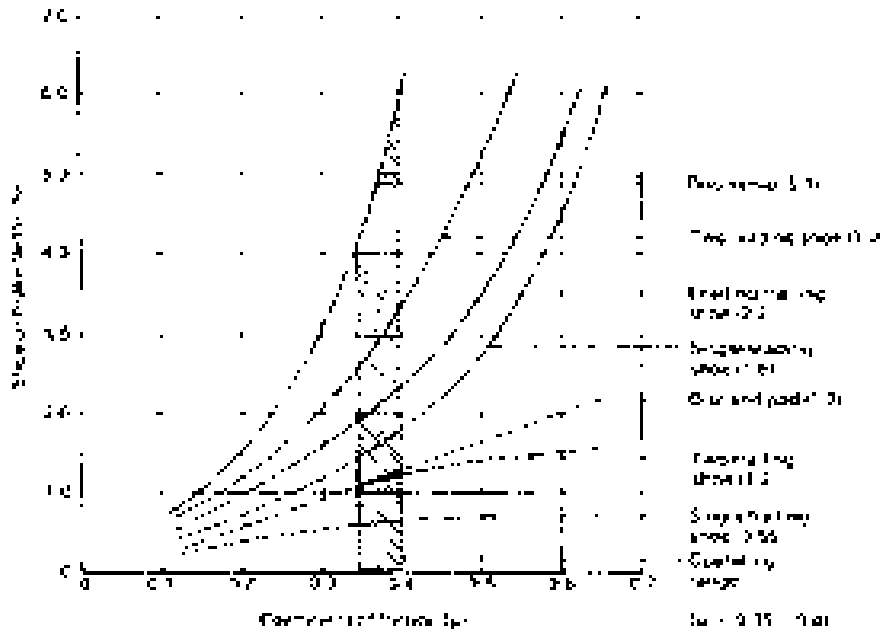
Fig. 11.5 Relationship of shoe or brake factors and the coefficient of friction for different shoe layouts and the disc brake

Generally, brakes with very high shoe factors are unstable and produce a relatively large change in shoe factor (output torque) for a small increase or decrease in the coefficient of friction between the rubbing surfaces. Layouts which have low shoe factors tend to produce a consistent output torque for a considerable shift in the coefficient of friction. Because of the instability of shoe layouts with high shoe factors, most vehicle designers opt for the front brakes to be either two leading shoes or disc and pads, and at the rear a leading and trailing shoe system. They then rely on vacuum or hydraulic servo assistance or full power air operation. Thus having, for example, a combined leading and trailing shoe brake provides a relatively high leading shoe factor but with only a moderate degree of stability, as opposed to a very stable trailing shoe which produces a very low shoe factor. The properties of each shoe arrangement complement the other to produce an effective and a reliable foundation brake. Leadings and trailing shoe brakes are still favoured on the rear wheels since they easily accommodate the hand brake mechanism and produce an extra self-energizing effect when the hand brake is applied, which in the case of the disc and pad brake is not obtainable and therefore requires a considerable greater clamping force for wheel lock condition.

11.2.9 Properties of friction lining and pad materials

riction level (Fig. 11.6) The average coefficient of friction with modern friction materials is between 0.3 and 0.5. The coefficient of friction should be sufficiently high to limit brake pedal effort and to reduce the expander leverage on commercial vehicles, but not so high as to produce *grab* and in the extreme case cause lock or *sprag* so that rotation of the drum becomes impossible. The most suitable grade of friction material must be used to match the degree of self-energization created by the shoe and pad configuration and applications.

Resistance to heat fade (Fig. 11.6) This is the ability of a lining or pad material to retain its coefficient of friction with an increase in rubbing temperature. The maximum brake torque the lining or pad is to absorb depends on the size and type of brake, gross vehicle weight, axle loading, the front to rear braking ratio and the maximum attainable speed. A good quality material should retain its friction level throughout the working temperature range of the drum and shoes or disc and pads. A reduction in the frictional level in the
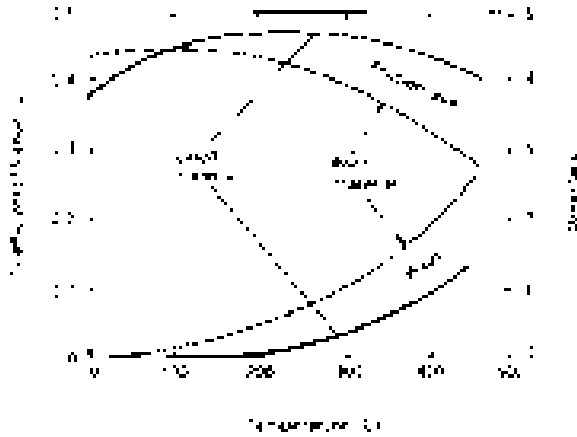
Fig. 11.6 Effects of temperature on the coefficient of friction

higher temperature range may be tolerated, provided that it progressively decreases, because a rapid decline in the coefficient of friction could severely reduce the braking power capability when the vehicle is being driven on long descents or subjected to continuous stop-start journey work. The consequences of a fall in the friction level will be greater brake pedal effort with a very poor retardation response. It has been established that changes in the frictional level which occur with rising working temperatures are caused partly by the additional curing of the pad material when it heats up in service and partly because chemical changes take place in the binder resin.

Recovery from fade (Fig. 11.6) This is a measure of the ability of a friction material to revert to its original friction level upon cooling after brake lining or pad temperature fade has taken place. The frictional characteristics of a good quality material will return on cooling, even after being subjected to repeatedly severe heating, but an inferior material may have poor recovery and the friction level may be permanently altered. Poor recovery is caused principally by a chemical breakdown in the ingredients. This may cause hardening, cracking, flaking, charring or even burning of the linings or pads. If the linings or pads are using thermoplastic binder resins a deposit may form on the rubbing surfaces which may distort the friction properties of the material.

Resistance to wear (Fig. 11.6) The life of a friction material, be it a lining or pad, will depend to a great extent upon the rubbing speed and pressure. The wear is greatly influenced by the working temperature. At the upper limits of the temperature range, the lining or pad material structure is weakened, so that there is an increase in the shear and tear action at the friction interface resulting in a higher wear rate.

Resistance to rubbing speed (Fig. 11.7) The coefficient of friction between two rubbing surfaces should in theory be independent of speed, but it has been found that the intensity of speed does tend to slightly reduce the friction level, particularly at the higher operating temperature range. Poor friction material may show a high friction level at low rubbing speeds, which may cause judder and grab when the vehicle is about to stop, but suffers from a relatively rapid decline in the friction lever as the rubbing speed increases.

Resistance to the intensity of pressure (Fig. 11.8) By the laws of friction, the coefficient of friction should not be influenced by the pressure holding the rubbing surfaces together, but with developed friction materials which are generally compounds held together with resin binders, pressure between the rubbing surfaces does reduce the level of friction. It has been found that small pressure increases at relative low pressures produce a marked reduction in the friction level, but as the intensity of
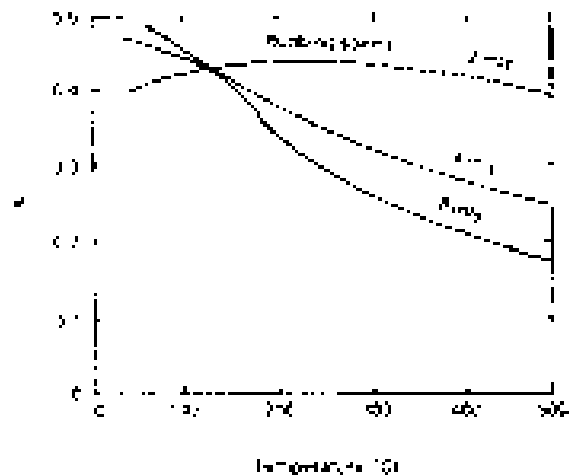
Fig. 11.7 Effects of rubbing speed on the level of friction over the temperature range
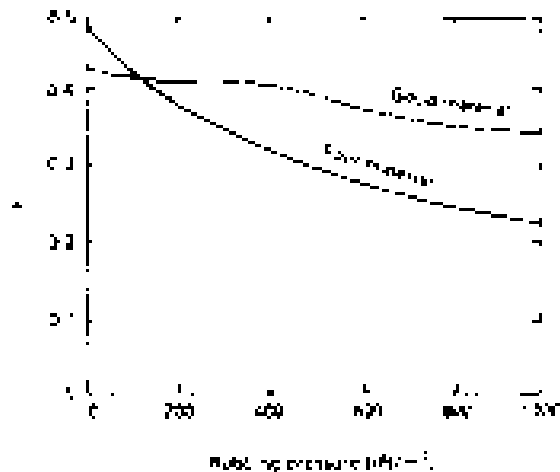
Fig. 11.8 Effects of rubbing pressure on the coefficient of friction

Fig. 11.9 Effects of water contamination on the material's friction recovery over a period of vehicle stops

pressure becomes high the decrease in friction level is much smaller. A pressure-stable lining will produce deceleration proportional to the pedal effort, but pressure-sensitive materials will require a relatively greater pedal force for a given braking performance. Disc brakes tend to operate better when subjected to high rubbing pressures, whereas shoe linings show a deterioration in performance when operating with similar pressures.

Resistance to water contamination (Fig. 11.9) All friction materials are affected by water contamination to some extent. Therefore, a safe margin of friction level should be available for wet conditions, and good quality friction materials should have the ability to recover their original friction level quickly and progressively (and not behave erratically during the drying out process). A poor quality material may either recover very slowly or

may develop over-recovery tendency (the friction level which is initially low due to the wetness rises excessively during the drying out period, falling again as the lining or pad dries out completely). Over-recovery could cause brake-grab and even wheel-lock, under certain driving conditions.

Resistance to moisture sensitivity The effects of atmospheric dampness, humidity or dew may increase the friction level for the first few applications, with the result that the brakes may become noisy and develop a tendency to grab for a short time. Moisture-sensitive friction materials should not be used on brakes which have high self-energizing characteristics.

riction materials Materials which may be used for linings or pads generally have their merits and limitations. Sintered metals tend to have a long life but have a relatively low coefficient of friction. Ceramics mixed with metals have much higher coefficient of friction but are very rigid and therefore must be made in sections. They tend to be very harsh on the drums and disc, causing them to suffer from much higher wear rates than the asbestos-based materials. There has been a tendency to produce friction materials which contain much less asbestos and much more soft metal, such as brass zinc inserts or aluminium granules. Non-asbestos materials are now available which contain DuPont's *evlar* a high strength aramid fibre. One manufacturer uses this high strength fibre in pulp form as the main body for the friction material,

ab e 11.3 Shoe factor, relative braking power and stability for various brake types

Type of brake	Shoe factor	Relative braking power	Stability
Single trailing shoe	0.55	Very low	Very high
Two trailing shoes	. 5	Very low	Very high
Disc and pad	.2	Low	High
Single leading shoe	.	High	Low
Leading and trailing shoes	2.2	Moderate	Moderate
Two leading shoes	3.0	High	Low
Duo-servo shoes	5.0	Very high	Very low

460

whereas another manufacturer uses a synthetically created body fibre derived from molten blast-furnace slag reinforced with evlar for the main body. Some non-asbestos materials do suffer from a drastic reduction in the coefficient of friction when operating in winter temperatures which, if not catered for in the brake design, may not be adequate for overnight parking brake hold.

11.3 Brake shoe e panders and ad usters

11.3.1 Self-adjusting sector and pawl brake shoe mechanism (Fig. 11.10(a, b and c))
With this leading and trailing shoe rear wheel brake layout the two shoes are actuated by opposing twin hydraulic plungers.

A downward hanging hand brake lever pivots from the top of the trailing shoe. A toothed sector lever pivots similarly from the top of the leading shoe, but its lower toothed sector end is supported and held in position with a spring loaded toothed pawl. Both shoes are interlinked with a strut bar.

Hand brake operation When the hand brake lever is applied the cable pulls the hand lever inwards, causing it to react against the strut. As it tilts it forces the trailing shoe outwards to the drum. At the same time the strut is forced in the opposite direction against the sector lever. This also pushes the leading shoe via the upper pivot and the lower toothed pawl towards the drum. The hand brake shoe expander linkage between the two shoes
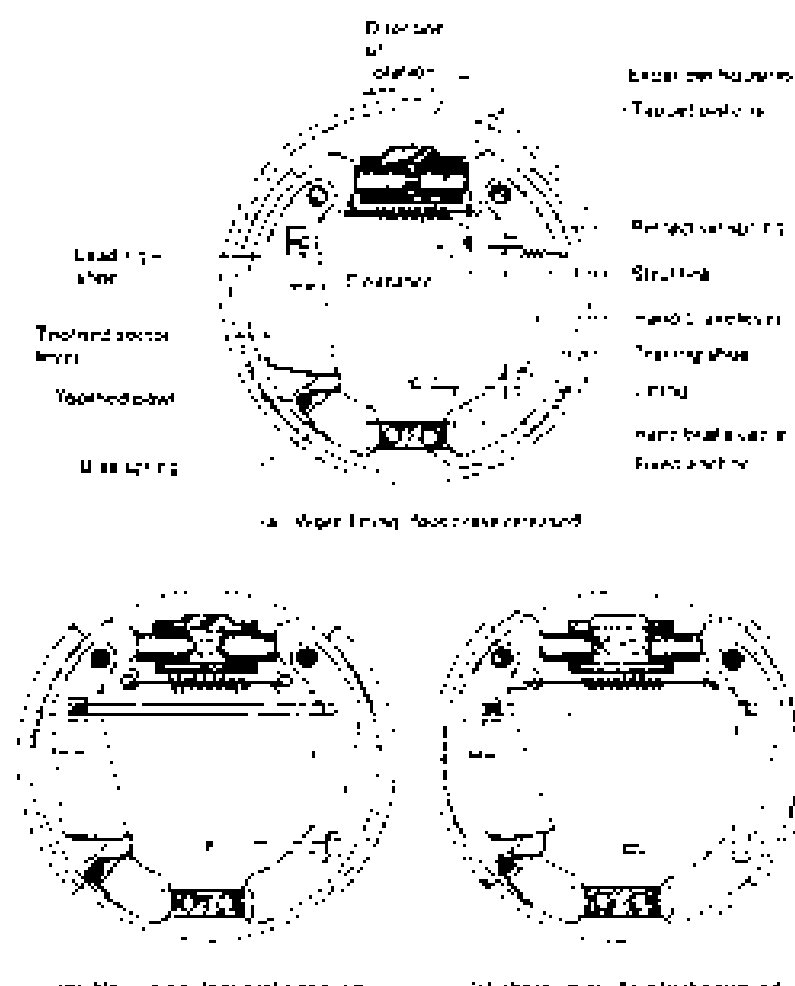
Fig. 11.10 (a–c) Self-adjusting sector and pawl shoes with forward full hand brake

461

therefore floats and equalizes the load applied by each shoe to the drum.

Automatic ad uster operation

Brake application with new linings (Fig. 11.10(b)) When the foot brake is applied, hydraulic pressure forces the twin plungers apart so that the shoes are expanded against the drum. If the linings are new and there is very little lining to shoe clearance, then the outward movement of the leading and trailing shoes will not be sufficient for the clearance between the rectangular slot in the sector lever and the strut inner edge to be taken up. Therefore the shoes will return to their original position when the brakes are released.

Brake application with worn lining (Fig. 11.10(c)) Applying the foot brake with worn linings makes the brake shoes move further apart. The first part of the outward movement of the leading shoe takes up the clearance between the strut's inner edge and the adjacent side of the rectangular slot formed in the sector lever. As the shoe moves further outwards, the strut restrains the sector lever moving with the leading shoe, so that it is forced to swivel

clockwise about its pivot. This permits the spring-loaded pawl to ride over the sector teeth until the shoe contacts the drum. At this point the pawl teeth drop into corresponding teeth on the sector, locking the sector lever to the leading shoe.

When the brakes are released, the shoes are pulled inwards by the retraction spring, but only back to where the rectangular slot contacts the outer edge of the strut (Fig. 11.10(a)). The next time the brakes are applied the shoes will not move far enough out for the slot to strut clearance to be taken up. When the brakes are released the shoe returns to the previous position and the sector to pawl ratchet action does not occur.

As the lining wears, eventually there will be sufficient slot to strut end clearance for the ratchet action to take place and for the pawl to slide over an extra sector tooth before re-engaging the sector in a more advanced position.

11.3.2 Self-adjusting quadrant and pinion brake shoe mechanism (Fig. 11.11(a–d))
This rear wheel brake layout incorporates leading and trailing brake shoes which have a hydraulically operated foot brake system and a mechanically actuated hand brake. The brake shoes are mounted
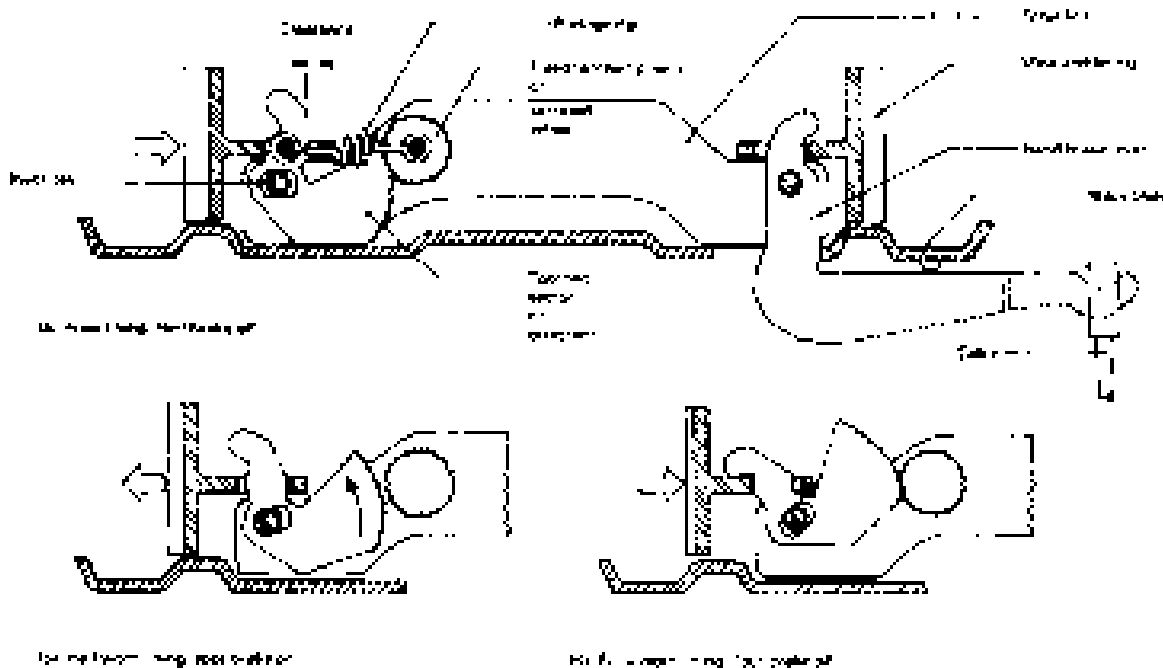
Fig. 11.11 (a–d) Self-adjusting sector and pinion brake shoes with cross-pull hand brake

462

Leading shoe

Expander housing

Tappet piston

Collar

Setting arm

Steady cup spring & band

Retraction spring

Tappet anchor

(c) Half on vehicle, foot brake off
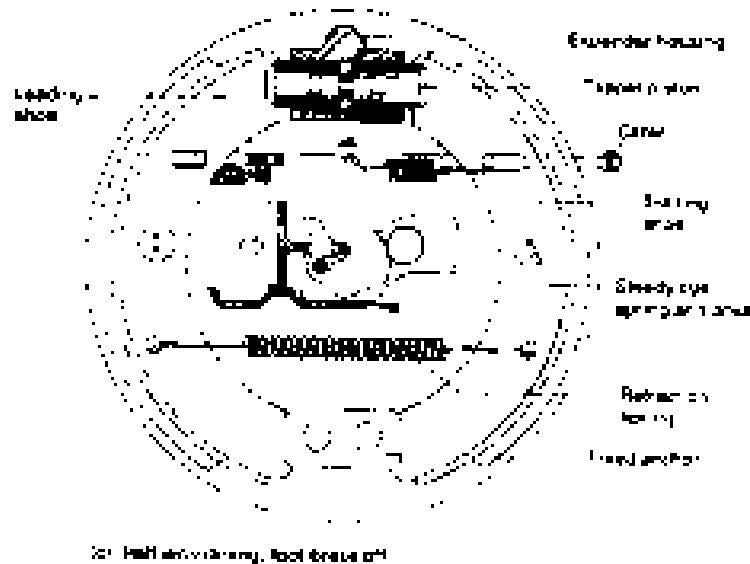
Fig. 11.11 *ontd*

on a back plate and the lower shoe tips are prevented from rotating by a fixed anchor abutment plate riveted to the back plate. The upper shoe tips are actuated by twin hydraulic plungers and a hand brake strut and lever mechanism which has a built-in automatic shoe clearance adjustment device.

The hand brake mechanism consists of a strut linking the two shoes together indirectly via a hand brake lever on the trailing shoe and a quadrant lever on the leading shoe.

Brake application with new linings (Fig. 11.11(a)) When the foot pedal is depressed, the hydraulic plungers are pushed apart, forcing the shoes into contact with the drum. When the brakes are released the retraction spring pulls the shoe inwards until the rectangular slot in the shoe web contacts the outer edge of the quadrant lever. The lever is then pushed back until the teeth on its quadrant near the end of the quadrant mesh with the fixed pinion teeth (or serrations). The position on the quadrant teeth where it meshes with the pinion determines the amount the shoe is permitted to move away from the drum and the gap between the quadrant's inner edge and the slot contact (the lining to shoe clearance).

Brake application with half worn linings (Fig. 11.11 (b and c)) When the foot brake applied, hydraulic pressure forces the shoe plungers outwards. The leading shoe moves out until the clearance between the inner edge of the quadrant lever and slot touch. Further outward shoe movement disengages the quadrant lever teeth from the adjacent pinion teeth and at the same time twists the lever. When the brakes are released the retraction spring pulls the shoes together. Initially the leading shoe web slot contacts the outer edge of the quadrant lever, and then further shoe retraction draws the quadrant lever teeth into engagement with the fixed pinion teeth, but with half worn linings the quadrant will mesh with the pinion somewhere mid-way between the outer edges of the quadrant. Consequently the shoes will only be allowed to move part of the way back to maintain a constant predetermined lining to drum clearance.

Brake application with fully worn linings (Fig. 11.11(d)) When the brakes are operated with fully worn linings the shoes move outwards before they contact the drum. During this outward movement the rear end of the slot contacts the inner edge of the quadrant lever, disengaging it from the pinion. At the same time the quadrant lever rotates until the fingered end of the lever touches the side of the shoe web. Releasing the brakes permits the shoes to retract until the quadrant lever contacts the pinion at its least return position near the quadrant's edge, furthest away from the new lining retraction position.

If any more lining wear occurs, the quadrant is not able to compensate by moving into a more

463

raised position and therefore the master cylinder pedal movement will become excessive, providing a warning that the linings need replacing.

11.3.3 Strut and cam brake shoe expander
(Fig. 11.12(a and b))
This type of shoe expander is used in conjunction with leading and trailing shoe brakes normally operated by air pressure-controlled brake actuators connected to a lever spline mounted on the camshaft, which is itself supported on a pair of plain bronze bushes.

The camshaft mounted in the expander housing is splined at its exposed end to support and secure the actuating lever (Fig. 11.12(a)). The other end, which is enclosed, supports an expander cam which has two spherical recesses to accommodate a pair of ball-ended struts. The opposite strut ends are located inside a hollow tappet plunger (follower). Mounted on the end of each tappet plunger is a

tappet head abutment which guides and supports the twin web shoes. This construction enables the linings to follow the drum shape more accurately. The tappet head abutments are inclined to provide a means for self-centralizing the brake shoes after each brake application. The cam strut lift relative to the camshaft angular movement tends to give an approximately constant lift rate for the normal angular operating range of the cam between new and worn linings conditions.

When the brakes are applied (Fig. 11.12(b)) the camshaft is rotated, causing the struts to move outwards against the hollow tapper plungers. The tappet head abutments force the shoes into contact with the drum, thereby applying the brakes.

11.3.4 edge shoe adjuster unit (Fig. 11.13)
The adjuster housing is made from malleable iron and is spigoted and bolted firmly to the back plate (Fig. 11.13). A hardened steel wedge is employed
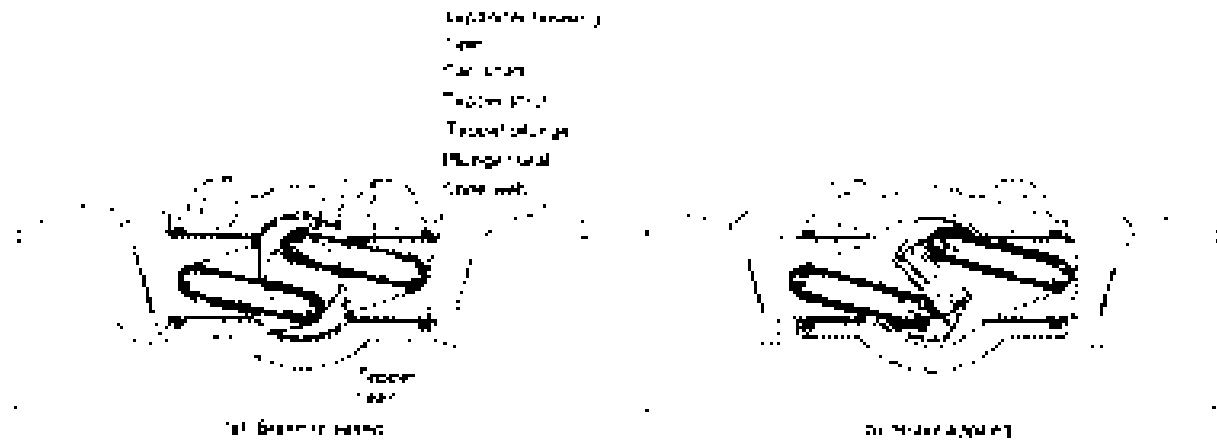
Fig. 11.12 (a and b) Strut and cam brake shoe expander
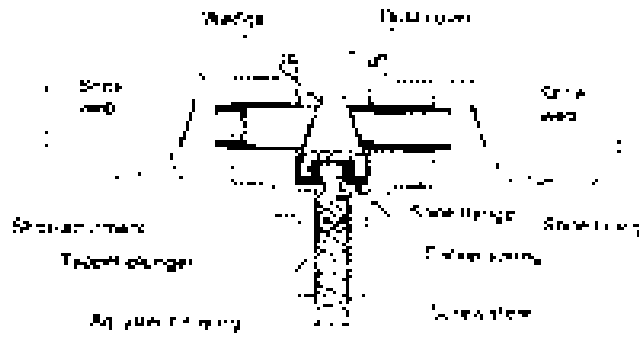
Fig. 11.13 Wedge shoe adjuster unit

464

with a screw adjuster stem rotating within the wedge which does not rotate, but moves at right angles to the inclined faced tappet plungers. So that accurate adjustment for each brake assembly is possible, a clicker spring is located between the screwed stem and the wedge. This spring fits onto two flats provided on the screw stem (not shown). The clicker spring has two embossed dimples which align and clip into shallow holes formed in the back of the wedge when the shoes are being adjusted; they therefore enable a fine adjustment to be made while also preventing the adjuster screw stem unwinding on its own while in service.

11.3.5 S cam shoe expander (Fig. 11.14)

Cam e pander requirements   The object of a cam brake shoe expander (Fig. 11.14(a and b)) is to convert an input camshaft leverage torque into a shoe tip force. The shape of the cam profile plays a large part in the effective expansion thrust imposed on the shoe tips as the shoe linings wear and the clearance between the drum and linings increase.

Early S-shaped cams were derived from an Archimedean spiral form of locus which gives a constant rate of lift per degree of cam rotation, but varying cam radius. The present tendency is for the S cam to be generated from an involute spiral (Fig. 11.14(a)) which gives a slight reduction in lift per degree of cam rotation, but maintains a constant cam effective radius so that the shoe tip force always acts in the same direction relative to the cam shoe roller, no matter which part of the cam profile is in contact with the roller (Fig. 11.14(b)). By these means the shoe tip force will remain approximately constant for a given input torque for the whole angular movement of the cam between new and worn lining conditions. Note this does not mean that the effective input torque will be constant. This depends upon the push or pull rod and the slack adjuster lever remaining perpendicular to each other which is unlikely.

Cam profile geometry   The involute to a base circle is generated when a straight line is rolled round a circle without slipping; points on the line will trace out an involute. The involute profile may be produced by drawing a base circle and a straight line equal to its circumference and dividing both into the same number of equal parts (Fig. 11.14(a)). From the marked points on the circle draw tangents to represent successive positions of the generated line. Step off the unwrapped portion of the circumference along each tangent and then plot a smooth curve passing through the extending tangential lengths. The locus generated is the involute to base circle, this shape being the basic shape of the so-called S cam.

Cam and shoe working conditions   With new shoe linings the leading shoe works harder than the trailing shoe so that initially the leading shoe wear will be higher than that of the trailing shoe. If there is adequate camshaft to bush bearing clearance, shoe wear will eventually be sufficient to permit the camshaft to float between the shoe tips, allowing the trailing shoe to produce the same friction drag as the leading shoe, thus producing the *e ual work* condition.

If the shoe tip force applied by the cam is equal, then the camshaft floats on its bearing. In practice, because the shoe tip force is not always equal, a resultant reaction force input will be provided by the camshaft to maintain equilibrium. Therefore the frictional force between the shaft and bearing can be significant in the mechanical losses between the input camshaft torque and the shoe tip force. The input camshaft torque may be derived from both the shaft frictional torque and the cam to roller contact torque (Fig. 11.14(c)).

Let $\quad_c$ = coefficient of camshaft to bearing friction
$\quad R$ = resultant camshaft radial load (N)
$\quad r_c$ = camshaft radius (m)
$\quad r_b$ = base circle radius (m)
F_1 and F_2 = roller contact forces (N)
Then
Camshaft frictional
$$torque = {}_c R r_c \ (Nm)$$
and
Cam to roller torque $= (F_1 + F_2)r_b$ (Nm)
Therefore
Input camshaft
$$torque \ T_c = {}_c R r_c + (F_1 + F_2)r_b \ (Nm)$$

Cam design considerations   To give the highest shoe factor, that is the maximum shoe frictional drag to input torque, a low rate of cam lift is desirable. This conflicts with the large total lift needed to utilize the full lining thickness which tends to be limited to 19 mm.

Typical rates of cam lift vary from 0.2 to 0.4 mm/deg which correspond to brake factors of about 12 to 16 with the involute cam profile.
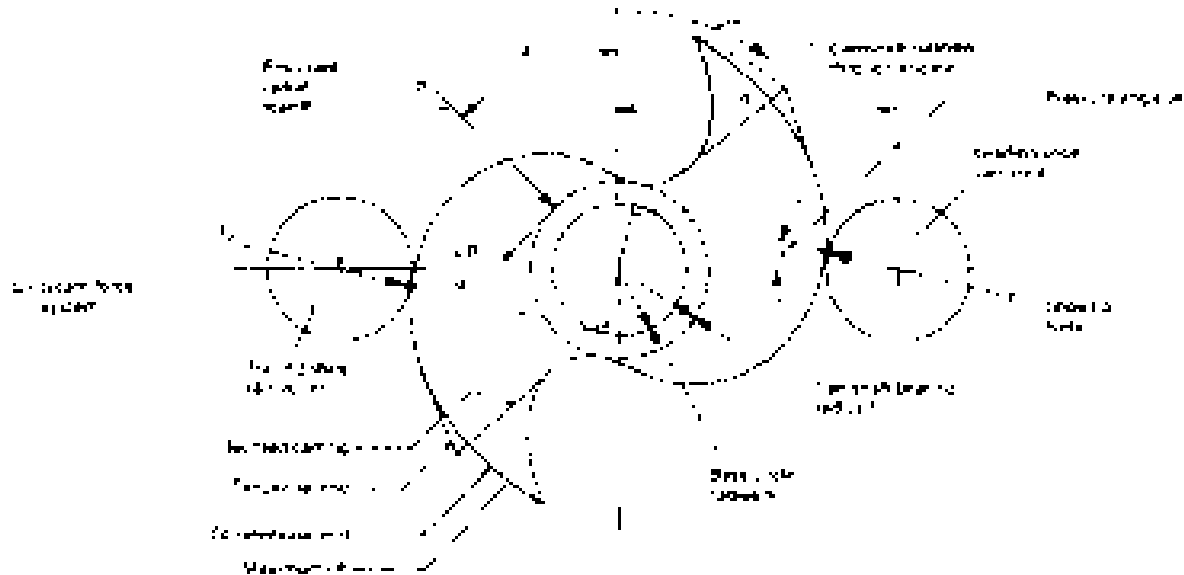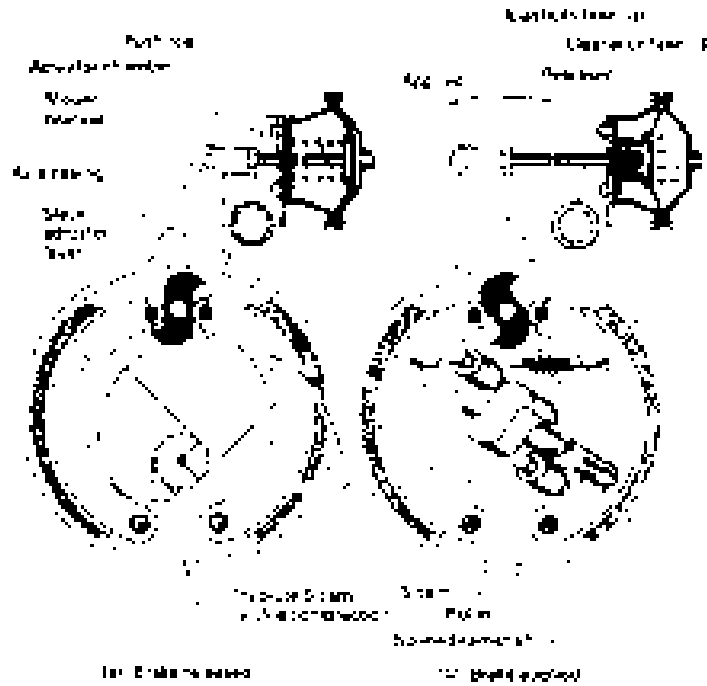
Fig. 11.14 (a–c) Air operated foundation brake assembly

As the cam lifts (Fig. 11.14(c)) the pressure angle which is made between the cam and roller centre lines and the base circle tangential line decreases. For the cam to be self-returning the pressure angle should not be permitted to be reduced below 10°.

One approach to maximize cam lift without the rollers falling off the end of the cam in the extreme wear condition is to use the involute cam up to the point where the lining rivets would contact the drum and relining would be required. Beyond this

466

point the cam is continued in a straight line, tangential to the cam profile (Fig. 11.14(c)). By this method, total cam lift is achieved for the normal thickness of lining within the designed angular movement of the cam, which is not possible with the conventional involute cam. Shoe tip force efficiency does drop off in the final tangential lift cam range but this is not a serious problem as it is very near the end of the linings' useful life. One important outcome of altering the final involute profile is that the blunting of the cam tips considerably strengthens the cam.

11.3.6 *edge type brake shoe expander with automatic adjustment* (Fig. 11.15)

The automatic brake shoe adjustment provides a self-adjusting mechanism actuated by the expander movement during the on/off brake application cycle, enabling a predetermined lining to drum clearance to be maintained. When a brake application takes place, an adjusting pawl mechanism senses the movement of an adjusting sleeve located in one of the wedge expander plungers. If the sleeve travel exceeds 1.52 mm, the spring-loaded pawl acting on the sleeve teeth drops into the next tooth and automatically makes the adjusting screw wind out a predetermined amount. An approximate 1.14 mm lining to drum clearance will be maintained when the brakes are released, but if the adjusting sleeve and plunger outward travel is less than 1.52 mm, then the whole plunger assembly will move back to its original position without any adjustment being made.

escription (Fig. 11.15) The automatic adjustment is built into one of the expander plungers. With this construction the adjusting screw is threaded into an adjusting sleeve, the sleeve being a free fit inside the hollow plunger. A hollow cap screw, spring, and an adjusting pawl are pre-assembled and act as a plunger guide. The end of the adjusting pawl has sawtooth type teeth which engage corresponding helical teeth on the outside of the adjusting sleeve.

Operation (Fig. 11.16) As the brakes are applied, the plunger sleeve and screw move outwards and the sloping face of the teeth on the adjusting sleeve lifts the adjusting pawl against the spring. When the brake is released, the rollers move down both the central wedge and the two outer plungers' inclined planes to their fully released position. As the linings wear, both the plunger strokes and

resulting pawl lift gradually increase until the pawl climbs over and drops into the next tooth space. The next time, when the brake is released and the plunger is pushed back into its bore, the upright face of the pawl teeth prevents the sleeve moving directly back. It permits the sleeve to twist as its outside helical teeth slide through the corresponding guide pawl teeth in its endeavour for the whole plunger assembly to contract inwards to the off position, caused by the inward pull of the shoe retraction spring. The partial rotation of the adjusting sleeve unscrews and advances the adjusting screw to a new position. This reduces the lining clearance. This cycle of events is repeated as the lining wears. The self-adjustment action only operates in the forward vehicle direction. Once the brake shoes have been installed and manually adjusted no further attention is necessary until the worn linings are replaced.

11.3.7 *anual slack adjuster* (Fig. 11.17)

Purpose A slack adjuster is the operating lever between the brake actuator chamber push rod and the camshaft. It is used with 'S' type cam shoe expanders and features a built-in worm and worm wheel adjustment mechanism enabling adjustment to be made without involving the removal and alteration of the push rod length.

Operation (Fig. 11.17) The slack adjuster lever incorporates in its body a worm and worm wheel type adjuster (Fig. 11.17). The slack adjuster lever is attached indirectly to the splined camshaft via the internal splines of the worm wheel located inside the slack adjuster body. For optimum input leverage the slack adjuster lever and the push rod should be set to maintain an inside angle just greater than 90° with the brakes fully applied. Once the push rod length has been set, further angular adjustment of the 'S' cam is made by rotating the worm shaft so that the large gear reduction between the meshing worm and worm wheel will slowly turn the worm wheel and camshaft until the cam flanks take up the excess shoe lining to drum clearance. Owing to the low reverse efficiency of the worm and worm wheel gearing, the worm and shaft will not normally rotate on its own. To prevent the possibility of the worm and shaft unwinding, caused perhaps by transmitted oscillatory movement of the slack adjuster during periods of applying and releasing the brakes, a lock sleeve is utilized.

Fig. 11.15 Twin wedge foundation brake expander and automatic adjuster

Ad ustment (Fig. 11.17) Cam adjustment is provided by the hexagon head of the worm shaft situated on the side of the slack adjuster body (Fig. 11.17). To adjust the cam relative to the slack adjuster lever, the lock sleeve is depressed against the worm lock spring by a suitable spanner until the worm shaft is free to turn. The worm shaft is then rotated with the spanner until all the excess

468

Fig. 11.16 Wedge expander and automatic clearance adjuster

shoe lining to drum clearance is eliminated. The worm shaft is then prevented from unwinding by the worm lock spring forcing the lock sleeve against the hexagonal head of the worm shaft. The removal of the spanner permits the worm lock spring to push the internally serrated lock sleeve up to and over the hexagonal bolt head. To prevent the lock sleeve rotating, a guide pin fixed in the slack adjuster body aligns with a slot machined on the sleeve.

11.3.8 Automatic slack adjuster
(Fig. 11.18(a, b, c and d))

Purpose Once set up automatic slack adjusters need no manual adjustment during the life of the brake linings. Self-adjustment takes place whilst the brakes are released (when the clearance between lining and drum exceeds 1.14 mm). This designed clearance ensures that there is no brake drag and adequate cooling exists for both shoe and drum.

Operation (Fig. 11.18)

Automatic adjustment When the foot or hand brake valve is operated, the brake actuator chamber push rod connected to the slack adjuster will rotate the slack adjuster body and camshaft in an anticlockwise direction (Fig. 11.18(a)). The restraining link rod of the ratchet lever which is connected to some fixed point on the axle will cause the ratchet lever to rotate clockwise relative to the slack adjuster body, and thus the ratchet pawl will ride up one of the teeth of the ratchet wheel.

When adjustment is required, the relative movement between the ratchet lever and the main slack adjuster body is sufficient for the pawl to ride over one complete tooth space and engage the next tooth, following which, on release of the brake (Fig. 11.18(b)), as the slack adjuster moves to its 'off' position, the ratchet wheel will be rotated by the amount of the one tooth space. As the ratchet
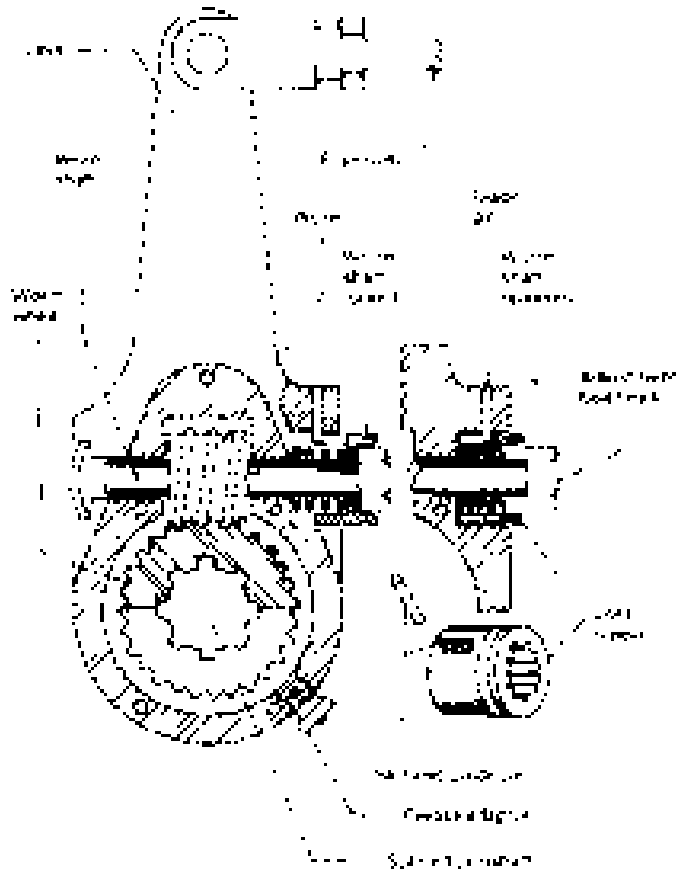
Fig. 11.17   Manual slack adjuster (Clayton Dewandre)

wheel rotates, so too will the ratchet worm, which in turn transmits motion to the ratchet worm wheel and splined main worm shaft. Thus the main worm, and subsequently the main worm wheel, are rotated, causing the slack adjuster lever to take up a new position relative to the camshaft splined into the main worm wheel.

Adjustment will only take place if the pawl moves sufficiently around the ratchet wheel to collect the next tooth. This can only occur if the relative movement between the ratchet lever and the main slack adjuster body exceeds an angle of 8.18°.

nitial manual ad ustment (Fig. 11.18(c))   Manual adjustment after fitting new shoe linings can be made by unscrewing the hexagonal cap from the extended slack adjuster body. This releases the ratchet worm wheel from the ratchet worm by the action of the spring behind the worm wheel, which

causes the worm wheel to slide outward along the splines to the stop washer. A spanner is then fitted over the square on the end of the ratchet worm wheel and rotated until the shoe to drum excess clearance has been taken up. After the shoes have been manually adjusted, the ratchet worm wheel should be pushed back until it meshes with the ratchet worm. The end cap should then be replaced.

11.4   isc brake pad support arrangements

11.4.1 *Swing yoke type brake caliper* (Fig. 11.19)
This is a lightweight, single cylinder, disc brake caliper. The caliper unit consists of a yoke made from a rigid steel pressing, a cylinder assembly, two pads and a carrier bracket which is bolted to the suspension hub carrier. The cylinder is attached by a tongue and groove joint to one side of the yoke frame whilst the yoke itself is allowed to pivot at

470
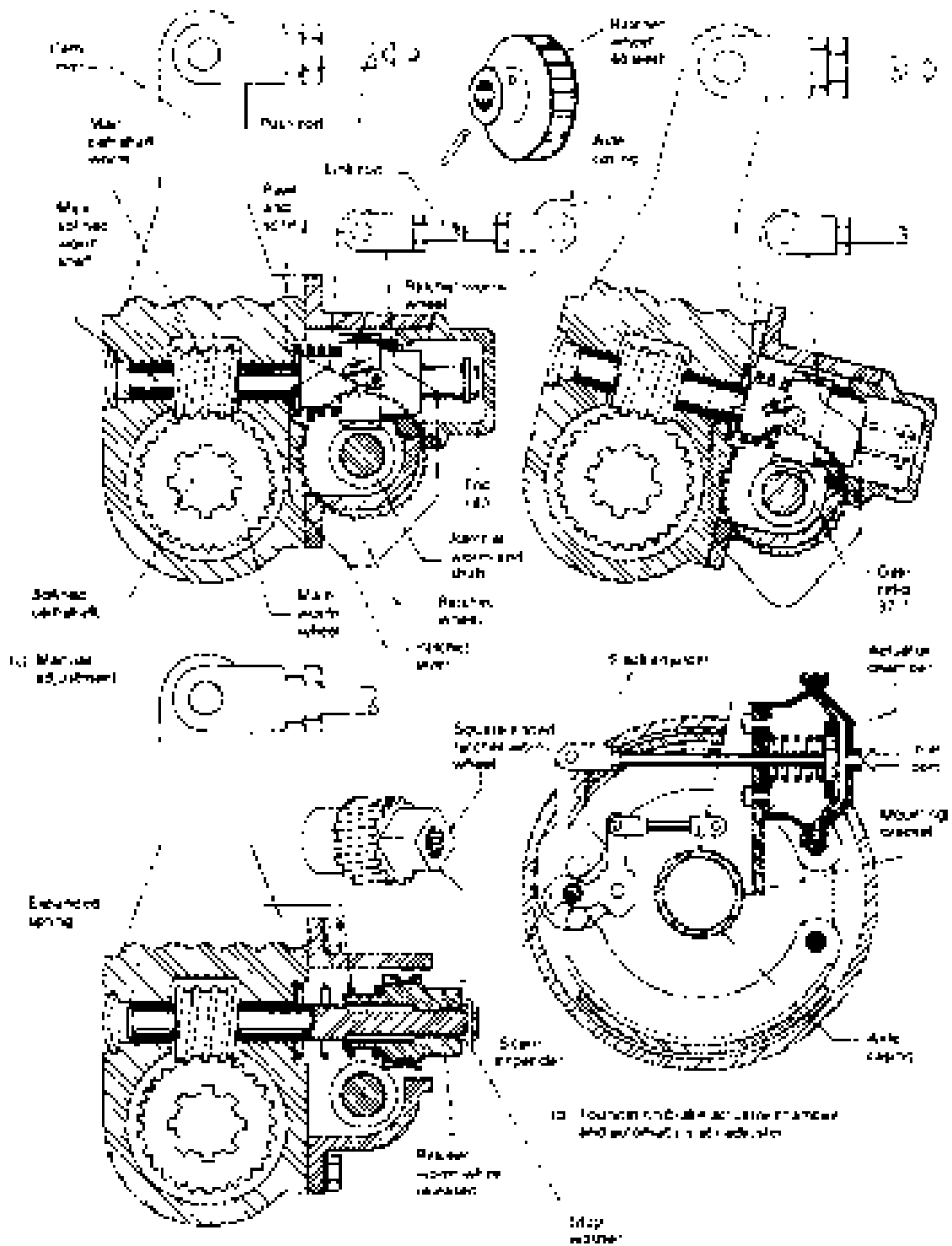
Fig. 11.18 (a–d) Automatic slack adjuster

one end on it supporting carrier bracket. The disc is driven by the transmission drive shaft hub on which it is mounted and the lining pads are positioned and supported on either side of the disc by the rectangular aperature in the yoke frame.

Operation (Fig. 11.19) When the foot brake is applied generated hydraulic pressure pushes the piston and inboard pad against their adjacent disc face. Simultaneously, the hydraulic reaction will move the cylinder in the opposite direction away from the disc. Consequently, as the outboard pad and cylinder body are bridged by the yoke, the latter will pivot, forcing the outboard pad against the opposite disc face to that of the inboard pad.

As the pads wear, the yoke will move through an arc about its pivot, and to compensate for this tilt the lining pads are taper shaped. During the wear life of the pad friction material, the amount of taper gradually reduces so that in a fully worn state the remaining friction material is approximately parallel to the steel backing plate.

The operating clearance between the pads and disc is maintained roughly constant by the inherent distortional stretch and retraction of the pressure seals as the hydraulic pressure is increased and reduced respectively, which accordingly moves the piston forwards and back.

11.4.2 Sliding yoke type brake caliper
(Fig. 11.20)
With this type of caliper unit, the cylinder body is rigidly attached to the suspension hub carrier, whereas the yoke steel pressing fits over the cylinder body and is permitted to slide between parallel grooves formed in the cylinder casting.

Operation (Fig. 11.20) When the foot brake is applied, hydraulic pressure is generated between the two pistons. The hydraulic pressure pushes the piston apart, the direct piston forces the direct pad against the disc whilst the indirect piston forces the yoke to slide in the cylinder in the opposite direction until the indirect pad contacts the outstanding disc face.

Further pressure build-up causes an equal but opposing force to sandwich the disc between the friction pads. This pressure increase continues until the desired retardation force is achieved.

During the pressure increase the pressure seals distort as the pistons move apart. When the hydraulic pressure collapses the rubber pressure seals retract
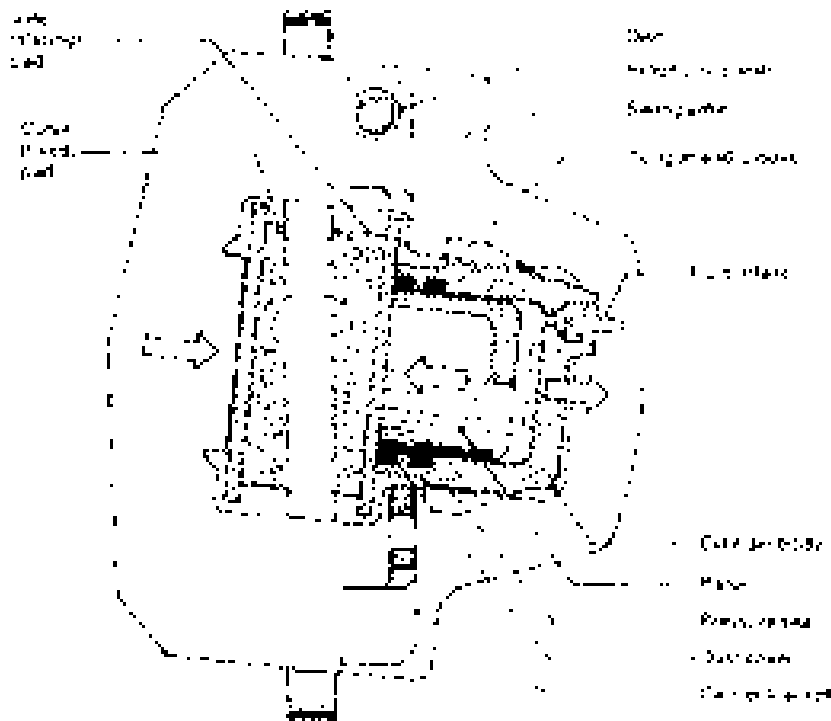
Fig. 11.19 Swing yoke type brake caliper

472
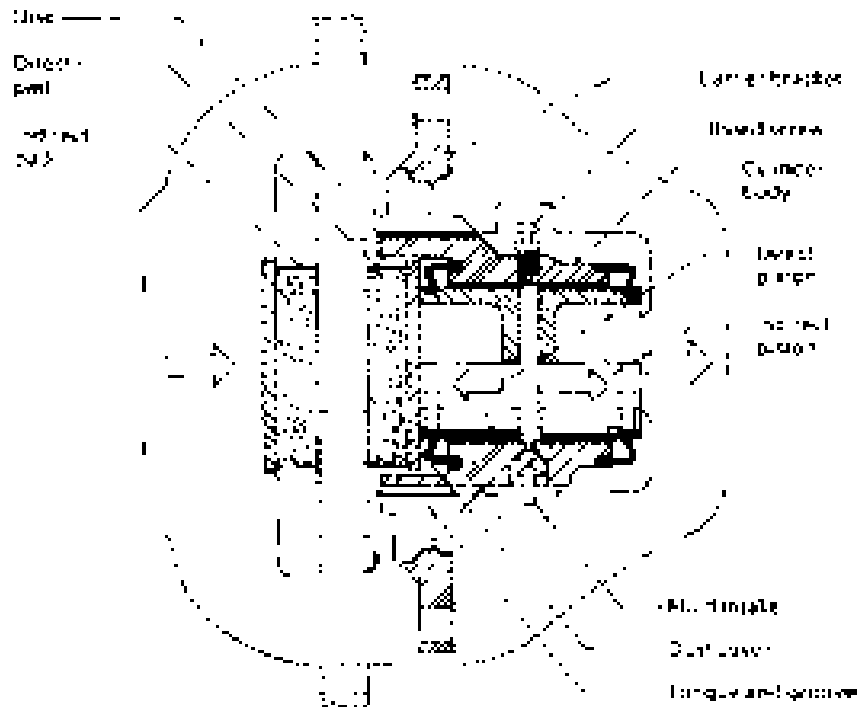
Fig. 11.20 Sliding yoke type brake caliper

and withdraw the pistons and pads from the disc surface so that friction pad drag is eliminated.

oke rattle between the cylinder and yoke frame is reduced to a minimum by inserting either a wire or leaf type spring between the sliding joints.

11.4.3 Sliding pin type brake caliper (Fig. 11.21)
The assembled disc brake caliper unit comprises the following; a disc, a carrier bracket, a cylinder caliper bridge, piston and seals, friction pads and a pair of support guide pins.

The carrier bracket is bolted onto the suspension hub carrier, its function being to support the cylinder caliper bridge and to absorb the brake torque reaction.

The cylinder caliper bridge is mounted on a pair of guide pins sliding in matched holes machined in the carrier bracket. The guide pins are sealed against dirt and moisture by dust covers so that equal frictional sliding loads will be maintained at all times. On some models a rubber bush sleeve is fitted to one of the guide pins to prevent noise and to take up brake deflection.

Frictional drag of the pads is not taken by the guide pins, but is absorbed by the carrier bracket. Therefore the pins only support and guide the caliper cylinder bridge.

As with all other types of caliper units, pad to disc free clearance is obtained by the pressure seals which are fitted inside recesses in the cylinder wall and grip the piston when hydraulic pressure forces the piston outwards, causing the seal to distort. When the brakes are released and the pressure is removed from the piston crown, the strain energy of the elastic rubber pulls back the piston until the pressure seal has been restored to its original shape.

Operation (Fig. 11.21)   When the foot brake is applied, the hydraulic pressure generated pushes the piston and cylinder apart. Accordingly the inboard pad moves up to the inner disc face, whereas the cylinder and bridge react in the opposite sense by sliding the guide pins out from their supporting holes until the outboard pad touches the outside disc face. Further generated hydraulic pressure will impose equal but opposing forces against the disc faces via the pads.

11.4.4 Sliding cylinder body type brake caliper
(Fig. 11.22)
This type of caliper unit consists of a carrier bracket bolted to the suspension hub carrier and a single piston cylinder bridge caliper which straddles

473
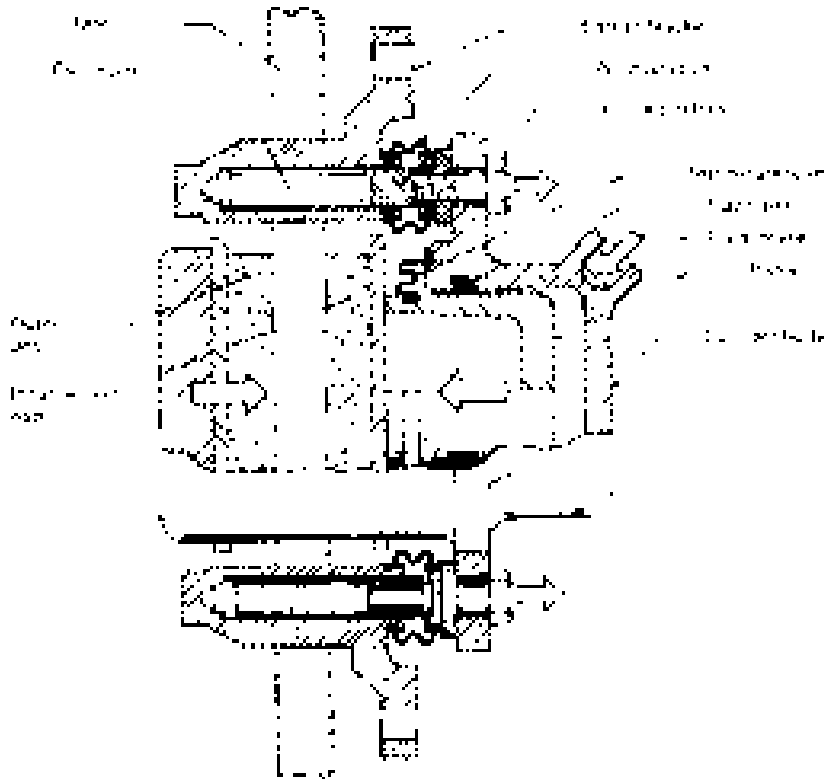
Fig. 11.21 Slide pin type brake caliper

the disc and is allowed to slide laterally on guide keys positioned in wedge-shaped grooves machined in the carrier bracket.

Operation (Fig. 11.22) When the foot brake is applied, the generated hydraulic pressure enters the cylinder, pushing the piston with the direct acting pad onto the inside disc face. The cylinder body caliper bridge is pushed in the opposite direction. As a result, the caliper bridge reacts and slides in its guide groove at right angles to the disc until the indirect pad contacts the outside disc face, thereby equalling the forces acting on both sides of the disc.

A pad to disc face working clearance is provided when the brakes are released by the retraction of the pressure seal, drawing the piston a small amount back into the cylinder after the hydraulic pressure collapes.

To avoid vibration and noise caused by relative movement between the bridge caliper and carrier bracket sliding joint, anti-rattle springs are normally incorporated alongside each of the two-edge-shaped grooves.

11.4.5 Twin floating piston caliper disc brake with hand brake mechanism (Fig. 11.23)

This disc brake unit has a pair of opposing pistons housed in each split half-caliper. The inboard half-caliper is mounted on a flanged suspension hub carrier, whereas the other half straddles the disc and is secured to the rotating wheel hub. Lining pads bonded to steel plates are inserted on each side of the disc between the pistons and disc rubbing face and are held in position by a pair of steel pins and clips which span the two half-calipers. Brake fluid is prevented from escaping between the pistons and cylinder walls by rubber pressure seals which also serve as piston retraction springs, while dirt and moisture are kept out by flexible rubber dust covers.

oot brake application (Fig. 11.23) Hydraulic pressure, generated when the foot brake is applied, is transferred from the inlet port to the central half-caliper joint, where it is then transmitted along passages to the rear of each piston.

As each piston moves forward to take the clearance between the lining pads and disc, the piston

474
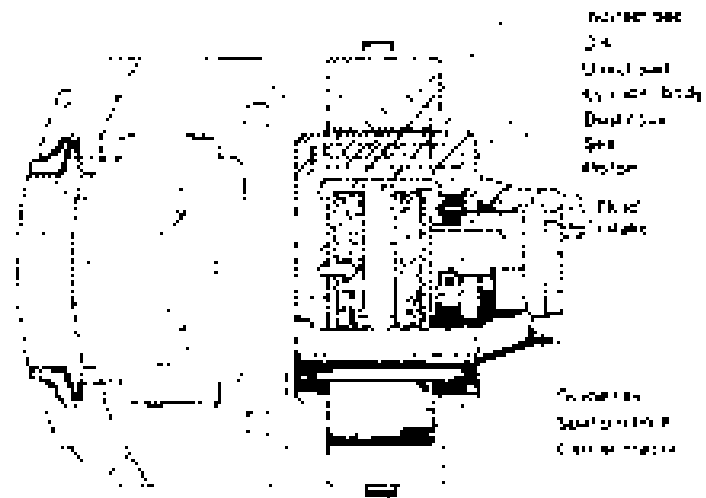
Fig. 11.22 Slide cylinder body brake caliper
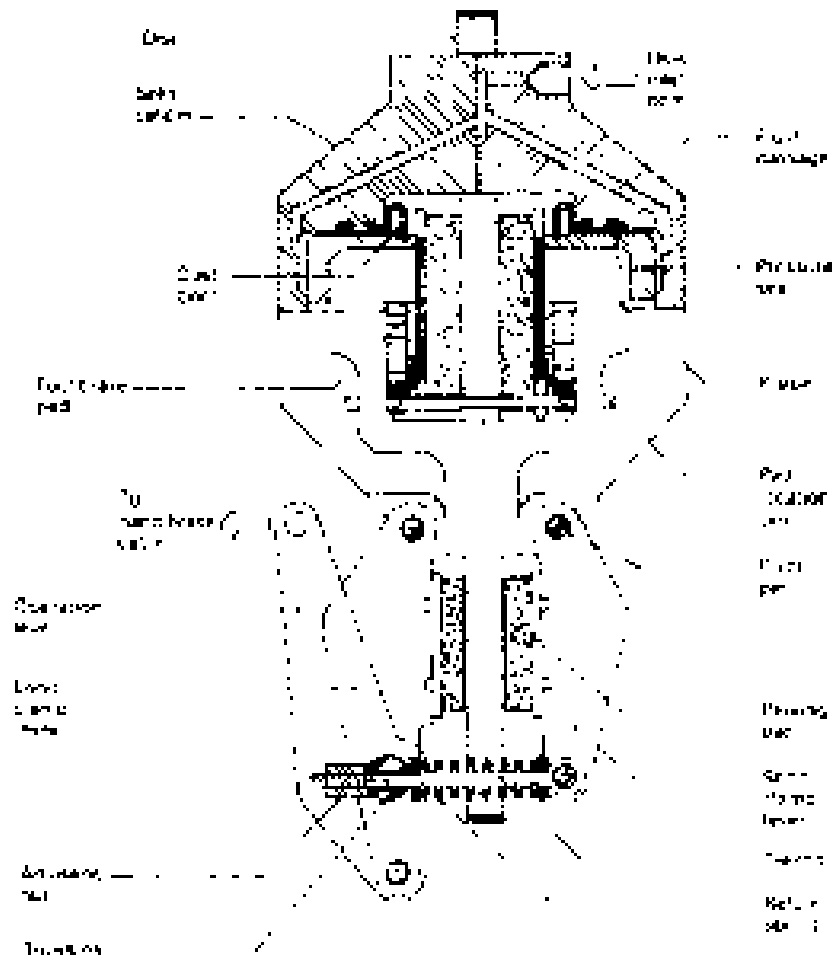
Fig. 11.23 Twin floating piston caliper disc brake with hand brake mechanism

pressure seals are distorted. Further pressure build-up then applies an equal but opposite force by way of the lining pads to both faces of the disc, thereby creating a frictional retarding drag to the rotating disc. Should the disc be slightly off-centre, the pistons will compensate by moving laterally relative to the rubbing faces of the disc.

Releasing the brakes causes the hydraulic pressure to collapse so that the elasticity within the distorted rubber pressure seals retracts the pistons and pads until the seals convert to their original shape.

The large surface area which is swept on each side of the disc by the lining pads is exposed to the cooling airstream so that heat dissipation is maximized.

Hand brake application (Fig. 11.23) The hand brake mechanism has a long and short clamping lever fitted with friction pads on either side of the disc and pivots from the lower part of the caliper. A tie rod with an adjusting nut links the two clamping levers and, via an operating lever, provides the means to clamp the disc between the friction pads. Applying the hand brake pulls the operating lever outwards via the hand brake cable, causing the tie rod to pull the short clamp lever and pad towards the adjacent disc face, whilst the long clamp and pad is pushed in the opposite direction against the other disc face. As a result, the lining pads grip the disc with sufficient force to prevent the car wheels rolling on relatively steep slopes.

To compensate for pad wear, the adjustment nut should be tightened periodically to give a maximum pad to disc clearance of 0.1 mm.

11.4.6 Combined foot and hand brake caliper with automatic screw adjustment (Bendix)

This unit provides automatic adjustment for the freeplay in the caliper's hand brake mechanism caused by pad wear. It therefore keeps the hand brake travel constant during the service life of the pads.

The adjustment mechanism consists of a shouldered nut which is screwed onto a coarsely threaded shaft. Surrounding the nut on one side of the shoulder or flange is a coiled spring which is anchored at its outer end via a hole in the piston. On the other side of the shouldered nut is a ball bearing thrust race. The whole assembly is enclosed in the hollow piston and is prevented from moving out by a thrust washer which reacts against the thrust bearing and is secured by a circlip to the interior of the piston.

oot brake application (Fig. 11.24(a)) When the hydraulic brakes are applied, the piston outward movement is approximately equal to the predetermined clearance between the piston and nut with the brakes off, but as the pads wear, the piston takes up a new position further outwards, so that the normal piston to nut clearance is exceeded.

If there is very little pad wear, hydraulic pressure will move the piston forward until the pads grip the disc without the thrust washer touching the ball race. However, as the pads wear, the piston moves forward until the thrust washer contacts the ball race. Further outward movement of the piston then forces the thrust washer ball race and shouldered nut together in an outward direction. Since the threaded shaft is prevented from rotating by the strut and cam, the only way the nut can move forward is by unwinding on the screw shaft. Immediately the nut attempts to turn, the coil spring uncoils and loses its grip on the nut, permitting the nut to screw out in proportion to the piston movement.

On releasing the foot brake, the collapse of the hydraulic pressure enables the pressure seals to withdraw the pads from the disc. Because the axial load has been removed from the nut, there is no tendency for it to rotate and the coil spring therefore contracts, gripping the nut so that it cannot rotate. Note that the outward movement of the nut relative to the threaded shaft takes up part of the slack in the mechanical linkage so that the hand brake lever movement remains approximately constant throughout the life of the pads. The threaded shaft and nut device does not influence the operating pad to disc clearance when the hydraulic brakes are applied as this is controlled only by the pressure seal distortion and elasticity.

Hand brake application (Fig. 11.24(b)) Applying the hand brake causes the cable to rotate the camshaft via the cam lever, which in turn transfers force from the cam to the threaded shaft through the strut. The first part of the screwed shaft travel takes up the piston to nut end-clearance. With further screw shaft movement the piston is pushed outwards until the pad on the piston contacts the adjacent disc face. At the same time an equal and opposite reaction causes the caliper cylinder to move in the opposite direction until the outside pad and disc face touch. Any further outward movement of the threaded shaft subsequently clamps the disc in between the pads. Releasing the hand brake lever relaxes the pad grip on the disc
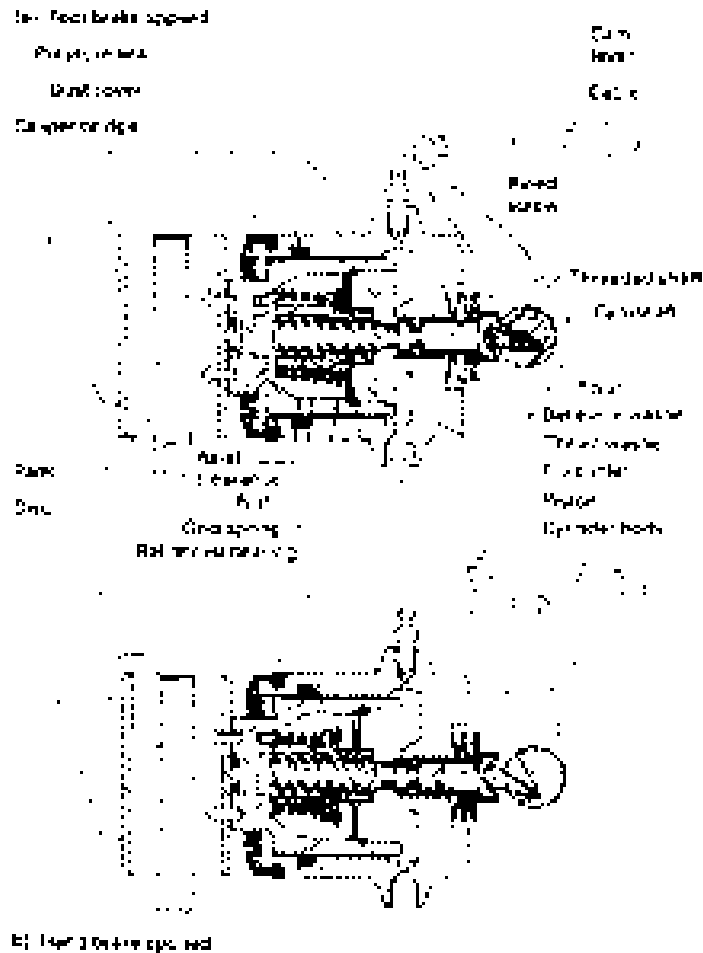
Fig. 11.24 (a and b) Combined foot and hand brake caliper with automatic screw adjustment

with the assistance of the Belleville washers which draws back the threaded shaft to the 'off' position to avoid the pads binding on the disc.

11.5 ual- or split-line braking systems

Dual- or split-line braking systems are used on all cars and vans to continue to provide some degree of braking if one of the two hydraulic circuits should fail. A tandem master cylinder is incorporated in the dual-line braking system, which is in effect two separate master cylinder circuits placed together end on so that it can be operated by a common push rod and foot pedal. Thus, if there is a fault in one of the hydraulic circuits, the other pipe line will be unaffected and therefore will still actuate the caliper or drum brake cylinders it supplies.

11.5.1 Front to rear brake line split
(Fig. 11.25(a))

With this arrangement, the two separate hydraulic pipe lines of the tandem master cylinder are in circuit with either both the front or rear caliper or shoe expander cylinders. The weakness with this pipe line split is that roughly two-thirds of the braking power is designed to be absorbed by the front calipers, and only one-third by the rear brakes. Therefore if the front brakes malfunction, the rear brake can provide only one-third of the original braking capacity.

11.5.2 Diagonally front to rear brake split
(Fig. 11.25(b))

To enable the braking effort to be more equally shared between each hydraulic circuit (if a fault should occur in one of these lines), the one front
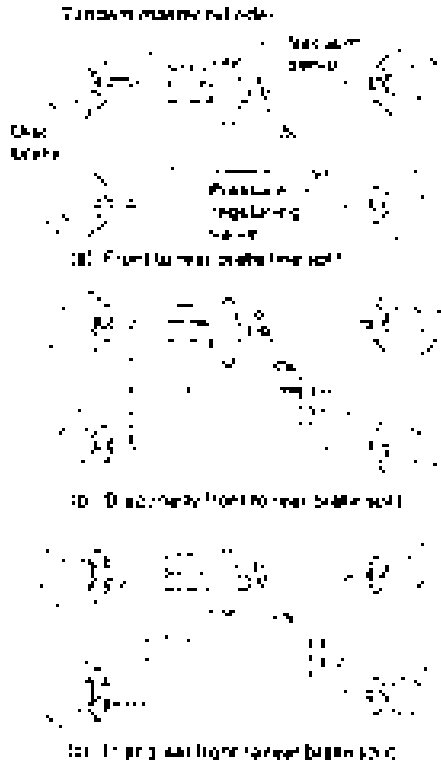
Fig. 11.25 (a–c) Dual- or split-line braking systems

and one diagonally opposite rear wheel are connected together. Each hydraulic circuit therefore has the same amount of braking capacity and the ratio of front to rear braking proportions do not influence the ability to stop. A diagonal split also tends to retard a vehicle on a relatively straight line on a dry road.

11.5.3 Triangular front to rear brake split
(Fig. 11.25(b))
This hydraulic pipe line system uses front calipers which have two independent pairs of cylinders, and at the rear conventional calipers or drum brakes. Each fluid pipe line circuit supplies half of each front caliper and one rear caliper or drum brake cylinder. Thus a leakage in one or the other hydraulic circuits will cause the other three pairs of calipers or cylinders or two pairs of caliper cylinders and one rear drum brake cylinder to provide braking equal to about 80% of that which is possible when both circuits are operating. When one circuit is defective, braking is provided on three wheels; it is then known as a *triangular split*

11.5.4 Compensating port type tandem master cylinder (Fig. 11.26(a–d))
Tandem master cylinders are employed to operate dual-line hydraulic braking systems. The master cylinder is composed of a pair of pistons functioning within a single cylinder. This enables two independent hydraulic cylinder chambers to operate. Consequently, if one of these cylinder chambers or part of its hydraulic circuit develops a fault, the other cylinder chamber and circuit will still continue to effectively operate.

Brakes off (Fig. 11.26(a))   With brakes in the 'off' position, both primary and secondary pistons are pushed outwards by the return springs to their respective stops. Under these conditions fluid is permitted to circulate between the pressure chambers and the respective piston recesses via the small compensating port, reservoir supply outlet and the large feed ports for both primary and secondary brake circuits.

Brakes applied (Fig. 11.26(b))   When the foot pedal is depressed, the primary piston moves inwards and, at the same time, compresses both the intermediate and secondary return springs so that the secondary piston is pushed towards the cylinder's blanked end.

Initial movement of both pistons causes their respective recuperating seals to sweep past each compensating port. Fluid is trapped and, with increased piston travel, is pressurized in both the primary and secondary chambers and their pipe line circuits, supplying the front and rear brake cylinders. During the braking phase, fluid from the reservoir gravitates and fills both of the annular piston recesses.

Brakes released (Fig. 11.26(a))   When the foot pedal effort is removed, the return springs rapidly expand, pushing both pistons outwards. The speed at which the swept volume of the pressure chambers increases will be greater than the rate at which the fluid returns from the brake cylinders and pipe lines. Therefore a vacuum is created within both primary and secondary pressure chambers.

As a result of the vacuum created, each recuperating seal momentarily collapses. Fluid from the annular piston recess is then able to flow through the horizontal holes in the piston head, around the inwardly distorted recuperating seals and into their respective pressure chambers. This extra fluid

478
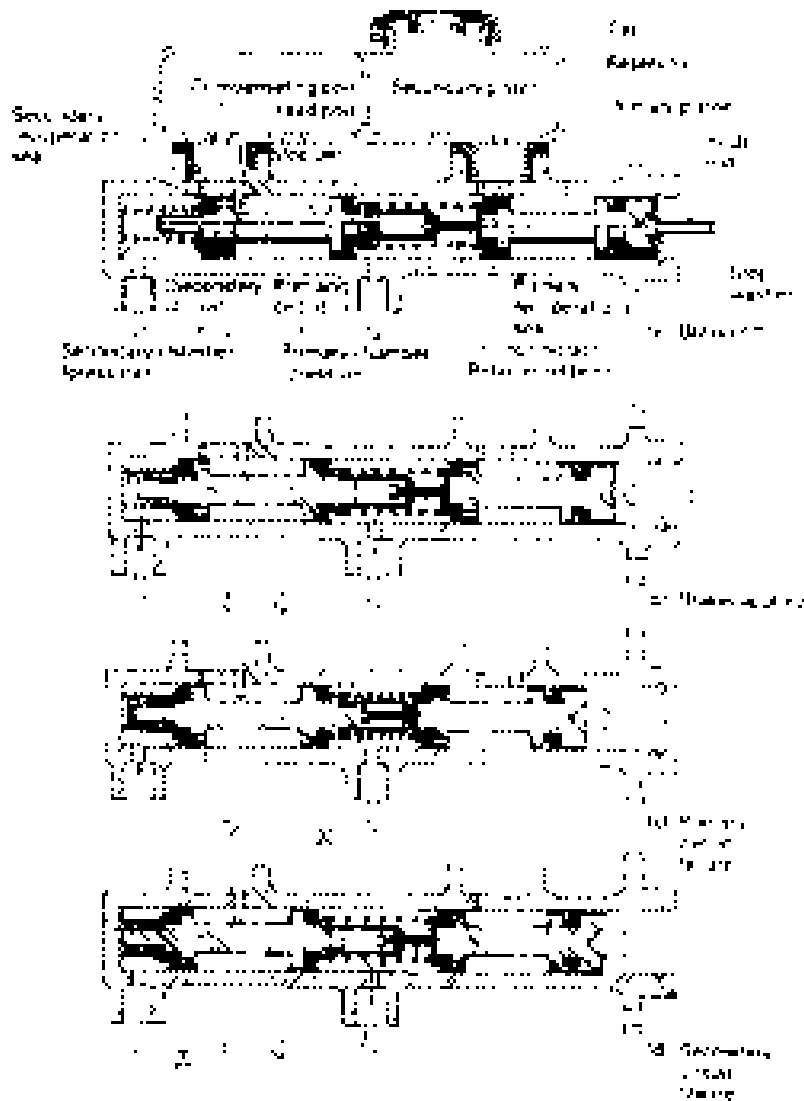
Fig. 11.26 (a–d) Tandem master cylinder

entering both pressure chambers compensates for any fluid loss within the brake pipe line circuits or for excessive shoe to drum clearance. But, if too much fluid is induced in the chambers, some of this fluid will pass back to the reservoir via the compensating ports after the return springs have fully retracted both pistons.

ailure in the primary circuit (Fig. 11.26(c)) Should a failure (leakage) occur in the primary circuit, there will be no hydraulic pressure generated in the primary chamber. When the brake pedal is depressed, the push rod and primary piston will move inwards until the primary piston abuts the secondary spring retainer. Further pedal effort will move the secondary piston recuperating seal beyond the compensating port, thereby pressurizing the fluid in the secondary chamber and subsequently transmitting this pressure to the secondary circuit pipe line and the respective brake cylinders.

ailure in the secondary circuit (Fig. 11.26(d)) If there is a failure (leakage) in the secondary circuit, the push rod will move the primary piston inwards until its recuperating seal sweeps past the compensating port, thus trapping the existing fluid

479

in the primary chamber. Further pedal effort increases the pressure in the primary chamber and at the same time both pistons, separated by the primary chamber fluid, move inwards unopposed until the secondary piston end stop contacts the cylinder's blanked end. Any more increase in braking effort raises the primary chamber pressure, which accordingly pressurizes the primary circuit brake cylinders.

The consequence of a failure in the primary or secondary brake circuit is that the effective push rod travel increases and a greater pedal effort will need to be applied for a given vehicle retardation compared to a braking system which has both primary and secondary circuits operating.

11.5.5 ecanindus (roll) pin type tandem master cylinder incorporating a pressure differential warning actuator (Fig. 11.27(a–d))

The tandem or split master cylinder is designed to provide two separate hydraulic cylinder pressure chambers operated by a single input push rod. Each cylinder chamber is able to generate its own fluid pressure which is delivered to two independent brake pipe line circuits. Thus if one hydraulic circuit malfunctions, the other one is unaffected and will provide braking to the wheel cylinders forming part of its system.

Operation of tandem master cylinder

Brakes off (Fig. 11.27(a))   With the push rod fully withdrawn, both primary and secondary pistons are forced outwards by the return springs. This outward movement continues until the central poppet valve stems contact their respective Mecanindus (roll) pins. With further withdrawal the poppet valves start opening until the front end of each elongated slot also contacts their respective roll pins, at which point the valves are fully open. With both valves open, fluid is free to flow between the primary and secondary chambers and their respective reservoirs via the elongated slot and vertical passage in the roll pins.

Brakes applied (Fig. 11.27(b))   When the brake pedal is applied, the push rod and the primary return spring pushes both pistons towards the cylinder's blank end. Immediately both recuperating poppet valves are able to snap closed. The fluid trapped in both primary and secondary chambers is then squeezed, causing the pressure in the

primary and secondary pipe line circuits to rise and operate the brake cylinders.

Brakes released (Fig. 11.27(a))   Removing the foot from the brake pedal permits the return spring to push both pistons to their outermost position. The poppet valve stem instantly contacts their respective roll pins, causing both valves to open. Since the return springs rapidly push back their pistons, the volume increase in both the primary and secondary chambers exceeds the speed of the returning fluid from the much smaller pipe line bore, with the result that a depression is created in both chambers. Fluid from the reservoir flows via the elongated slot and open poppet valve into the primary and secondary chambers to compensate for any loss of fluid or excessive shoe to drum or pad to disc clearance. This method of transferring fluid from the reservoir to the pressure chamber is more dynamic than relying on the collapse and distortion of the rubber pressure seals as in the conventional master cylinder.

Within a very short time the depression disappears and fluid is allowed to flow freely to and fro from the pressure chambers to compensate for fluid losses or fluid expansion and contraction caused by large temperature changes.

11.5.6 Operation of the pressure differential warning actuator

As a warning to the driver that there is a fault in either the primary or secondary hydraulic braking circuits of a dual-line braking system, a pressure differential warning actuator is usually incorporated as an integral part of the master cylinder or it may be installed as a separate unit (Fig. 11.27).

The switch unit consists of a pair of opposing balance pistons spring loaded at either end so that they are normally centrally positioned. Mounted centrally and protruding at right angles into the cylinder is an electrical conducting prod, insulated from the housing with a terminal formed at its outer end. The terminal is connected to a dashboard warning light and the electrical circuit is completed by the earth return made by the master cylinder.

Operation (Fig. 11.27(b))   If, when braking, both hydraulic circuits operate correctly, the opposing fluid pressure imposed on the outer ends of the balance piston will maintain the pistons in their equilibrium central position.
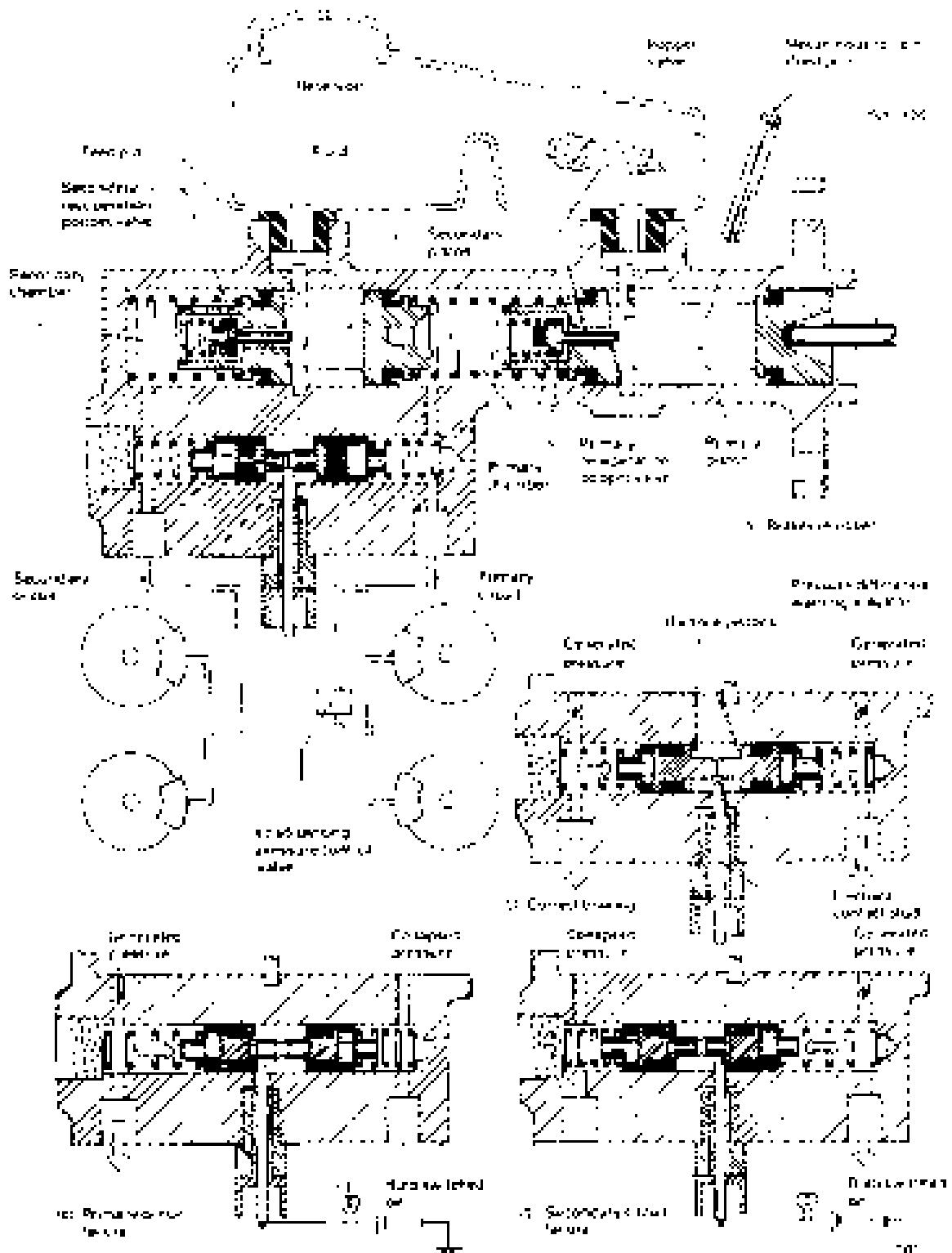
Fig. 11.27 (a–d) Tandem master cylinder with pressure differential warning actuator

Should one or the other of the dual circuits develop a pressure drop fault due to fluid leakage (Fig. 11.27(c and d)), then if the pressure difference of 10 bar or more exists between the two circuits, an imbalance of the fluid pressure applied against the outer ends of the pistons will force both pistons to move in the direction of the faulty circuit. The sideways movement of the pistons will cause the shoulder of the correctly operating circuit balance piston to contact the protruding prod, thus automatically completing the dashboard warning light electrical circuit, causing it to illuminate. Removing the brake pedal effort causes the fluid pressure in the effective circuit to collapse, thereby enabling the balance pistons to move back to their central positions. This interrupts the electrical circuit so that the warning light switches off.

11.6 Apportioned braking

11.6.1 Pressure limiting valve (Fig. 11.28)
The object of the pressure limiting valve is to interrupt the pressure rise of fluid transmitted to the rear wheel brakes above some predetermined value, so that the rear brakes will be contributing a decreasing proportion of the total braking with further increased pedal effort and master cylinder generated line pressure. By imposing a maximum brake line pressure to the rear brake cylinders, the rear wheels will be subjected to far less overbraking when the vehicle is heavily braked. It therefore reduces the tendency for rear wheel breakaway caused by wheels locking. Note that with this type of valve unit under severe slippery conditions the rear wheels are still subjected to lock-up.

Operation (Figs 11.28 and 11.29) Under light brake pedal application, fluid pressure from the master cylinder enters the valve inlet port and passes through the centre and around the outside of the plunger on its way to the outlet ports via the wasted region of the plunger (Fig. 11.28(a)).

When heavy brake applications are made (Fig. 11.28(b)), the rising fluid pressure acting on the large passage at the rear of the plunger displaces the plunger assembly. Instantly the full cross-sectional area equivalent to the reaction piston is exposed to hydraulic pressure, causing the plungers to move forward rapidly until the plunger end seal contacts the valve seat in the body of the valve unit. The valve closing pressure is known as the *cut off pressure* Under these conditions the predetermined line pressure in the rear pipe line will be maintained constant (Fig. 11.29), whereas the front brake pipe line pressure will continue to rise unrestricted, according to the master cylinder pressure generated by the depressed brake pedal.

11.6.2 Load sensing pressure limiting valve (Fig. 11.20)
To take into account the weight distribution between the front and rear wheels between an unladen and fully laden vehicle, a load sensing valve may be incorporated in the pipe line connecting the master cylinder to the rear wheel brakes. The function of the valve is to automatically separate the master cylinder to rear brake pipe line by closing a cut-off valve when the master cylinder's generated pressure reaches some predetermined maximum. This cut-off pressure will vary according to the weight imposed on the rear axle.
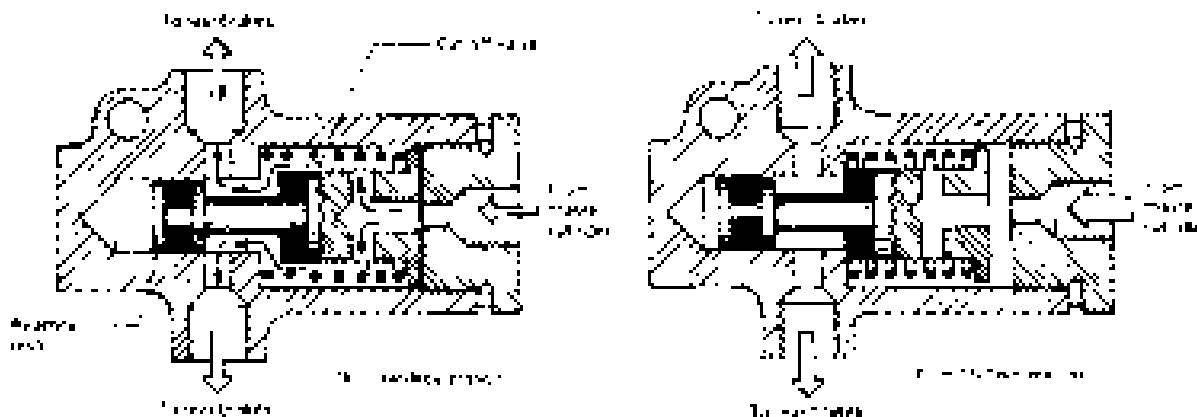
Fig. 11.28 (a and b) Pressure limiting valve
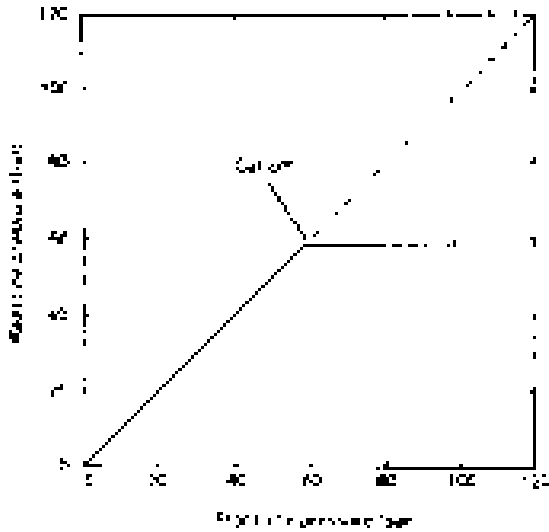
Fig. 11.29 Pressure limiting valve front to rear brake line characteristics

Operation (Figs 11.30 and 11.31) This valve device consists of a plunger supporting a rubber face valve which is kept open by the tension of a variable rate leaf spring. The inward thrust on the plunger keeping the end face valve open is determined by the leaf spring pre-tension controlled by the rear suspension's vertical deflection via the interconnecting spring and rod link. When the vehicle's rear suspension is unloaded, the leaf spring will be partially relaxed, but as the load on the rear axle increases, the link spring and rod pulls the leaf spring towards the valve causing it to

stiffen and increase the inward end thrust imposed on the plunger.

With a light brake pedal application, fluid pressure generated by the master cylinder enters the inlet port and passes around the wasted plunger on its way out to the rear brake pipe line (Fig. 11.30(a)).

If a heavy brake application is made (Fig. 11.30(b)), the rising fluid pressure from the master cylinder passes through the valve from the inlet to the outlet ports until the pressure creates a force at the end of the plunger (Force = Pressure × Area) which opposes the spring thrust, pushing back the plunger until the face valve closes. Any further fluid pressure rise will only be transmitted to the front brake pipe lines, whereas the sealed-off fluid pressure in the rear brake pipe lines remains approximately constant.

If the load on the rear axle alters, the vertical deflection height of the suspension will cause the leaf spring to stiffen or relax according to any axle load increase or decrease.

A change in leaf spring tension therefore alters the established pressure (Fig. 11.31) (at which point the cut-off valve closes) and the maximum attainable pressure trapped in the rear brake pipe lines.

11.6.3 *Load sensing progressive pressure limiting valve* (Fig. 11.32)

The load sensing progressive pressure limiting valve regulates the fluid pressure transmitted to the rear brake cylinders once the master cylinder's generated pressure has risen above some predetermined value corresponding to the weight carried on the rear axle.
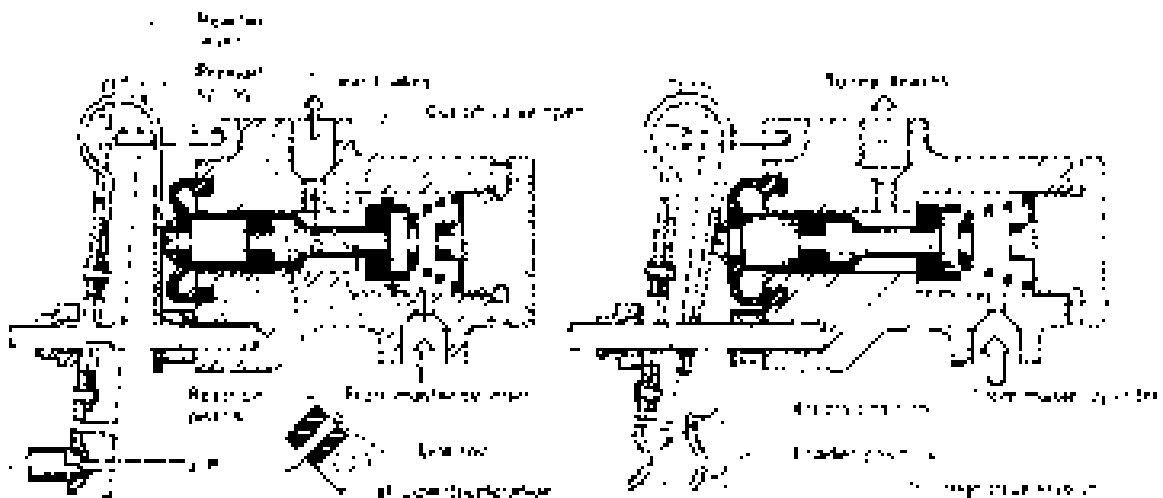
Fig. 11.30 (a and b) Load sensing pressure limiting valve
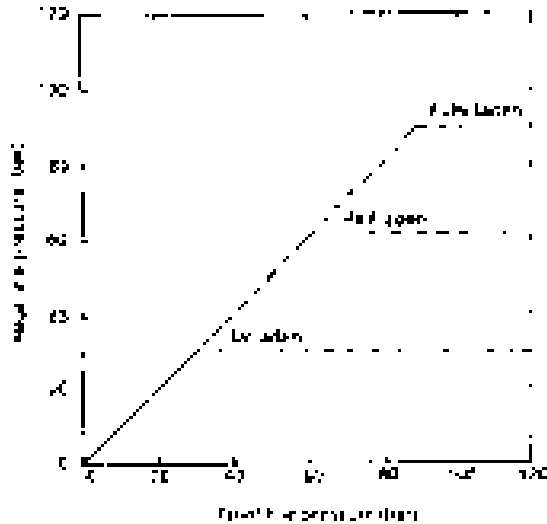
Fig. 11.31 Load sensing pressure limiting valve front to rear brake line characteristics

The reduced rate of pressure increase, in proportion to the pedal effort in the rear brake pipe line, provides a braking ability for both the front and rear brakes which approximately matches the load distribution imposed on the front and rear wheels, so that the tendency for the rear wheels to be either under or over braked is considerably reduced.

Operation (Fig. 11.32) When the foot pedal is applied lightly (Fig. 11.32(a)), pressure generated by the master cylinder will be transferred through the centre of the stepped reaction piston, between the cone and seat and to the rear brake pipe lines.

If the brake pedal is further depressed (Fig. 11.32(b)), increased fluid pressure acting on the large piston area produces an end force, which, when it exceeds the opposing link spring tension and fluid pressure acting on the annular piston face, causes the stepped piston to move outwards. This outward movement of the piston continues until the valve stem clears the cylinder's blanked end, thereby closing the valve. The valve closure is known as the *cut off point* since it isolates the rear brake pipe lines from the master cylinder delivery.

Further generation of master cylinder pressure exerted against the annular piston face produces an increase in force which moves the piston inwards, once again opening the valve. The hydraulic connection is re-established, allowing the rear brake pipe line fluid pressure to increase. However, the pressure exerted against the end face of the piston immediately becomes greater than the spring force and hydraulic force pushing on the annular piston face, and so the piston moves outwards, again closing the valve.

Every time the valve is opened with rising master cylinder pressure, the rear brake pipe line pressure increases in relation to the previous closing of the valve. Over a heavy braking pressure rise phase the piston oscillates around a position of balance, causing a succession of valve openings and closings. It subsequently produces a smaller pressure rise in the rear brake pipe line than with the directly connected front brake pipe lines.
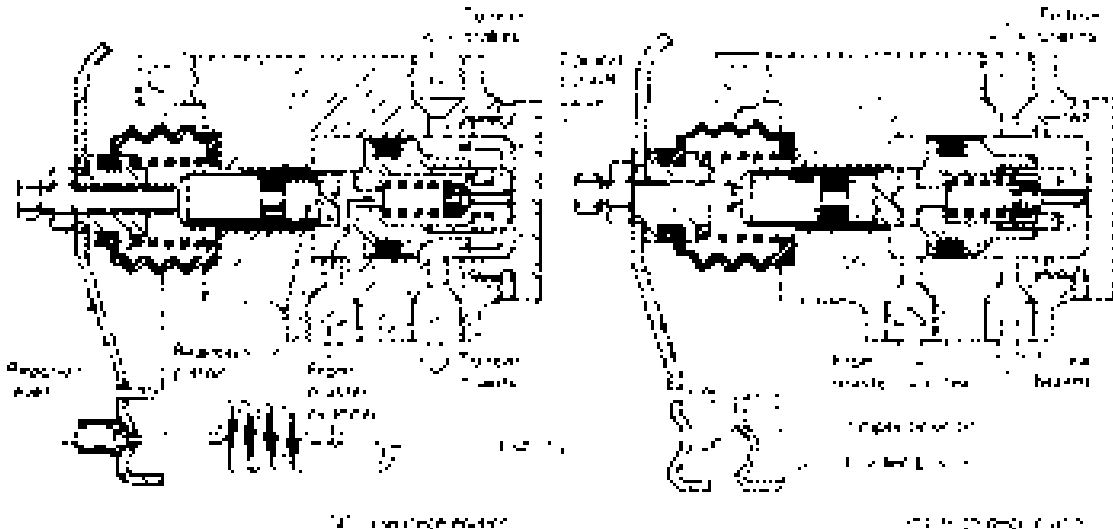
Fig. 11.32 (a and b) Load sensing progressive pressure limiting valve
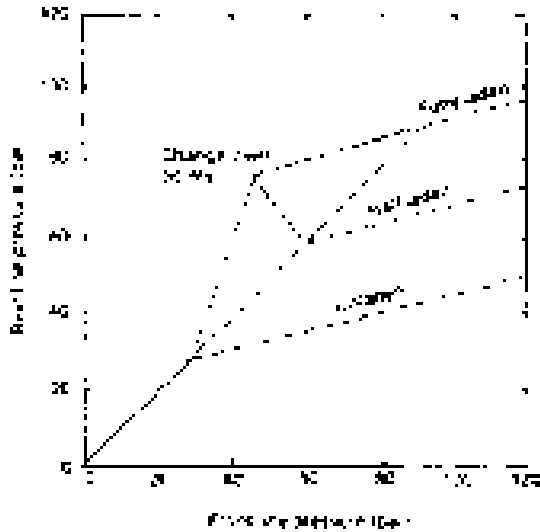
Fig. 11.33 Load sensing and progressive pressure limiting valve front to rear brake line characteristics

The ratio of the stepped piston face areas determines the degree of rear brake pipe line increase with respect to the front brake pipe lines (Fig. 11.33).

The cut-off or change point depends on the ten-sioning of the pre-setting spring which varies with the rear suspension deflection. The brake force distribution between the front and rear brakes is not only affected by the static laden condition, but even more so by the dynamic weight transference from the rear to the front axle.

11.6.4 *nertia pressure limiting valve* (Fig. 11.34)

The inertia pressure limiting valve is designed to restrict the hydraulic line pressure operating the rear wheel brakes when the deceleration of the vehicle exceeds about 0.3 g. In preventing a further rise in the rear brake line pressure, the unrestricted front brake lines will, according to the hydraulic pressure generated, increase their proportion of braking relative to the rear brakes.

Operation (Figs 11.34 and 11.35)   The operating principle of the inertia valve unit relies upon the inherent inertia of the heavy steel ball rolling up an inclined ramp when the retardation of the vehicle exceeds some predetermined amount (Fig. 11.34(b)). When this happens, the weight of the ball is removed from the stem of the disc valve, enabling the return spring fitted between the inlet port and valve shoulder to move the valve into the cut-off position.

At this point, the fluid trapped in the rear brake pipe line will remain constant (Fig. 11.35), but fluid flow between the master cylinder and front brakes is unrestricted and therefore will continue to rise with increased pedal force. As a result, the front brakes will contribute a much larger proportion of the total braking effort than the rear brakes.

When the vehicle has slowed down sufficiently or even stopped, the steel ball will gravitate to its lowest point, thereby pushing open the cut-off valve. Fluid is now free again to move from the master cylinder to the rear wheel brakes (Fig. 11.34(a)).
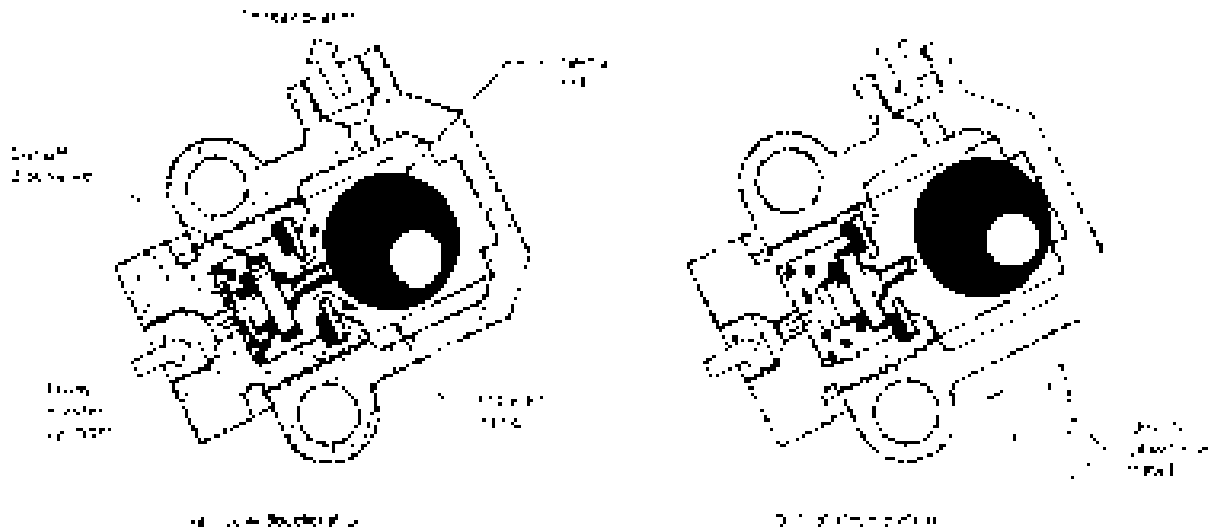
Fig. 11.34 (a and b)   Inertia pressure limiting valve

485
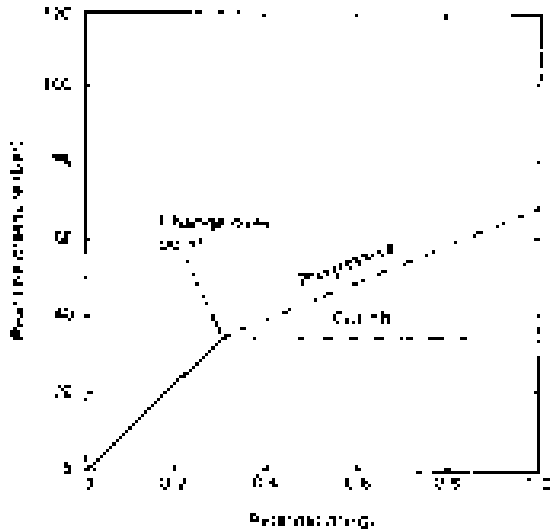
Fig. 11.35 Inertia pressure limiting valve and inertia progressive pressure limiting front to rear brake line

11.6.5 *nertia and progressive pressure limiting valve* (Fig. 11.36)

The inertia and progressive pressure limiting valve unit enables the braking power between the front and rear wheels to be adjusted to match the weight transference from the rear wheels to the front wheels in proportion to the vehicle's deceleration rate. This two stage valve unit allows equal fluid pressure to flow between the front and rear brakes for light braking, but above some predetermined deceleration of the vehicle, the direct pressure increase to the rear brakes stops. With moderate to heavy braking, the front brake line pressure will equal the master cylinder generated pressure. The rear brake line pressure will continue to increase but at a much slower rate compared to that of the front brakes.

Operation (Figs 11.35 and 11.36)   This inertia and progressive valve unit differs from the simple inertia pressure limiting valve because it incorporates a stepped piston (two piston dimeters) and the ball performs the task of the cut-off valve.

If the vehicle is lightly braked (Fig. 11.36(a)), fluid will flow freely from the master cylinder inlet port, through the dispersing diffuser, around the ball, along the piston central pin passage to the outlet port leading to the rear wheel brakes.

As the brake pedal force is increased (Fig. 11.36(b)), the vehicle's rate of retardation will cause the ball to continue to move forward by

rolling up the inclined ramp until it seals off the central piston passage. This state is known as the *point of cut off*.

Further foot pedal effort directly increases the front brake pipe line pressure and the pressure in the ball chamber. It does not immediately increase the rear brake pipe line pressure on the output of the valve.

Under these conditions, the trapped cut-off pressure in the rear brake lines reacts against the large piston cross-sectional area, whereas the small piston cross-sectional area on the ball chamber side of the piston is subjected to the master cylinder hydraulic pressure.

As the master cylinder's generated pressure rises with greater foot pedal effort, the input force produced on the small piston side (Input force = Master cylinder pressure × Small piston area) will increase until it exceeds the opposing output force produced on the large piston area (Output force = Rear brake line pressure × Large piston area).

A further rise in master cylinder pressure which will also be experienced in the ball chamber pushes the stepped piston backwards. Again, the rear brake line pressure will start to rise (Fig. 11.35), but at a reduced rate determined by the ratio of the small piston area to large piston area, i.e. A_S/A_L. For example, if the piston area ratio is 2:1, then the rear brake line pressure increase will be half the input master cylinder pressure rise.

i e $\quad P_o = \dfrac{P_i}{2}$

where $\quad P_i$ = input pressure
$\qquad P_o$ = output pressure

To safeguard the rear brake pipe lines, should the piston reach its full extent of its travel, the centre pin will stand out from the piston. Consequently the ball is dislodged from its seat so that fluid pressure is permitted to pass to the rear brake pipe lines.

If there are two separate rear brake pipe line circuits, each line will have its own rear brake pressure reducing valve.

11.7 Antilocking brake system (ABS)

With conventional brake systems one of the road wheels will always tend to lock sooner than the other, due to the continuously varying tyre to road grip conditions for all the road wheels. To prevent individual wheels locking when braking, the pedal should not be steadily applied but it

486
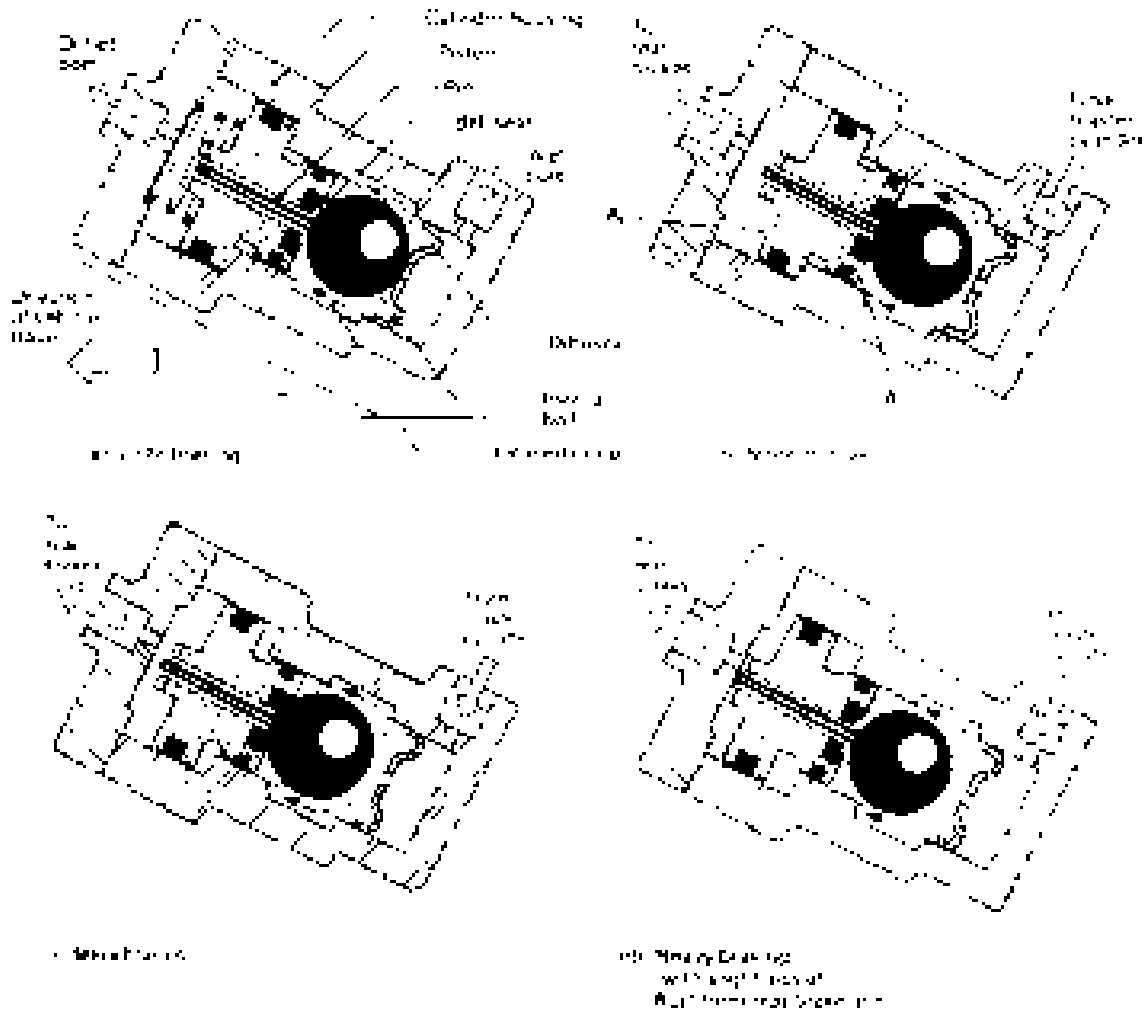
Fig. 11.36 (a–d) Inertia and progressive pressure limiting valve

should take the form of a series of impulses caused by rapidly depressing and releasing the pedal. This technique of pumping and releasing the brake pedal on slippery roads is not acquired by every driver, and in any case is subjected to human error in anticipating the pattern of brake pedal application to suit the road conditions. An antilock brake system does not rely on the skill of the driver to control wheel lock, instead it senses individual wheel slippage and automatically superimposes a brake pipe line pressure rise and fall which counteracts any wheel skid tendency and at the same time provides the necessary line pressure to retard the vehicle effectively.

When no slip takes place between the wheel and road surface, the wheel's circumference (periphery) speed and the vehicle's speed are equal. If, when the brakes are applied, the wheel circumference speed is less than the vehicle speed, the speed difference is the slip between the tyre and road surface. When the relative speeds are the same the wheels are in a state of pure rolling. When the wheels stop rotating with the vehicle continuing to move forward the slip is 100%, that is, the wheel has locked.

To attain optimum brake retardation of the vehicle, a small amount of tyre to ground slip is necessary to provide the greatest tyre tread to road surface interaction. For peak longitudinal braking force an approximately 15% wheel slip is necessary (Fig. 11.37), whereas steerability when braking depends upon a maximum sideways tyre to ground resistance which is achieved only with the

487
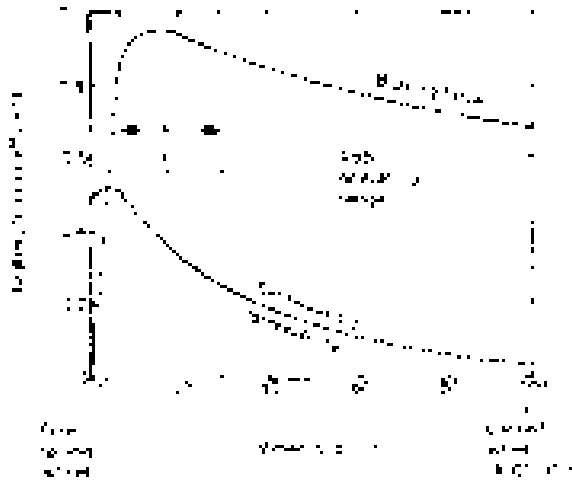
Fig. 11.37 Relationship of braking force coefficient and wheel slip

minimum of slip (Fig. 11.37). Thus there is conflict between an increasing braking force and a decreasing sideways resistance as the percentage of wheel slip rises initially. As a compromise, most anti-skid systems are designed to operate within an 8–30% wheel slip range.

11.7.1 Hydro-mechanical antilock brake system (ABS) suitable for cars (SCS Lucas Girling)
(Figs 11.38 and 11.39)

This hydro-mechanical antilock braking system has two modular units, each consisting of an integrated flywheel decelerating sensor, cam operated piston type pump and the brake pressure modulator itself (Fig. 11.38). Each modulator controls the adjacent wheel brake and the diagonally opposite rear wheel via an apportioning valve. The modular flywheel sensor is driven by a toothed belt at 2.8 times the wheel speed. The flywheel sensor deter-

Fig. 11.38 Stop control braking system (SCS) layout

mines when the front wheel is approaching a predetermined deceleration. In response to this the modulator reduces the pressure in the respective brake circuits. When the wheel speeds up again, the pump raises that pressure in order to bring the braking force back to a maximum level. This sequence of pressure reduction and build-up can be up to five times a second to avoid the wheel locking and also to provide the necessary deceleration of the car.

Braking as normal (Fig. 11.39(a)) Under normal braking conditions, the master cylinder fluid output is conveyed to the wheel brakes through the open cut-off valve. The dump valve is closed and the pump piston is held out of engagement from the rotating eccentric cam by the return spring.

Brake pressure reducing (Fig. 11.39(b)) When the deceleration of the front wheel, and therefore the drive shaft, exceeds a predetermined maximum (the wheels begin to lock), the flywheel overruns the drive shaft due to its inertia. The clutch balls then roll up their respective ramps, forcing the flywheel to slide inwards and causing the dump valve lever to tilt and open the dump valve. The fluid pressure above the deboost piston drops immediately. The much higher brake line pressure underneath the deboost piston and the pump piston forces the pump piston against its cam and raises the deboost piston. Fluid above both pistons is displaced back to the reservoir via the dump valve. The effect of the deboost piston rising is to close the ball cut-off valve so that the master cylinder pipe line fluid output and the wheel cylinder pipe line input become isolated from each other. As a result, the sealed chamber space below the deboost piston is enlarged, causing a rapid reduction in the fluid pressure delivered to the wheel cylinders and preventing the wheels connected to this brake circuit locking.

Brake pressure increasing (Fig. 11.39(c)) The pressure reduction resulting from the previous phase releases the brakes and allows the wheel to accelerate to the speed of the still decelerating flywheel. When the drive shaft and the flywheel are at roughly equal speeds, the clutch balls roll down their respective ramps, enabling the dump valve lever return spring to slide the flywheel over. The dump valve lever then pivots and closes the needle-type dump valve. The flywheel is again coupled to

488
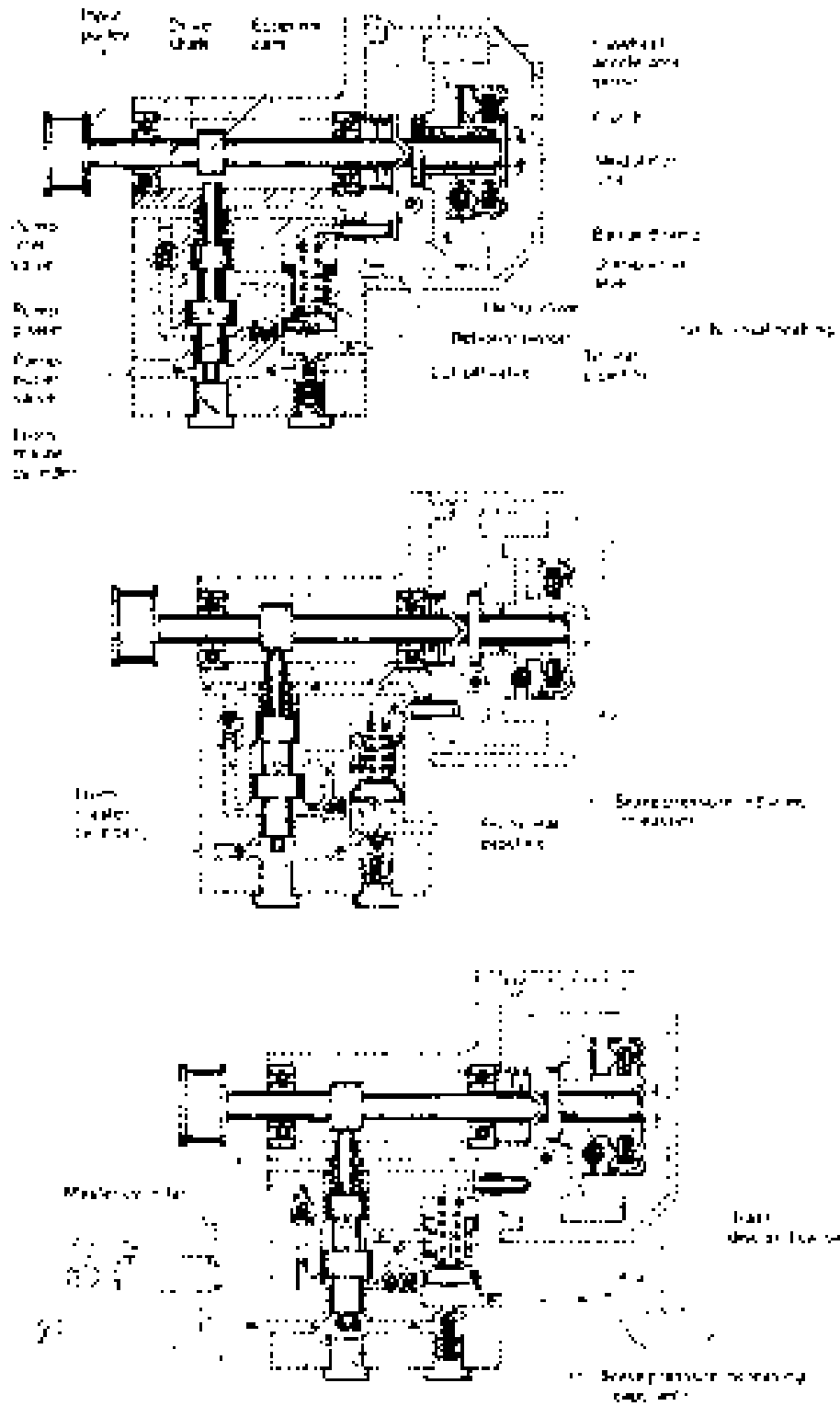
Fig. 11.39 (a–c) Antilock braking system (ABS) for front wheel drive

489

the drive shaft so that its speed rises with the drive shaft. At the same time the pump piston commences to build up pressure above the deboost piston by the action of the pump inlet and outlet valves. The output pressure generated by the pump pushes the deboost piston downward and, because the space underneath the deboost piston forms part of the brake pipe line circuit leading to the wheel cylinders, the total fluid volume is reduced. The brake pipe line pressure will be restored in steps due to the pump action until the downward movement of the deboost piston stem once again opens the cut-off valve. The pump piston then disengages and thereafter further pressure rise in the brake pipe line will be provided by the master cylinder in the normal way.

11.7.2 Hydraulic-electric antilock brake system (ABS) suitable for cars (Bosch) (Figs 11.40 and 11.41)

Speed sensor and e citor (Fig. 11.40)   The speed sensor uses the variable reluctance magnetic sensing principle, whereby a cylindrical permanent magnetic core with a coil wire wound around it, mounted on the stationary hub carrier, axle casing or back plate, produces a magnetic field (flux) which overlaps the rotating excitor ring. The excitor may be of the tooth ring or rib-slot ring type attached to the rotating wheel hub or drive shaft. A number of teeth or slots are arranged radially which, with the speed of rotation of the road wheel, determine the frequency of the signal transmitted to the electronic-control unit. As the wheel and excitor revolve, the teeth and gaps or ribs and slots of the excitor pass through the magnetic field of the sensor. The coil wrapped around the mag-

netic cone senses the changing intensity of the magnetic field as the teeth or ribs pass through the flux lines and so an alternating voltage is induced in the coil, whose frequency is proportional to the speed of the rotating wheel. The voltage is transmitted to the control unit whenever the road wheels are rotating, regardless of whether the brakes are applied.

The road wheel speed measured by the speed sensor provides the wheel deceleration and wheel acceleration signals for the electronic-control unit. The merging and processing of the individual wheel speed sensor signals by the control unit provide a single reference speed which is roughly the vehicle speed. A comparison of any individual wheel speed with the reference speed supplies the wheel to road slip (wheel tending to lock) signal.

lectronic-control unit (Fig. 11.41(a))   The function of the electronic-control unit is to receive, amplify, process, compute and energize the individual solenoid control valves. That is, to evaluate the minimum wheel deceleration and maximum wheel acceleration for optimum braking and accordingly supply the energizing current to the individual solenoid control valves so they can regulate the necessary wheel cylinder pipe line pressures.

Hydro/electric modulator (Fig. 11.41(a))   This unit combines the solenoid control valves; one for each wheel, an accumulator for each of the dual-brake circuits and a twin cylinder return flow pump driven from an electric motor. The solenoid valve switches half or fully on and off through the control unit's solid-state circuits, causing the master cylinder to wheel cylinder fluid supply to be interrupted many times per second. The reduced pressure accumulator rapidly depressurizes the wheel cylinder pipe line fluid when the solenoid valve opens the return passage, due to the diaphragm chamber space instantly enlarging to absorb the outflow of fluid. The return flow pump, with its inlet and outlet ball valves, transfers fluid under pressure from the reducer accumulator to the master cylinder output leading to the brake cylinders. By these means, the wheel cylinder fluid pressure is matched to the optimum braking severity relative to the condition of the road surface.

In the following description of the anti-skid system operating, only one wheel is considered for simplicity.
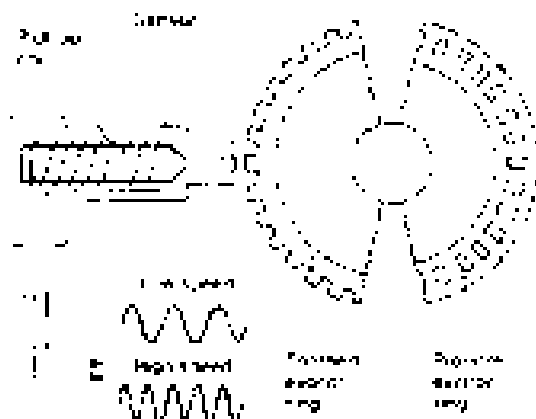
Fig. 11.40   Magnetic speed sensor and excitor
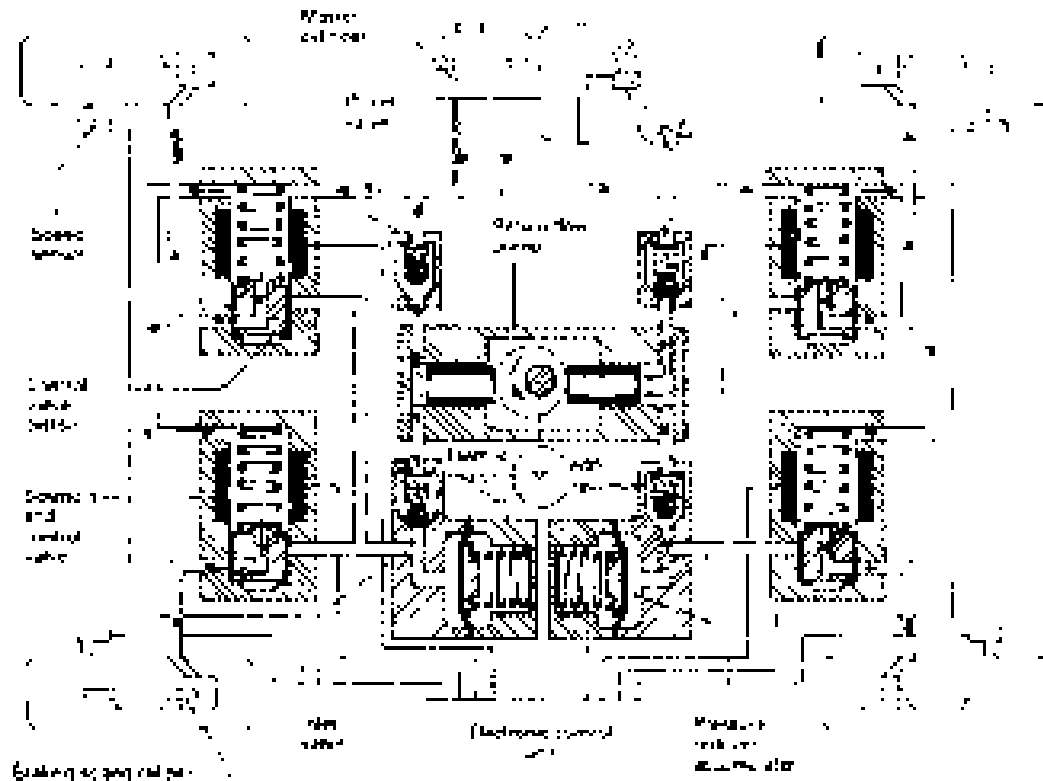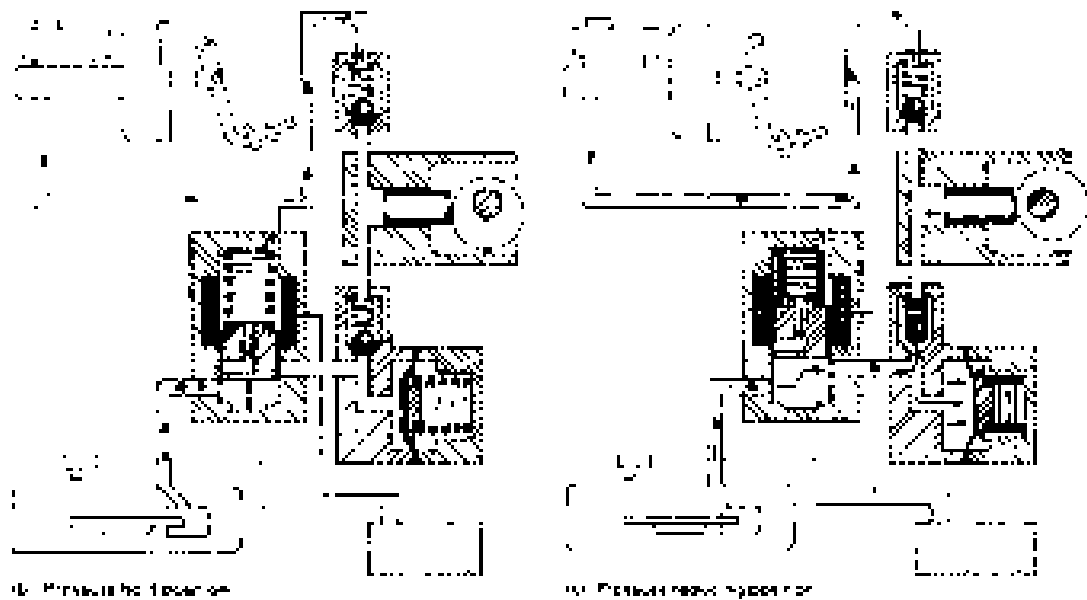
Fig. 11.41 (a–c)   Antilocking brake system (ABS) for cars

491

ormal braking conditions (Fig. 11.41(a)) Under normal braking conditions, the solenoid is disengaged and the armature valve is held in its lowest position by the return spring. When the brakes are applied, fluid flows unrestricted from the master cylinder to the wheel cylinder via the solenoid piston armature type valve central passage. This continues until the required pressure build-up against the caliper piston produces the desired retardation to the vehicle.

Pressure hold (Fig. 11.41(b)) When the wheel deceleration approaches some predetermined value, the speed sensor signals to the computer control unit the danger of the wheel locking. The control unit immediately responds by passing a small electric current to the appropriate solenoid valve. Accordingly, the solenoid coil is partially energized. This raises the armature valve until it blocks the flow of fluid passing from the master cylinder to the wheel cylinder pipe line. The fluid pressure in the pipe line is now held constant (Fig. 11.42).

Pressure reducing (Fig. 11.41(c)) Should the wheel sensor still signal an abnormally rapid speed reduction likely to cause the wheel to lock, the control unit increases the supply of current to the solenoid coil, causing the armature valve to lift

still further to a position where it uncovers the return flow passage. The 'hold' line pressure collapses instantly because the highly pressurized fluid is able to escape into the pressure reducer accumulator. At the same time as the accumulator is being charged, surplus fluid is drawn from the accumulator into the return flow pump via the inlet valve whence it is discharged back into the appropriate pressurized master cylinder output pipe line. Consequently, the reduction in pressure (Fig. 11.42) permits the wheel to accelerate once again and re-establish its grip with the road surface. During the time fluid is pumped back into the master cylinder output pipe line, a light pressure pulsation will be experienced on the foot pedal by the driver due to the cyclic discharge of the pump.

Pressure increasing (Fig. 11.41(a)) Once the wheel rotational movement has changed from a deceleration back to acceleration, the sensor signals to the control unit to switch off the solenoid valve current supply. The return spring instantly snaps the solenoid valve into its lowest position and once again the fluid passage between the master cylinder output pipe line and the wheel caliper cylinder pipe line is re-established, causing the brake to be re-applied (Fig. 11.42). The sensitivity and response time of the solenoid valve is such that the pulsating regulation takes place four to ten times per second.

11.7.3 Air/electric antilock brake system (ABS) suitable for commercial vehicles (ABCO)
(Figs 11.43 and 11.44)
The antilock brake system (ABS) consists of wheel sensors and exitors which detect the deceleration and an acceleration of individual wheels by generating alternating voltages the frequency of which are proportional to the wheel speed (Fig. 11.43(a)).

Sensors on each wheel (Fig. 11.40) continually measure the wheel speed during braking and this information is transmitted to an electronic (processor) control unit which senses when any wheel is about to lock. Signals are rapidly relayed to solenoid control valve units which quickly adjust the brake air line pressure so that the wheels are braked in the optimum slip range.

Each wheel is controlled according to the grip available between its tyre and the road. By these means, the vehicle is brought to a halt in the shortest time without losing vehicle stability and steerability.

Fig. 11.42 Typical antilock brake system (ABS) pressure, wheel and vehicle speed characteristics with respect to time
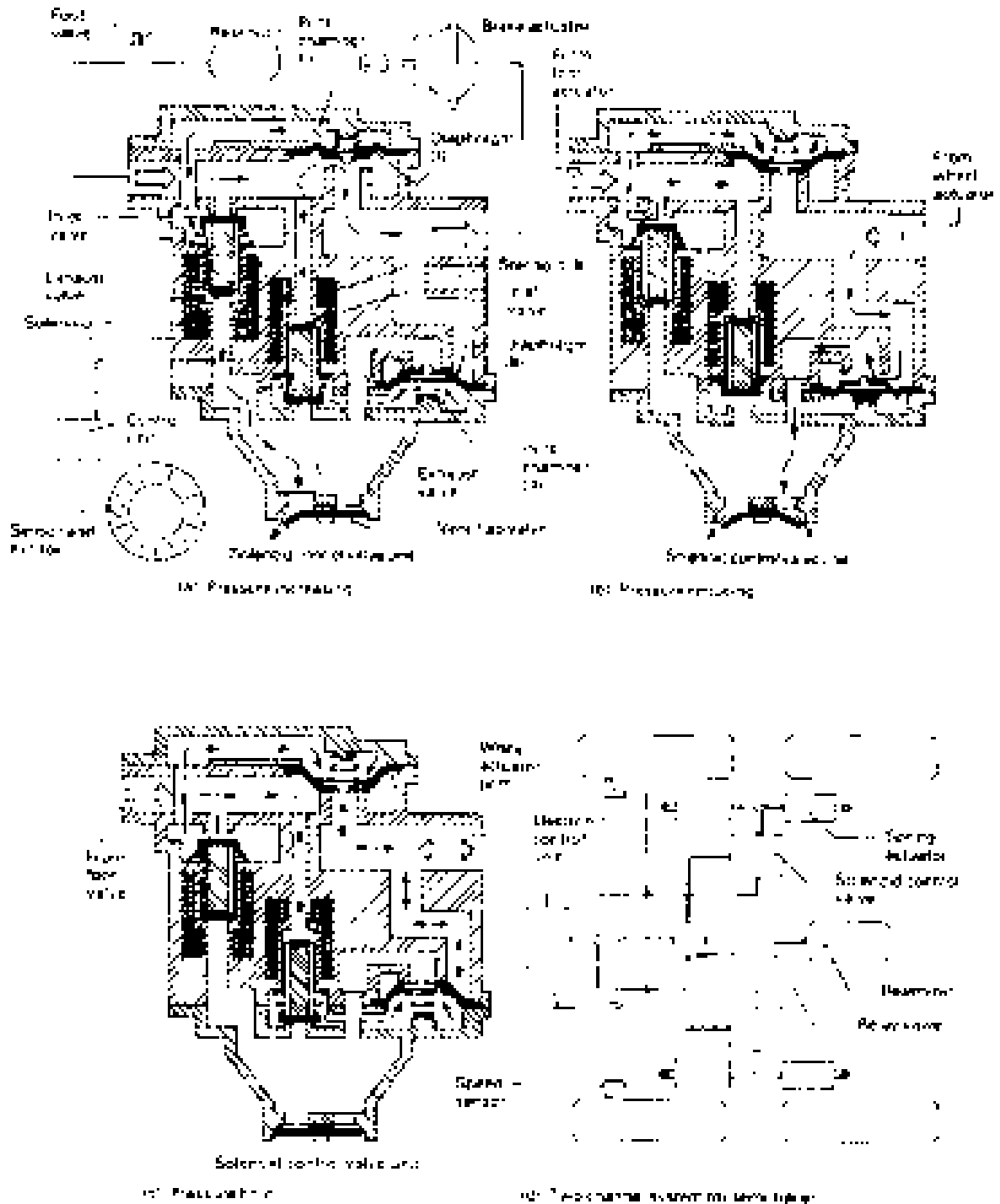
Fig. 11.43 (a–d) Antilock brake system for commercial vehicles (ABS)

Pressure increasing (Fig. 11.43(a)) When the foot pedal is depressed, initially both solenoids are switched off so that their armatures are moved to their outermost position by the return springs. Consequently the first solenoid's inlet valve (I) is closed and its exhaust valve (I) is open whereas the second solenoid valve's inlet valve (II) is open and its exhaust valve (II) is closed.

Under these conditions, pilot chamber (I) is exhausted of compressed air so that air delivered from the foot valve enters the solenoid control valve unit inlet port and pushes open diaphragm (I) outlet passage, enabling compressed air to be supplied to the wheel brake actuator. At the same time pilot chamber (II) is filled with compressed air so that diaphragm (II) closes off the exhaust passage leading to the atmosphere. As a result, the foot pedal depression controls the rising air pressure (Fig. 11.44) delivered from the foot valve to the wheel actuator via the solenoid control valve unit.

Pressure reducing (Fig. 11.43(b)) As soon as wheel deceleration or wheel slip threshold values are exceeded, the sensor transmits this information to the electronic-control unit which signals to the solenoid valve unit to reduce the wheel actuator pipe line air pressure.

Both solenoids are energized. This opens inlet valve (I) whilst inlet valve (II) is closed and exhaust valve (II) is opened. The open inlet valve (I) allows air to enter and pressurize pilot chamber (I) so that diaphragm (I) closes the outlet passage, thus preventing any more air from the foot valve passing through to the outlet passage port.

At the same time, solenoid (II) closes inlet valve (II) and opens exhaust valve (II). This exhausts air from pilot chamber (II), permitting compressed air from the wheel actuator to push open diaphragm (II) outlet exhaust passage, causing the air pressure in the actuator pipe line to reduce quickly (Fig. 11.44).

Pressure hold (Fig. 11.43(c)) When the road wheel acceleration reaches some predetermined value, the sensor relays this information to the electronic-control unit, which in turn signals the solenoid control valve unit to hold the remaining pipe line actuator pressure.

Solenoid (I) remains energized but solenoid (II) is de-energized. Therefore solenoid (I) inlet valve (I) and exhaust valve (I) remain open and closed respectively. Inlet valve (II) allows compressed air into pilot chamber (I) so that diaphragm (I) closes the outlet passage leading to the wheel actuator pipe line.

Conversely, solenoid (II) is now de-energized causing its return spring to move the armature so that the inlet valve (II) opens and exhaust valve (II) closes. Compressed air from the foot valve now flows through the open inlet valve (II) along the passage leading to the underside of diaphragm (II), thus keeping the outlet exhaust passage closed. Compressed air at constant pressure (Fig. 11.44) is now trapped between both closed diaphragm outlet passages and the wheel actuator pipe line. This pipe line pressure is maintained until the sensor signals that the wheel is accelerating above its threshold, at which point the electronic-control unit signals the solenoid control valve to switch to its rising pressure mode.

11.8 Brake servos

11.8.1 Operating principle of a vacuum servo
(Fig. 11.45)

The demand for a reduction in brake pedal effort and movement, without losing any of the sensitivity and response to the effective braking of cars and vans, has led to the adoption of vacuum servo assisted units as part of the braking system for most light vehicles. These units convert the induction manifold vacuum energy into mechanical energy to assist in pressurizing the brake fluid on the output side of the master cylinder.

A direct acting vacuum servo consists of two chambers separated by a rolling diaphragm and power piston (Fig. 11.45). The power piston is coupled to the master cylinder outer primary piston by a power push rod. The foot pedal is linked through a pedal push rod indirectly to the power piston via a vacuum-air reaction control valve.
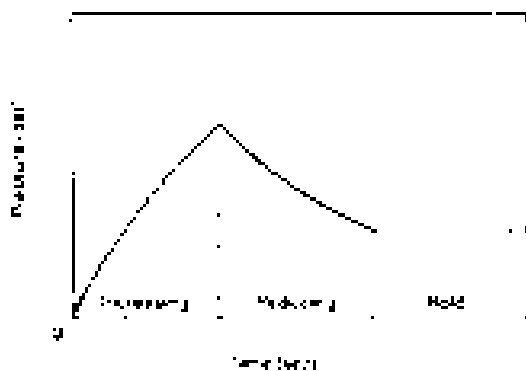
Fig. 11.44 Air/electric antilock brake system (ABS) pressure/time characteristics
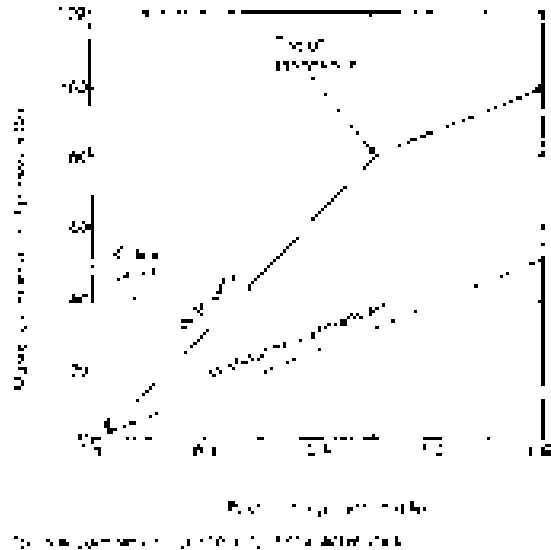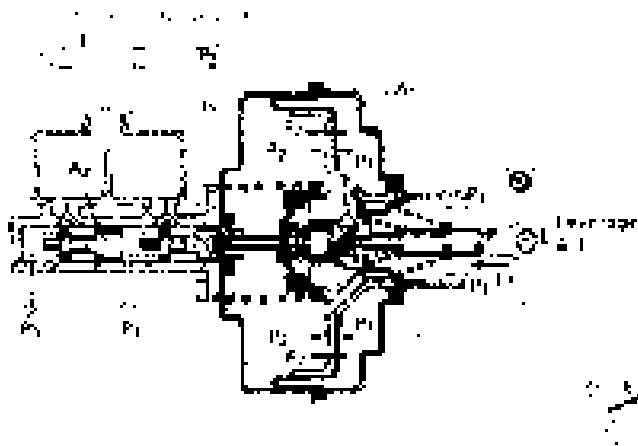
Fig. 11.45 (a and b) Operating principle and characteristics of a vacuum servo

When the brakes are in the 'off' position, both sides of the power piston assembly are subjected to induction manifold pressure. When the brakes are applied, the vacuum in the front chamber remains undisturbed, whilst the vacuum in the rear chamber is replaced by atmospheric air closing the vacuum supply passage, followed by the opening of the air inlet passage to the rear chamber. The resulting difference of pressure across the power piston causes it to move towards the master cylinder, so that the thrust imposed on both the primary and secondary pistons in the master cylinder generates fluid pressure for both brake lines.

The operating principle of the vacuum servo is best illustrated by the following calculation:

Example (Fig. 11.45(a)) A direct acting vacuum servo booster has a 200 mm diameter power piston suspended on both sides by the induction manifold vacuum (depression), amounting to a gauge reading of 456 mm Hg, that is 0.6 bar below atmospheric pressure.
(*Note* 1 bar = 760 mm Hg = 100 N/m^2).

The foot pedal leverage ratio is 4:1 and the master cylinder has 18 mm diameter.

Determine the following when a pedal effort of 300 N is applied and the rear power piston chamber which was occupied with manifold vacuum is now replaced by atmospheric air (Fig. 11.45(a)).

a) The push rod thrust and generated primary and secondary hydraulic brake line pressures due only to the foot pedal effort.

b) The power push rod thrust and the generated fluid pressures in the pipe lines due only to the vacuum servo action.

c) The total pedal push rod and power piston thrust and the corresponding generated fluid pressure in the pipe lines when both foot pedal and servo action are simultaneously applied to the master cylinder.

Let F = foot pedal effort (N)
F_1 = pedal push rod thrust (N)
F_2 = power piston thrust (N)
P_1 = pressure in the rear chamber (kN/m^2)
P_2 = manifold pressure (kN/m^2)
P_3 = fluid generated pressure (kN/m^2)
A_1 = cross-sectional area of power piston (m^2)
A_2 = cross-sectional area of master cylinder bore (m^2)

a) Pedal push rod thrust $F_1 = F \times 4$

$$= 300 \times 4$$

$$= 1200 \text{ N or } 1.2 \text{ kN}$$

Master cylinder fluid pressure $P_3 = \dfrac{F_1}{A_2}$

$$= \frac{1.2}{\frac{\pi}{4}(0.018)^2}$$

$$= 4715.7 \text{ kN/m}^2 \text{ or } 47.2 \text{ bar}$$

495

b) Power piston thrust $\quad F_2 = A_1(p_1 - p_2)$

$$= \frac{\pi}{4}(0.2)^2(100 - 40)$$

$$= 1.88 \,\text{kN}$$

Master cylinder fluid pressure p_3
$$= \frac{F_2}{A_2}$$

$$= \frac{1.88}{\frac{\pi}{4}(0.018)^2}$$

$$= 7387.93 \,\text{kN/m}^2 \text{ or } 73.9 \,\text{bar}$$

c) Total power piston and pedal push rod thrust
$$= F_1 + F_2$$

$$= 1.2 + 1.88$$

$$= 3.08 \,\text{kN}$$

Total master cylinder fluid pressure p_3
$$= \frac{F_3}{A_2}$$

$$= \frac{3.08}{\frac{\pi}{4}(0.018)^2}$$

$$= 12103.635 \,\text{kN/m}^2 \text{ or } 121.04 \,\text{bar}$$

11.8.2 Direct acting suspended vacuum-assisted brake servo unit (Fig. 11.46(a, b and c))

Brake pedal effort can be reduced by increasing the leverage ratio of the pedal and master cylinder to wheel cylinder piston sizes, but this is at the expense of lengthening the brake pedal travel, which unfortunately extends the brake application time. The vacuum servo booster provides assistance to the brake pedal effort, enabling the ratio of master cylinder to wheel cylinder piston areas to be reduced. Consequently, the brake pedal push rod effective stroke can be reduced in conjunction with a reduction in input foot effort for a given rate of vehicle deceleration.

Operation

Brakes off (Fig. 11.46(a)) With the foot pedal fully released, the large return spring in the vacuum chamber forces the rolling diaphragm and power piston towards and against the air/vac chamber stepped steel pressing.

When the engine is running, the vacuum or negative pressure (below atmospheric pressure) from the induction manifold draws the non-return valve away from its seat, thereby subjecting the whole vacuum chamber to a similar negative pressure to that existing in the manifold.

When the brake pedal is fully released, the outer spring surrounding the push rod pulls it and the relay piston back against the valve retaining plate. The inlet valve formed on the end of the relay piston closes against the vac/air diaphragm face and at the same time pushes the vac/air diaphragm away from the vacuum valve. Negative pressure from the vacuum chamber therefore passes through the inclined passage in the power piston around the seat of the open vacuum valve where it then occupies the existing space formed in the air/vac chamber to the rear of the rolling diaphragm. Hence with the air valve closed and the vacuum valve open, both sides of the power piston are suspended in vacuum.

Brakes applied (Fig. 11.46(b)) When the foot pedal is depressed the pedal push rod moves towards the diaphragm power piston, pushing the relay piston hard against the valve retaining plate. Initially the vac/air diaphragm closes against the vacuum valve's seat and with further inward push rod movement the relay piston inlet seat separates from the vac/air diaphragm face. The air/vac chamber is now cut off from the vacuum supply and atmospheric air is now free to pass through the air filter, situated between the relay piston inlet valve seat and diaphragm face, to replace the vacuum in the air/vac chamber. The difference in pressure between the low primary vacuum chamber and the high pressure air/vac chamber causes the power piston and power push rod to move forward against the master cylinder piston so the fluid pressure is generated in both brake circuits to actuate the front and rear brakes.

Brake held on (Fig. 11.46(c)) Holding the brake pedal depressed momentarily continues to move the power piston with the valve body forward under the influence of the greater air pressure in the air/vac chamber, until the rubber reaction pad is compressed by the shoulder of the power piston against the opposing reaction of the power push rod. As a result of squeezing the outer rubber rim of the reaction pad, the rubber distorts and extrudes towards the centre and backwards in the relay piston's bore. Subsequently, only the power piston and valve body move forward whilst the relay piston and pedal push rod remain approximately in the same position until the air valve seat closes against the vac/air diaphragm face. More

Fig. 11.46 (a and b) Vacuum-assisted brake servo unit

497

atmospheric air cannot now enter the air chamber so that there is no further increase in servo power assistance. In other words, the brakes are on hold. The reaction pad action therefore provides a progressive servo assistance in relation to the foot pedal effort which would not be possible if only a simple reaction spring were positioned between the reaction piston and the relay piston.

If a greater brake pedal effort is applied for a given hold position, then the relay piston will again move forward and compress the centre region of the reaction pad to open the air valve. The extra air permitted to enter the air/vac chamber therefore will further raise the servo assistance proportionally. The cycle of increasing or decreasing the degree of braking provides new states of hold which are progressive and correspond to the manual input effort.

Brakes released (Fig. 11.46(a)) Releasing the brake pedal allows the pedal push rod and relay piston to move outwards; first closing the air valve and secondly opening the vacuum valve. The existing air in the air/vac chamber will then be extracted to the vacuum chamber via the open vacuum valve, the power piston's inclined passage, and finally it is withdrawn to the induction manifold. As in the brakes 'off' position, both sides of the power piston are suspended in vacuum, thus preparing the servo unit for the next brake application.

acuum servo operating characteristics (Fig. 11.45(b)) The benefits of vacuum servo assistance are best shown in the input to output characteristic graphs (Fig. 11.45(b)). Here it can be seen that the output master cylinder line pressure increases directly in proportion to the pedal push rod effort for manual (unassisted) brake application. Similarly, with vacuum servo assistance the output line pressure rises, but at a much higher rate. Eventually the servo output reaches its maximum. Thereafter any further output pressure increase is obtained purely by direct manual pedal effort at a reduced rate. The extra boost provided by the vacuum servo in proportion to the input pedal effort may range from 1 :1 to 3:1 for direct acting type servos incorporated on cars and vans.

Servo assistance only begins after a small reaction force applied by the foot pedal closes the vacuum valve and opens the air inlet valve. This phase where the servo assistance deviates from the manual output is known as the *crack point*

11.8.3 Types of vacuum pumps
(Fig. 11.47(a, b and c))
For diesel engines which develop very little manifold depression, a separate vacuum pump driven from the engine is necessary to operate the brake servo. Vacuum pumps may be classified as *recipro cating diaphragm* or *piston* or *rotary vane* types.

In general, for high speed operation the vane type vacuum pump is preferred and for medium speeds the piston type pump is more durable than the diaphragm vacuum pump.

These pumps are capable of operating at depressions of up to 0.9 bar below atmospheric pressure. One major drawback is that they are continuously working and cannot normally be offloaded by interrupting the drive or by opening the vacuum chamber to the atmosphere.

Reciprocating diaphragm or piston type vacuum pump (Fig. 11.47(a and b)) These pumps operate very similarly to petrol and diesel engine fuel lift pumps.

When the camshaft rotates, the diaphragm or piston is displaced up and down, causing air to be drawn through the inlet valve on the downstroke and the same air to be pushed out on the upward stroke through the discharge valve.

Consequently, a depression is created within the enlarging diaphragm or piston chamber causing the brake servo chamber to become exhausted (drawn out) of air, thereby providing a pressure difference between the two sides of the brake servo which produces the servo power.

Lubrication is essential for plungers and pistons but the diaphragm is designed to operate dry.

Rotary vane type vacuum pump (rotary e hauster) (Fig. 11.47(c)) When the rotor revolves, the cell spaces formed between the drum blades on the inlet port side of the casing increase and the spaces between the blades on the discharge port side decrease, because of the eccentric mounting of the rotor drum in its casing.

As a result, a depression is created in the enlarging cell spaces on the inlet side, causing air to be exhausted (drawn out) directly from the brake vacuum servo chamber or from a separate vacuum reservoir. However on the discharge side the cells are reducing in volume so that a positive pressure is produced.

The drive shaft drum and vanes require lubricating at pressure or by gravity or suction from the
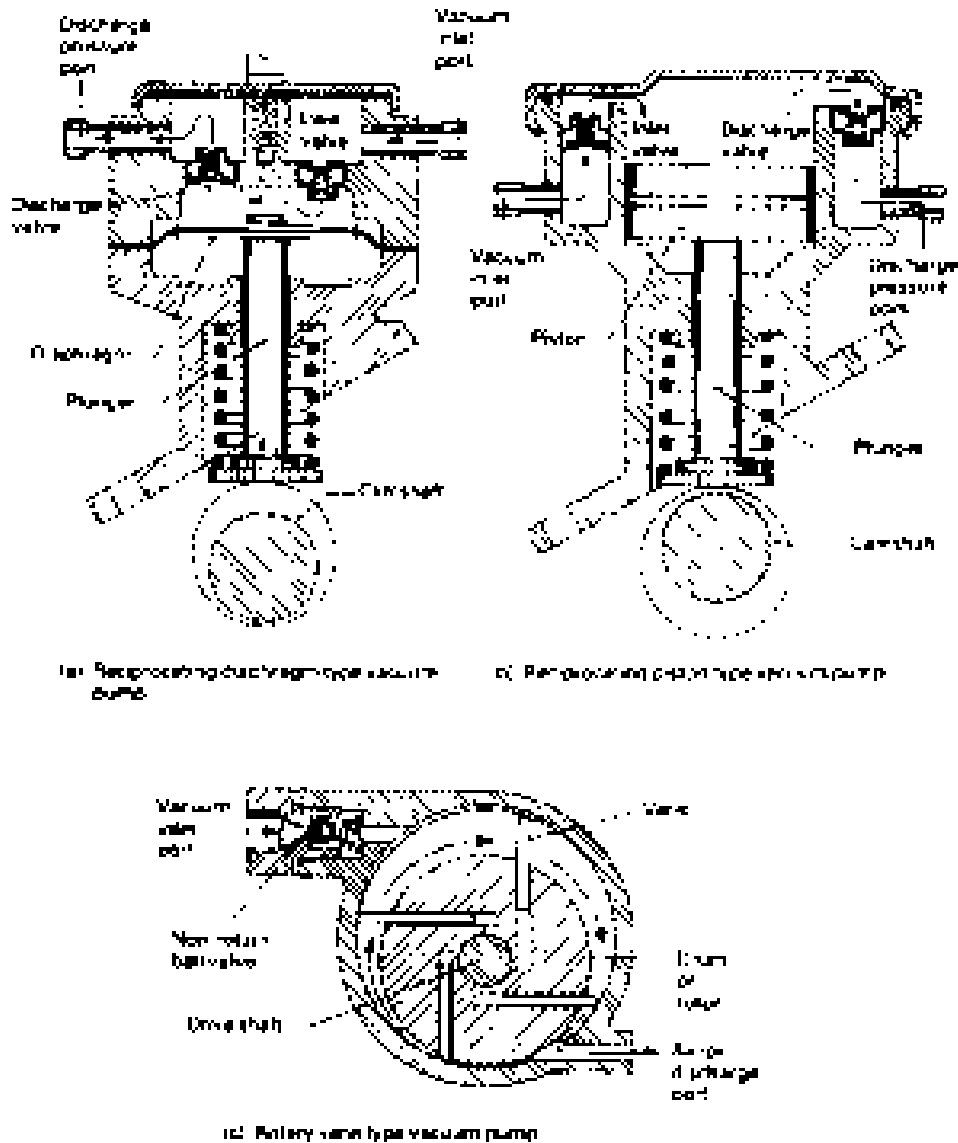
Fig. 11.47 (a–c) Types of vacuum pumps

engine oil supply. Therefore, the discharge port
returns the oil-contaminated air discharge back to
the engine crank case.

11.8.4 Hydraulic servo assisted steering and brake system

ntroduction to hydraulic servo assistance (Fig. 11.48)
The alternative use of hydraulic servo assistance is
particularly suited where emission control devices
to the engine and certain types of petrol injection
system reduce the available intake manifold

vacuum, which is essential for the effective opera-
tion of vacuum servo assisted brakes. Likewise,
diesel engines, which produce very little intake
manifold vacuum, require a separate vacuum
source such as a vacuum pump (exhauster) to oper-
ate a vacuum servo unit; therefore, if power assis-
tant steering is to be incorporated it becomes
economical to utilize the same hydraulic pump
(instead of a vacuum pump) to energize both the
steering and brake servo units.

The hydraulic servo unit converts supplied fluid
energy into mechanical work by imposing force
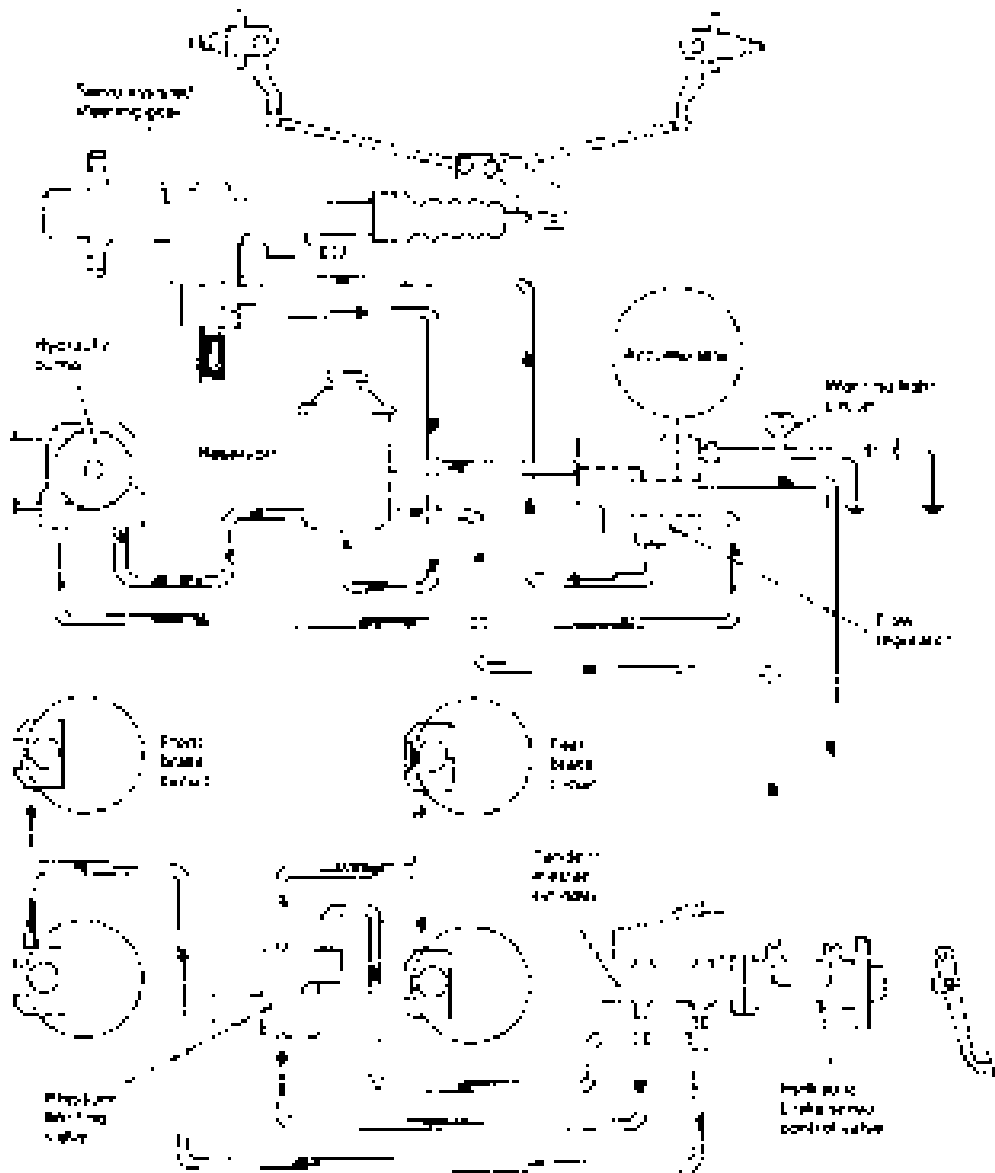
Fig. 11.48 Hydraulic servo-assisted and brake system (ATE)

and movement to a power piston. A vane type pump provides the pressure energy source for both the power assisted steering and for the brake servo. When the brake accumulator is being changed approximately 10% of the total pump output is used, the remaining 90% of the output returns to the power steering system. When the accumulator is fully charged, 100% of the pump output returns via the power steering control unit to the reservoir. Much higher operating pressures are used in a hydraulic servo compared to the vacuum type servo. Therefore the time needed to actuate the brakes is shorter.

The proportion of assistance provided to the pedal effort is determined by the cross-sectional area ratio of both the power piston and reaction piston. The larger the power piston is relative to the reaction piston, the greater the assistance will be and vice versa.

In the event of pump failure the hydraulic accumulator reserves will still provide a substantial number of power assisted braking operations.

Pressure accumulator with flow regulator and cut-out valve unit (Fig. 11.49(a and b)) The accumulator provides a reserve of fluid under pressure if the engine should stall or in the event of a failure of the source of pressure. This enables several brake applications to be made to bring the vehicle safely to a standstill.

The pressure accumulator consists of a spherical container divided in two halves by a rubber diaphragm. The upper half, representing the spring media, is pressurized to 36 bar with nitrogen gas and the lower half is filled with the operating fluid under a pressure of between 36 and 57 bar. When the accumulator is charged with fluid, the diaphragm is pushed back, causing the volume of the nitrogen gas to be reduced and its pressure to rise. When fluid is discharged, the compressed nitrogen gas expands to compensate for these changes and the flexible diaphragm takes up a different position of equilibrium. At all times both gas and fluid pressures are similar and therefore the diaphragm is in a state of equilibrium.

Accumulator being charged (Fig. 11.49(a)) When the accumulator pressure drops to 36 bar, the cut-out spring tension lifts the cut-out plunger against the reduced fluid pressure. Immediately the cut-out ball valve opens and moves from its lower seat to its uppermost position. Fluid from the vane type pump now flows through the cut-out valve, opens the non-return conical valve and permits fluid to pass through to the brake servo unit and to the under side of the accumulator where it starts to compress the nitrogen gas. The store of fluid energy will therefore increase. At the same time, the majority of fluid from the vane type pump flows to the power assisted steering control valve by way of the flutes machined in the flow regulator piston.

Accumulator fully charged (Fig. 11.49(b)) When the accumulator pressure reaches its maximum 57 bar, the cut-out valve ball closes due to the fluid pressure pushing down the cut-out plunger. At the same time, pressurized fluid in the passage between the non-return valve and the rear of the flow regulating piston is able to return to the reservoir via the clearance between the cut-out plunger and guide bore. The non-return valve closes and the fluid pressure behind the flow regulating piston drops. Consequently the fluid supplied from the pump can now force the flow regulator piston further back against the spring so that the total fluid flow passes unrestricted to the power assisted steering control valve.

Hydraulic servo unit (Fig. 11.50(a, b and c)) The hydraulic servo unit consists of a power piston which provides the hydraulic thrust to the master cylinder. A reaction piston interprets the response from the brake pedal input effort and a control tube valve, which actuates the pressurized fluid delivery and release for the servo action.

Brakes released (Fig. 11.50(a)) When the brake pedal is released, the push rod reaction piston and control tube are drawn towards the rear, firstly causing the radial supply holes in the control tube to close and secondly opening the return flow hole situated at the end of the control tube. The pressurized fluid in the operating chamber escapes along the centre of the control tube out to the low pressure chamber via the return flow hole, where it then returns to the fluid reservoir (container). The power piston return spring pushes the power piston back until it reaches the shouldered end stop in the cylinder.

Brakes normally applied (Fig. 11.50(b)) When the brake pedal is depressed, the reaction piston and control tube move inwards, causing the return flow hole to close and partially opening the control tube supply holes. Pressurized fluid from either the accumulator or, when its pressure is low, from the pump, enters the control tube central passage and passes out into the operating chamber. The pressure build-up in the operating chamber forces the power piston to move away from the back end of the cylinder. This movement continues as long as the control tube is being pushed forwards (Fig. 11.50(b)).

Holding the brake pedal in one position prevents the control tube moving further forwards. Consequently the pressure build-up in the operating chamber pushes the power piston out until the radial supply holes in both the power piston and control tube are completely misaligned. Closing the radial supply holes therefore produces a state of balance between the operating chamber fluid thrust and the pressure generated in the tandem master cylinder.

The pressure in the operating chamber is applied against both the power piston and the reaction piston so that a reaction is created opposing the pedal effort in proportion to the amount of power assistance needed at one instance.

501
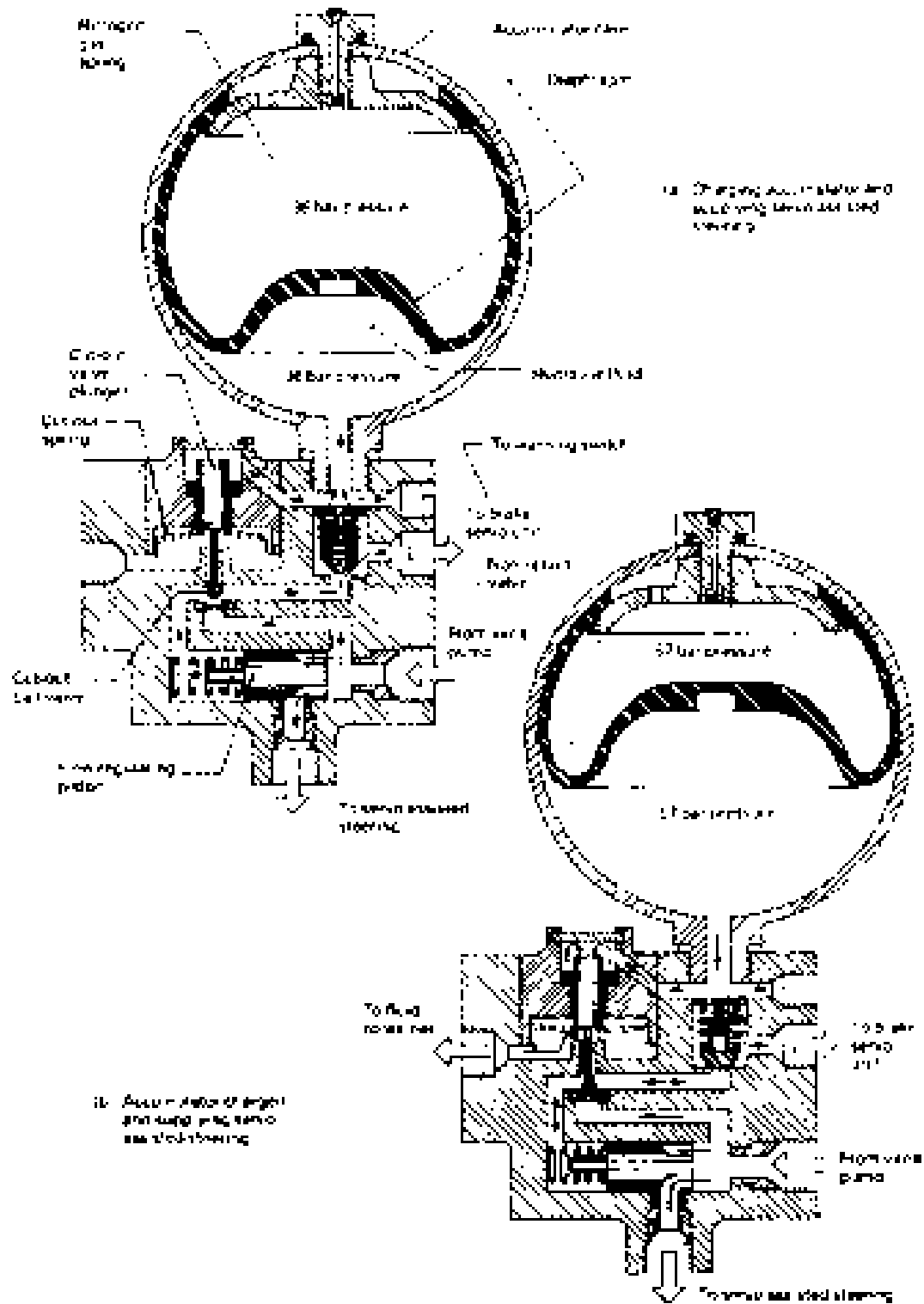
Fig. 11.49 (a and b) Flow regulator with pressure accumulator
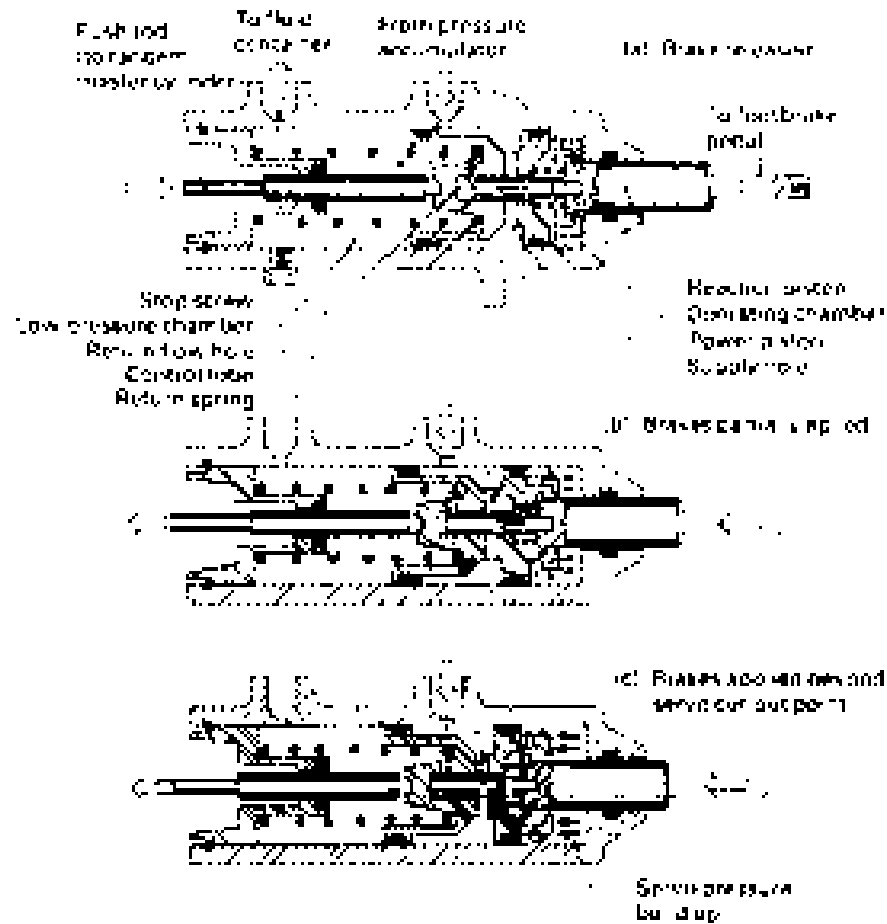
Fig. 11.50 (a–c) Hydraulic servo unit

Braking beyond the cut out point (Fig. 11.50(c)) When the accumulator cut-out pressure is reached, the control tube touches the power piston, causing the radial supply holes in the control tube to fully align with the power piston. Under these conditions, the accumulator is able to transfer its maximum pressure to the operating chamber. The power piston is therefore delivering its greatest assistance. Any further increase in master cylinder output line pressure is provided by the brake pedal effort alone, as shown in Fig. 11.51, at the minimum and maximum cut-out pressures of 36 and 57 bar respectively.

Rear brake circuit pressure regulator and cut-out device (Fig. 11.52(a, b and c)) The rear brake pressure regulator and cut-off device provide an increasing front to rear line pressure ratio, once the line pressing in the rear pipe line has reached

some predetermined minimum value. In other words, the pressure rise in both front and rear pipe lines increases equally up to some pre-set value, but beyond this point, the rear brake pipe line pressure increases at a much reduced rate relative to the front brakes. An additional feature is that if the front brake circuits should develop some fault, then automatically the pressure regulator is bypassed to ensure that full master cylinder fluid pressure is able to operate the rear brakes.

Low brake fluid pressure (Fig. 11.52(a)) When the brakes are lightly applied, the pressure in the front pipe line circuit pushes the cut-off piston over against the opposing spring force. Simultaneously, fluid from the master cylinder enters the inlet port, passes through the open pressure reducing valve, then flows around the wasted cut-off piston on its way out to the rear brake pipe line circuit.

503
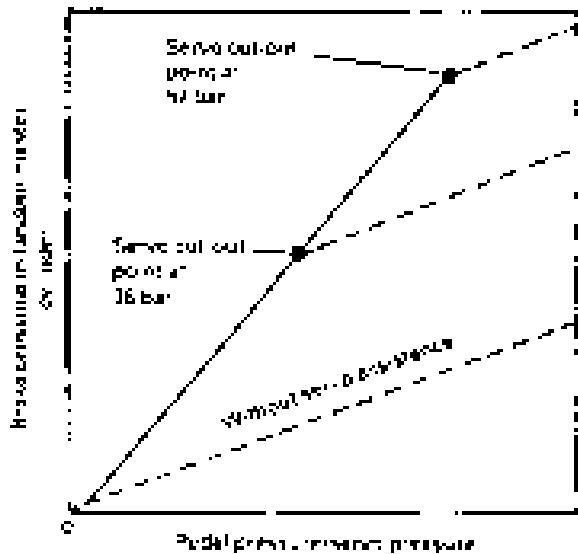
Fig. 11.51 Hydraulic servo action pressure characteristics

High braking fluid pressure (Fig. 11.52(b))   With increased foot pedal effort, the fluid pressure entering the inlet port and passing through the pressure reducing valve, on the way to the rear brake circuit outlet port, rises proportionally. Eventually the resultant force imposed on the stepped piston, caused by the fluid pressure acting on the large and small surface areas of the piston, pushes it outwards against the resistance of the preload spring until the pressure reducing valve closes. Further master cylinder generated pressure acting on the annular face of the stepped piston forces the piston to move in the opposite direction, thereby increasing the rear brake pipe line fluid pressure on the large surface area side of the piston, but at a reduced rate to that of the master cylinder output fluid pressure. The pressure reducing valve is immediately dislodged from its seat. The pressure reducing valve opens and closes repeatedly with rising master cylinder output fluid pressure until the reduced pressure on the large surface area output side of the piston has adjusted itself. These pressure characteristics are shown in Fig. 11.52(d).

 ront brake circuit fail condition (Fig. 11.52(c))   If the front brake circuit should fail, the pressure imposed on the cut-off piston collapses, enabling the spring at the opposite end to push over the cut-off piston so that the left hand side of the shuttle valve opens and the right hand side closes. The

pressure reducing valve passage to the rear brake line is immediately cut off and the direct passage via the left hand shuttle valve is opened. Pressure from the master cylinder is therefore transmitted unrestricted directly to the rear brake pipe line. The effect of failure in the front brake circuit will be a considerable increase in foot pedal movement.

11.9 Pneumatic operated disc brakes (for trucks and trailers)

Heavy duty disc brake arrangements normally use a floating-caliper design which does not resort to hydraulic actuation, but instead relies on compressed air to supply the power source via a diaphragm operated air chamber actuator. The disc brake unit consists of a rotating disc attached to the road-wheel hub and a floating caliper supported on the caliper carrier which is itself bolted to the stub-axle or casing.

11.9.1 Floating caliper with integral half eccentric lever arm (Fig. 11.53(a and b))

When the brakes are applied air pressure pushes the actuator chamber diaphragm to the left hand side and so tilts the actuator lever about the two half needle roller bearing pivots (Fig. 11.53(a and b)). This results in the eccentric (off-set) bearing pin pushing the right hand friction pad towards the right hand side of the disc via the bridge block, see Fig.11.53(b). Simultaneously as the right hand friction pad bears against the right hand side of the disc, a reaction force now acts on the caliper and is transferred to the opposite friction pad so that both pads squeeze the disc with equal force. Thus the caliper in effect floats; this therefore centralizes the friction pads so that both pads apply equal pressure against their respective faces of the disc. The brake torque produced depends upon the air pressure relayed to the brake actuator chamber, the effective diaphragm area of the chamber and the leverage ratio created by the lever arm 'R' and eccentric off-set 'r', i.e. R/r. When the brakes are released the pull-off spring pushes the bridge block assembly back to the off position, thus producing a running clearance between the pads and disc, see Fig. 11.53(a).

 loating caliper with eccentric shaft and lever (Fig. 11.54(a))   With this type of heavy duty commercial vehicle disc brake a floating caliper is used in conjunction with an eccentric and lever to clamp the pads against the friction faces of the disc. The eccentric part of the eccentric shaft is surrounded
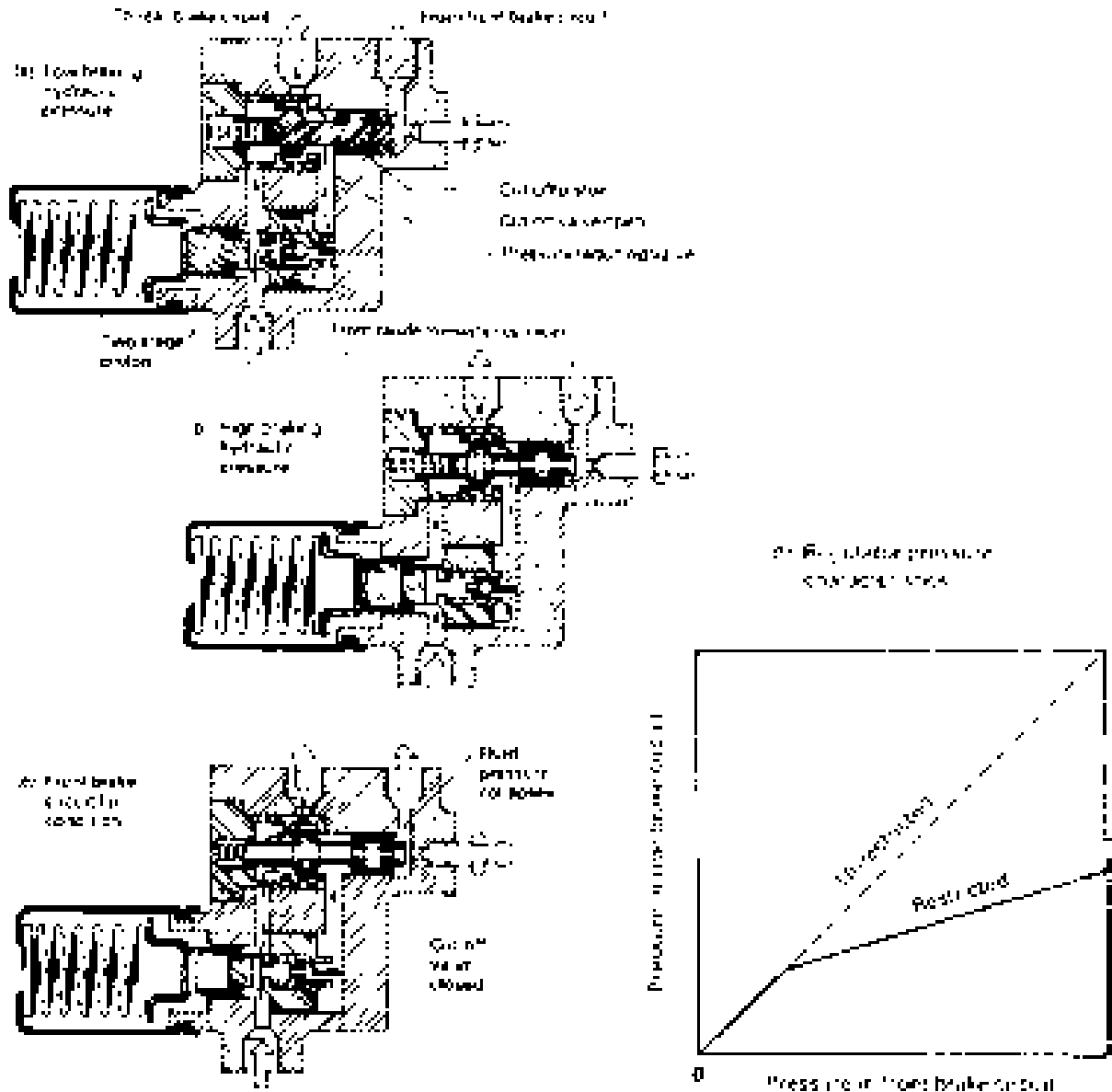
Fig. 11.52 (a–d) Rear brake circuit pressure regulator and cut-off device

by needle rollers positioned inside a bored hole in the bridge block and is connected to the inside pad via two threaded adjustment barrels and a load plate. On either side of the eccentric are stub shafts which are mounted via needle rollers in the caliper.

When the brakes are released the lever arm takes up a position in which the lobe side of the eccentric leans slightly to the right hand side of the vertical (Fig. 11.54(c)). As the brake is applied (Fig. 11.54(b)) the lever arm and eccentric swivels so that the lobe moves to the vertical position or just beyond, hence the bridge block will have moved to the right hand side (towards the disc face) by $x = x_2 - x_1$ where x equals pad take-up clearance. Thus when the brakes are applied compressed air is released into the actuator chamber; this pushes the diaphragm and push rod to the left hand side, causing the lever arm to rotate anticlockwise. As a result the eccentric lobe forces

505

Actuator air chamber

Lever arm

Half eccentric

Bearing pin

Bridge block

Pull-off spring

Pad (right hand)

Ventilated disc

Pad (left hand)

Caliper

R

r

(a) brake released

Force

Reaction
force

(b) brake applied
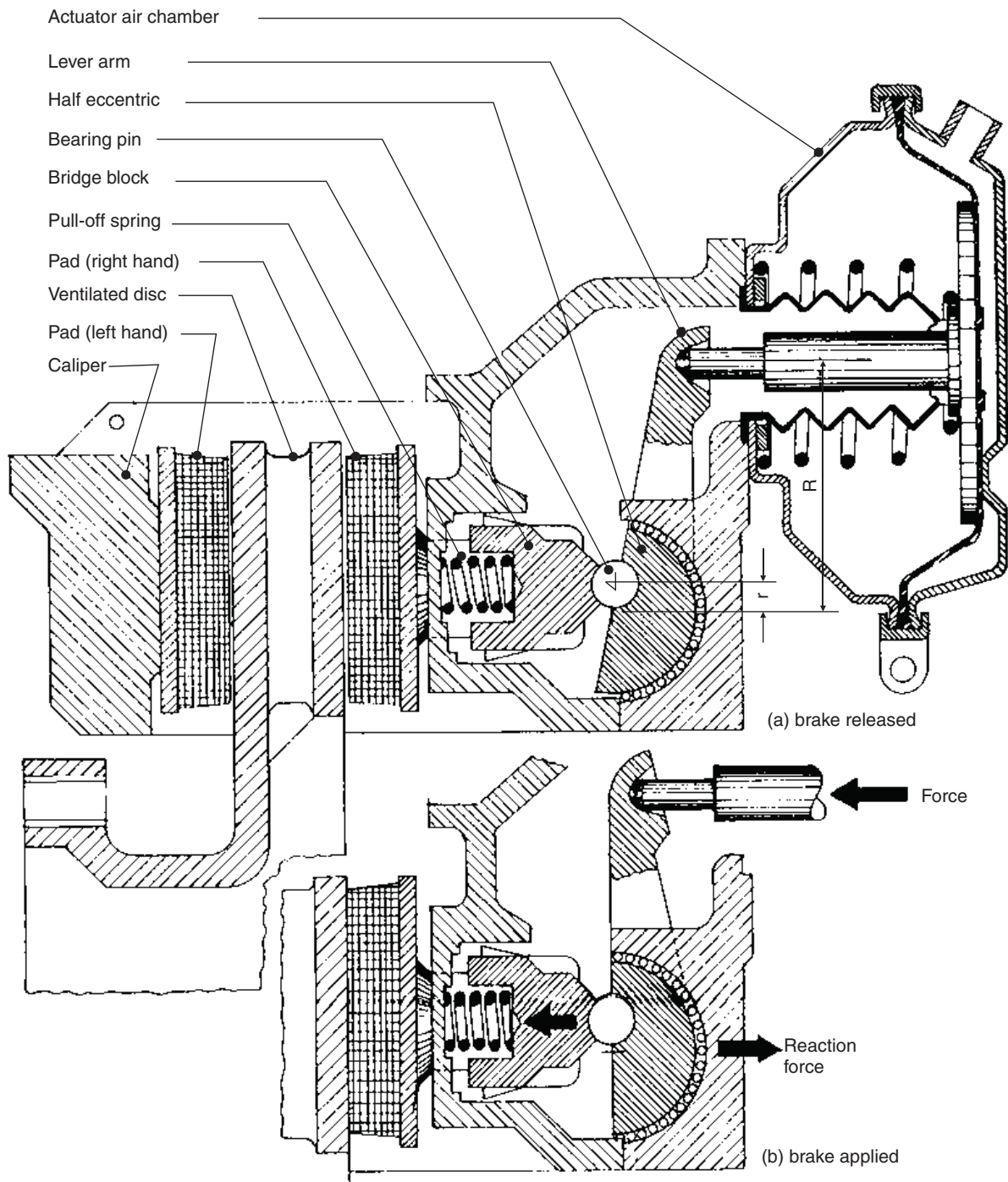
Fig. 11.53 (a and b)   Pneumatic operated disc brake – floating caliper with integral half eccentric lever arm

506

(b) Brake applied

Force

Shaft and eccentric

(c) Brake released

Bridge block

Needle rollers

Eccentric shaft

Lever arm

Actuator air chamber

$x = x_2 - x_1$
where
x = pad take-up

Caliper carrier

(a) Plan view

Adjuster gear train cassette

Override clutch

Threaded adjustment barrel

Bridge block

Caliper

Pads

Vented disc

Load plate

Slide pin

Threaded adjustment post

Gear plate segment

Bevel gear
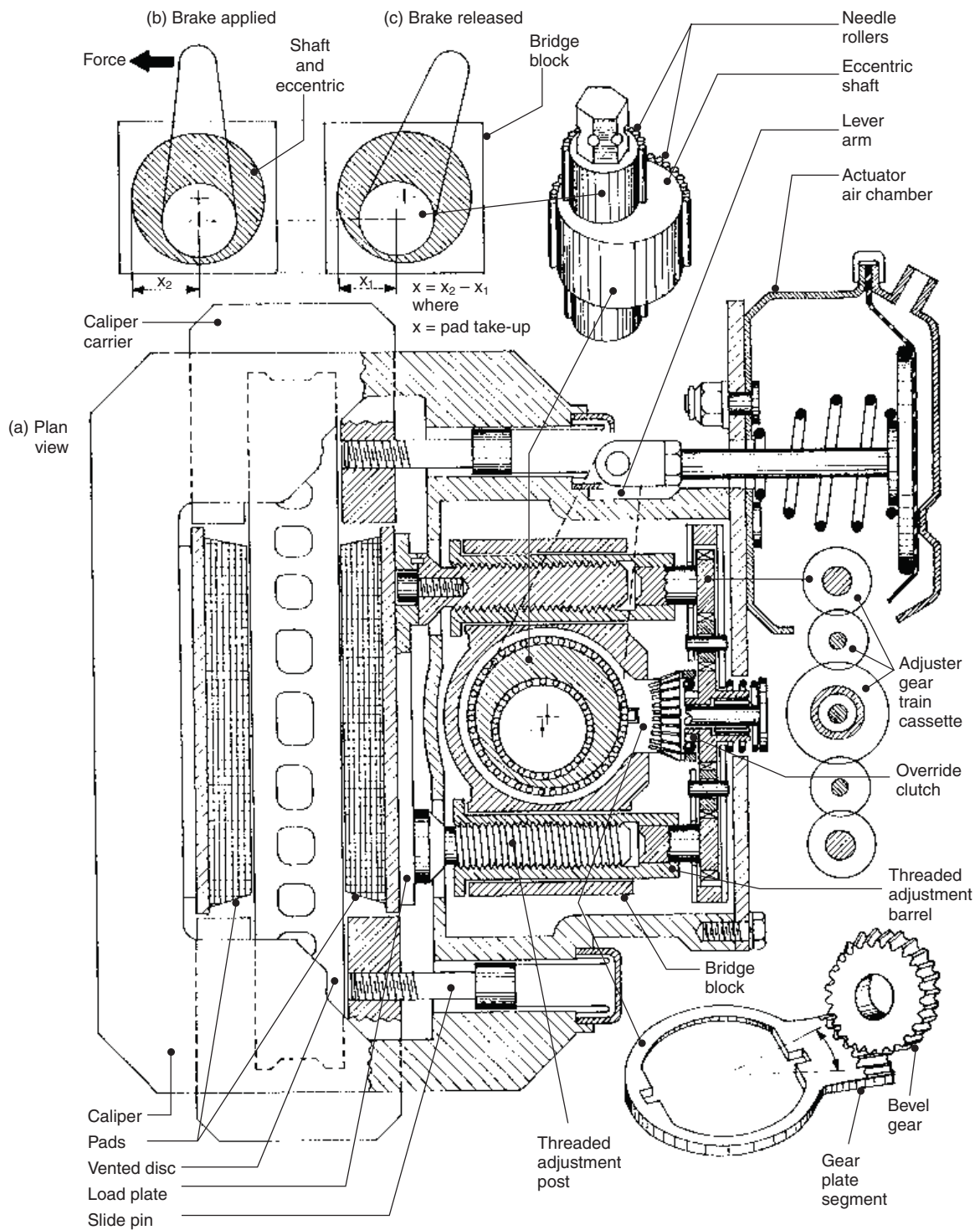
Fig. 11.54 (a–c) Pneumatic operated disc brake – eccentric shaft and lever with gear driven automatic adjustment mechanism

507

(b) Brake applied   (c) Brake released

Sleeve fork
& barrel rotated
while brake is
applied

Sleeve fork
in the brake
released position

Force

Bearing
pin

Needle
rollers

(a) Plan view

Sleeve
fork

Strike
pins

Half
eccentric

Override
clutch

Drive
spindle

Sleeve
fork

Chain
sprocket
wheel

Fork
prongs

Strike pins

Adjuster
chain

Bearing pin

Pull-off
spring

Needle
rollers

Sprocket
wheel &
spindle

Half eccentric
and lever arm

Threaded
adjustment
barrel

Bridge block

Slide pin & sleeve

Load plate

Pads

Ventilated disc

Caliper carrier

Fig. 11.55 (a–c)   Pneumatic operated disc brake – half eccentric shaft and lever with gear driven automatic adjustment mechanism

the bridge block and consequently the inner pad towards the right hand face of the disc. Conversely a reaction force acting though the eccentric stub shaft and caliper pulls the whole caliper, and subsequently the outer pad, towards the left hand face of the disc until the desired amount of friction force is generated between the pads and disc to either slow down or park the vehicle.

Automatic pad clearance gear-driven type ad uster mechanism (Fig. 11.54(a, b and c)) A constant running clearance between the pad and disc is maintained with this mechanism; this device operates by the to and fro movement of the lever arm about the eccentric stub shafts every time the brakes are applied and released (Fig. 11.54(a)). Drawing together of the brake pads is achieved by partial rotation of the eccentric lobe within the bridge block, thus movement is transmitted to the pads via the two threaded adjustment barrels which are screwed either side of the eccentric onto the threaded adjuster posts which are rigidly attached to the inner brake pad load plate.

A gear-plate segment is attached to one side of the eccentric via a slot and tongue. The segment teeth mesh with a bevel gear which houses the override clutch (one-way clutch operating between balls rolling up and down inclined plains), see Fig. 11.54(a). Any partial rotation of the eccentric is transferred to the threaded adjustment barrels via the override clutch and the train of gears.

Thus every time the brake lever arm moves from the released to the applied position, the threaded adjustment barrels are partially screwed out from the threaded adjustment posts, thereby causing the load plate and pad to move further towards the inside face of the disc. Conversely each time the lever arm moves from the applied to the released position, the override one-way clutch disengages, so preventing the threaded adjustment barrels being screwed in again. Eventually after many braking applications, the threaded adjustment barrels will have screwed out the threaded adjustment posts sufficiently to cause the inner pad to touch the inner face of the disc.

Eventually after many braking applications, the threaded adjustment barrels will have screwed the threaded adjustment posts sufficiently out to cause the inner pad to touch the inner face of the disc. At this point, the slight tightening between the male and female threads generates sufficient friction in the screw threads and underneath the flange head

of the threaded adjustment barrels to cause the override clutch to slip, hence further rotation of the threaded adjustment barrels ceases. As pad and disc wear occurs, the threaded adjustment barrels once again commence to turn; a constant running clearance is thus maintained during service.

Automatic pad clearance chain-driven type ad uster (Fig. 11.55(a, b and c)) This mechanism maintains a constant running clearance between the pads and disc; the adjuster is operated by the rocking movement of the lever arm each time the brakes are applied and released (Fig. 11.55(a and b)). Braking force is transferred from the lever arm (Fig. 11.55(a)) to the brake pad via the bridge block and the two threaded adjustment barrels which are screwed on either side of the lever arm into the bridge block; the shouldered blind ends of the threaded barrels fit into recesses formed in the load plate. On one side of the lever arm (Fig. 11.55(a, b and c)) are two fork pins which mesh with three prongs formed on the fork sleeve; this sleeve slides over a drive spindle situated inside one of the threaded adjustment barrels. An override clutch is formed at the opposite end to the pronged teeth of the fork sleeve (Fig. 11.55(a)), this consists of a race of balls rolling on ramps (inclined plains) formed by the ball-guide grooves.

Every time the brakes are applied the lever arm tilts causing the meshing strike pins to twist the fork sleeve clockwise and then back to its original position when the brakes are released (Fig. 11.55 (b and c)). Thus the clockwise movement of the sleeve is relayed to the threaded adjustment barrel via the override clutch, but the fork sleeve anticlockwise return movement of the override clutch releases the threaded adjustment barrel, thus the threaded barrel is progressively screwed towards the disc thereby taking up the running clearance caused by pad and disc wear. The running clearance is maintained by the slackness between the strike pin and prong teeth. Even take-up of the running clearance is obtained via the chain sprocket wheels and chain (Fig. 11.55(a)) which transfers the same rotary motion to the second threaded adjustment barrel. Any over-adjustment will cause the override clutch to slip. The sum clearance of both sides of the disc, that is, the total running clearance, should be within 0.6 and 0.9 mm. A larger clearance will cause a take-up clearance time delay whereas a very small clearance may lead to overheating of the discs and pads.

12 Air operated power brake equipment and vehicle retarders

12.1 ntroduction to air powered brakes

As the size and weight of road vehicles increase there comes a time when not only are manual brakes inadequate, but there is no point in having power assistance because the amount of braking contributed by the driver's foot is insignificant relative to the principal source of power, be it vacuum or hydraulic energy, and therefore power operated brakes become essential. A further consideration is that the majority of heavy commercial or public service vehicles are propelled by diesel engines which do not have a natural source of vacuum and therefore require a vacuum pump (exhauster) driven from the engine to supply the vacuum energy. However, if a separate pump has to be incorporated to provide the necessary power transmitting media, a third energy source with definite advantages and few disadvantages can be used; that is compressed air.

Reciprocating compressors driven off the engine can operate efficiently and trouble-free at pressures in the region of 7–8 bar, whereas vacuum assisted brakes can only work at the most with depressions of 0.9 bar below atmospheric pressure. Consequently compressed air has a power factor advantage of between 7 and 8 times that for an equivalent vacuum source when used as a force transmitting media.

Conversely, hydraulic pumps are compelled to work at pressures of between 50 and 60 bar. The pressures generated in the pipe lines may reach values of 100 bar or even more. Consequently, because of these high pressures, small diameter servo cylinders and small bore pipes are utilized. This may appear to merit the use of hydraulic energy but, because of the very high working pressure in a hydraulic operated brake system, much more care has to be taken to avoid fluid leakage caused by wear or damage. Compressed air as a power transmitting media would operate at pressures of only one-tenth of an equivalent hydraulic source, but for large vehicles where there is more space, there is no real problem as much larger diameter cylinders can be used. In addition, if there is a leakage fault in a hydraulic layout it will eventually drain the supply fluid so that the brakes cannot continue to function, whereas small leakages of air in an air power operated braking system will not prevent the brakes operating even if this does take place at slightly reduced stopping efficiency.

12.2 Air operated power brake systems

12.2.1 Truck air over hydraulic brake system (Fig. 12.1)

Compressed air supply Air is drawn into the compressor and then discharged into and out of the wet tank where it is semi-dried; it then flows to the multi-circuit protection valve, here it divides to feed the two service reservoirs. At the same time, pressurized air from the reservoirs combine through internal passages in the multi-circuit protection valve to supply the remote spring brake actuator via the hand control valve.

Service line circuit (Fig. 12.1) There are two service lines feeding into a tandem power cylinder controlled by a dual foot valve, so that if a fault develops in one service line the air supply to the other circuit will not be interrupted. The air pressure is then converted to hydraulic pressure by the power piston push rod pushing the tandem master cylinder hydraulic piston forward. The hydraulic fluid supply is split into two circuits feeding the front and rear brake expander cylinders. To balance the proportion of braking provided by the rear axle according to the load carried, a hydraulic load sensing valve is installed on the tandem master cylinder rear axle output circuit. This therefore modifies the fluid pressure reaching the rear brake cylinders.

Secondary line circuit (Fig. 12.1) With the dual air and hydraulic lines, both systems operate independently and therefore provide a safeguard against failure of one or the other circuit. Thus the hand control valve is used only to park the vehicle.
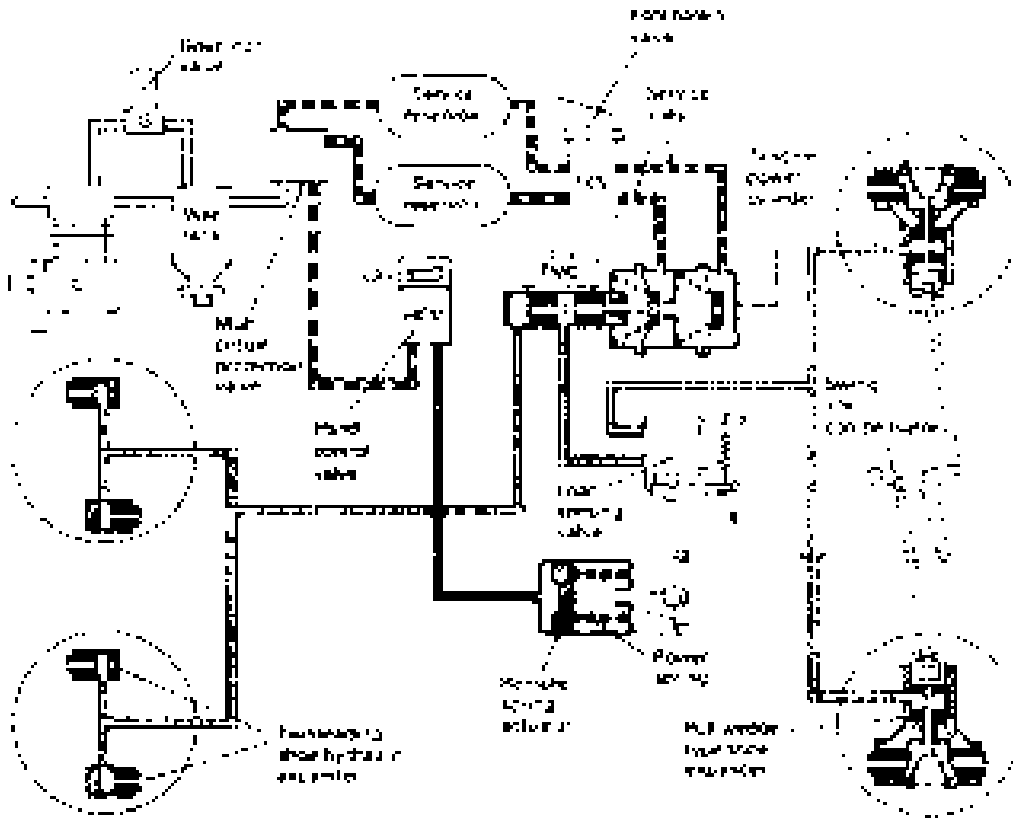
Fig. 12.1 Truck air over hydraulic system

Moving the hand valve lever from 'off' to 'park' position exhausts air from the remote spring actuator chamber. This permits the power spring within the actuator to expand and exert its full pull to the mechanical parking brake rod linkage.

12.2.2 Tractor three line brake system (Fig. 12.2)

Compressed air supply (Fig. 12.2)   The compressor in this arrangement is controlled by a separate unloader valve. An alcohol evaporator is installed in the air intake, so that in cold weather alcohol can be introduced into the airstream to lower the freezing point of any water which may be present. When the compressor is running light, a check valve built into the evaporator prevents alcohol entering the air intake. Pressurized air from the compressor is then delivered to both the service and secondary park reservoirs via the check valves on the inlet side of each reservoir.

Service line circuit (Fig. 12.2)   When the foot pedal is depressed, air from the service reservoir is permitted to flow directly to the tractor's front and rear service line chambers in each of the double diaphragm actuators which are mounted on the tractor axles. At the same time, a pressure signal is passed to the relay valve piston. This opens the valve so that the service storage line pressure flows from the service reservoir to the service line coupling (yellow) via the pressure protection valve. The pressure protection valve in the service storage (emergency) line and the relay valve in the service line safeguard the tractor's air supply, should a large air leak develop in the flexible tractor to trailer coupling hose or if any other fault causes a loss of air pressure.

Secondary line circuit (Fig. 12.2)   Applying the hand control valve lever delivers a controlled air pressure from the secondary park reservoir to the front wheel secondary chambers, which form part of the double diaphragm actuators, and to the secondary line (red) coupling, which then delivers pressurized air to the trailer brakes via a flexible hose. Note that there is no secondary braking to

511
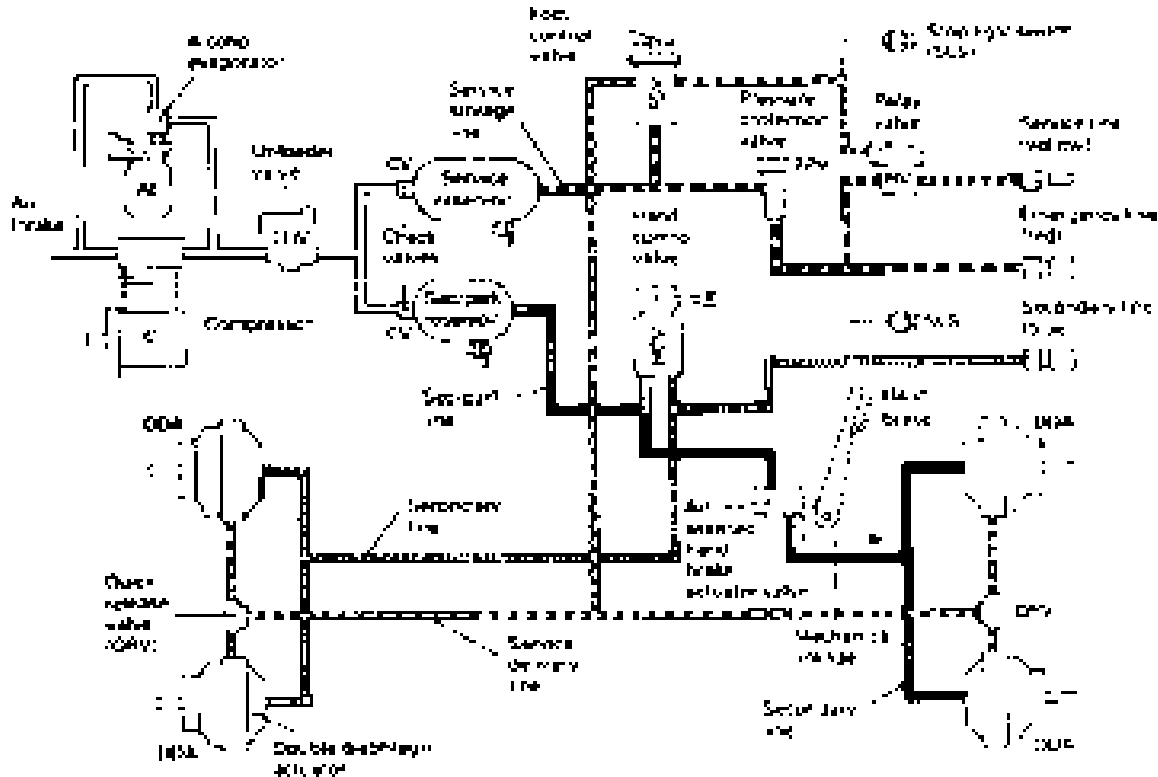
Fig. 12.2 Tractor three line brake system

the tractor's rear axle to reduce the risk of jack-knifing during an emergency application.

Parking circuit (Fig. 12.2) Applying the hand brake lever opens the hand brake valve so that pressurized air flows to the rear axle parking line chambers within the double diaphragm actuators to apply the brakes. At the same time, the mechanical parking linkage locks the brake shoes in the applied position and then releases the air from the parking actuator chambers. This parking brake is therefore mechanical with air assistance.

12.2.3 Trailer three line brake system (Fig. 12.3)
All trailer air braking systems have their own reservoir which is supplied through the emergency line from the tractor's service reservoir.

Service line circuit (Fig. 12.3) When applying the brakes, air pressure from the tractor's relay valve signals the emergency relay valve to open and supply air pressure from the trailer's own reservoir to the trailer's service line brake actuator chambers relative to that applied to the tractor brakes. The
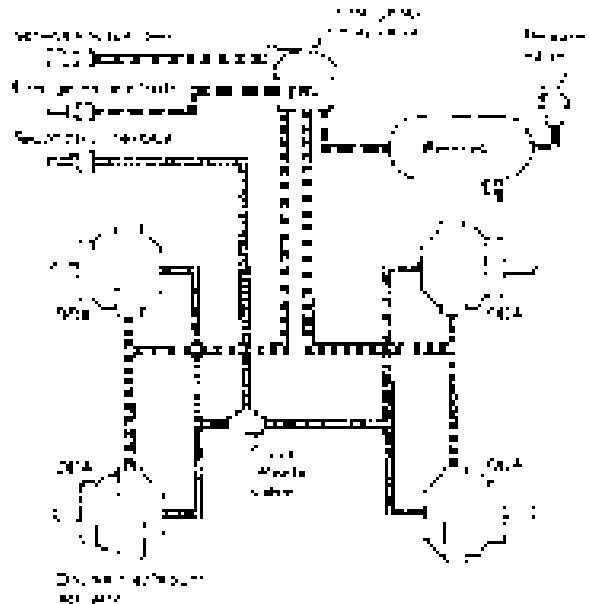
Fig. 12.3 Trailer three line brake system

object of the separate reservoir and relay valve installed on the trailer is to speed up the application and release of the trailer brakes, which are at some distance from the driver's foot control valve. Should there be a reduction in emergency line pressure below some predetermined minimum, the emergency relay valve will sense this condition and will automatically apply the trailer service brakes.

Secondary line circuit (Fig. 12.3) The secondary braking system of the trailer is controlled by the hand control valve mounted in front of the driver. Moving the hand control valve lever towards the applied position delivers a graduable air pressure via the secondary lines to the secondary chamber within each double diaphragm actuator. A quick release valve incorporated at the junction between the trailer's front and rear brakes speeds up the exhausting of the secondary chambers and, therefore, the release of the secondary brakes.

To release the trailer brakes when the trailer is detached from the tractor caused by the exhausting of the emergency line, a reservoir release valve is provided which should be moved to the 'open' piston to release the trailer brakes.

12.2.4 Towing truck or tractor spring brake three line system (Fig. 12.4)

Compressed air supply (Fig. 12.4) Air pressure is supplied by a compressor driven off the engine. Built into the compressor head is an unloaded mechanism which is controlled by a governor valve and which senses pressure change through the wet tank. Installed on the intake side of the compressor is an alcohol evaporator which feeds in very small quantities of alcohol spray when the compressor is pumping. As a result, it lowers the freezing temperature of the wet air induced into the compressor cylinder. When the compressor is running light, a check valve prevents alcohol spray entering the airstream, thereby reducing the alcohol consumption. The compressor supplies pressurized air to both service and secondary/ park reservoirs via non-return check valves.

Service line circuit (Fig. 12.4) When the driver depresses the dual foot valve, air flows from the service reservoir through the service delivery line (yellow) directly to the front wheel service line actuator chamber, and indirectly via a variable load valve which regulates the air pressure,
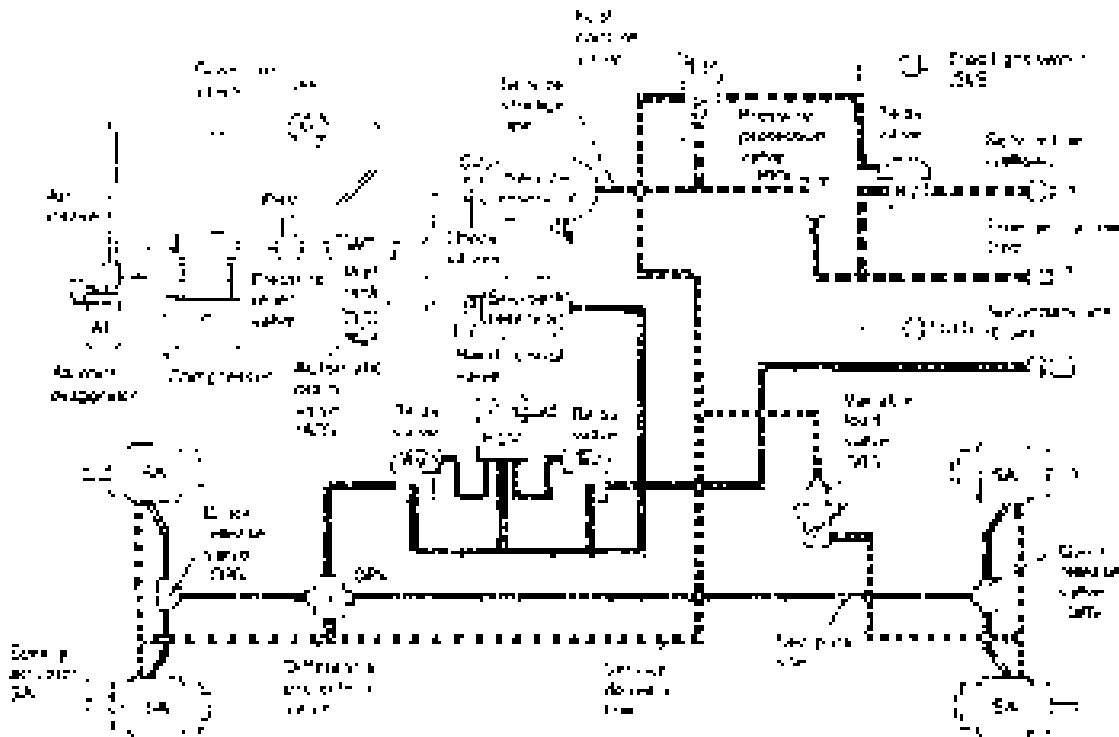
Fig. 12.4 Towing truck or tractor spring brake three line system

according to the loading imposed on the rear axle, to the rear wheel service chamber actuators. Compressed air is also delivered to both the service and the emergency line couplings via the relay valve and the pressure protection valve. This therefore safeguards the tractor air supply should there be a hose failure between the tractor and trailer. A differential protection valve is installed between the service line and the secondary/park line to prevent both systems operating simultaneously which would overload the foundation brakes.

Secondary/park line circuit (Fig. 12.4)   Air is supplied from the secondary/park reservoir to the hand control valve and to a pair of relay valves. One relay valve controls the air delivered to the spring brake actuator, the other controls the service line air supply to the trailer brakes. With the hand control valve in the 'off' position, air is delivered through the secondary/park line relay valve to the spring brakes. The secondary/park spring brakes are held in the released position due to the compression of each power spring within the actuator. As the spring brakes are being released, the secondary line to the trailer is exhausted of compressed air via its relay valve. Moving the hand

control valve lever to the 'on' position progressively reduces the secondary/park line pressure going to the spring brake. The secondary line pressure going to the trailer coupling increases, thereby providing a tractor to trailer brake match. Moving the hand control valve to the 'park' position exhausts the air from the trailer secondary line and the spring brake secondary/park line. The tractor foundation brakes are then applied by the thrust exerted by the power spring within the actuator alone. The release of the parking brake is achieved by delivering air to the spring brake when the hand control valve is moved to the 'off' position again.

12.2.5 *Towing truck or tractor spring brake two line system* (Fig. 12.5)

Compressed air supply (Fig. 12.5)   The air supply from the compressor passes through the air dryer on its way to the multi-circuit protection. The output air supply is then shared between four reservoirs; two service, one trailer and one secondary/ park reservoirs.

Service line circuit (Fig. 12.5)   The air delivered to the service line wheel actuator chambers is
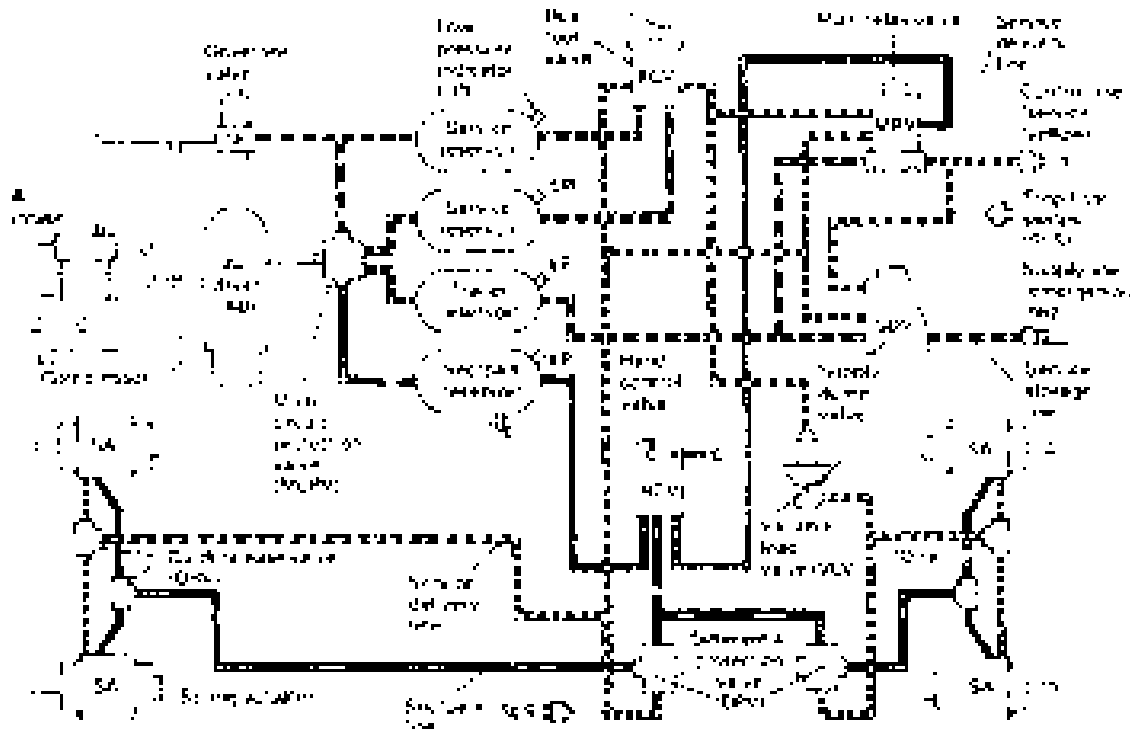
Fig. 12.5   Towing truck or spring brake two line system

514

provided by a dual foot valve which splits the service line circuits between the tractor's front and rear wheels. Therefore, if one or other service line circuit should develop a fault, the other circuit with its own reservoir will still function. At the same time as the tractor service brakes are applied, a signal pressure from the foot valve passes to the multi-relay valve. This opens an inlet valve which permits air from the trailer reservoir to flow to the control line (service line — yellow) trailer coupling.

To prevent both service line and secondary/park line supplies compounding, that is, operating at the same time, and overloading the foundation brakes, a differential protection valve is included for both the front and rear axle brakes.

Secondary/park line circuit (Fig. 12.5) A secondary braking system which incorporates a parking brake is provided by spring brakes which are installed on both front and rear axles. Control of the spring brakes is through a hand valve which provides an inverse signal to the multi-relay valve so that the trailer brakes can also be applied by the hand control valve.

With the hand control valve in the 'off' position the secondary line from the hand valve to the multi-relay valve, and the secondary/park line, also from the hand valve, going to the spring brake actuators via the differential protection valves, are both pressurized. This compresses the power springs, thereby releasing the spring brakes. During this period no secondary line pressure signal is passed to the trailer brakes via the multi-relay valve.

When the hand valve is moved towards the 'applied' position, the secondary line feeding the multi-relay valve and the secondary/park line going to the spring brakes reduces their pressures so that both the tractor's spring brakes and the trailer brakes are applied together in the required tractor to trailer proportions.

Moving the hand valve lever to the 'park' position exhausts the secondary/park line going to the spring brakes and pressurizes the secondary line going to the multi-relay valve. As a result, the power springs within the spring actuators exert their full thrust against the foundation brake cam lever and at the same time the trailer control line (service line) is exhausted of compressed air. Thus the vehicle is held stationary solely by the spring brakes.

Multi-relay valve (Fig. 12.25(a–d)) The purpose of the multi-relay valve is to enable each of the two service line circuits to operate independently

should one malfunction, so that trailer braking is still provided. The multi-relay valve also enables the hand control valve to operate the trailer brakes so that the valve is designed to cope with three separate signals; the two service line pressure signals controlled by the dual foot valve and the hand valve secondary pressure signal.

Supply dump valve (Fig. 12.26(a, b and c)) The purpose of the supply dump valve is to automatically reduce the trailer emergency line pressure to 1.5 bar should the trailer service brake line fail after the next full service brake application within two seconds. This collapse of emergency line pressure signals to the trailer emergency valve to apply the trailer brakes from the trailer reservoir air supply, overriding the driver's response.

12.2.6 Trailer two line brake system (Fig. 12.6)
The difference with the two and three line trailer braking systems is that the two line only has a single control service line, whereas the three line has both a service line and a secondary line.

Control (service) line circuit (Fig. 12.6) On making a brake application, a pressure signal from the tractor control (service) line actuates the relay
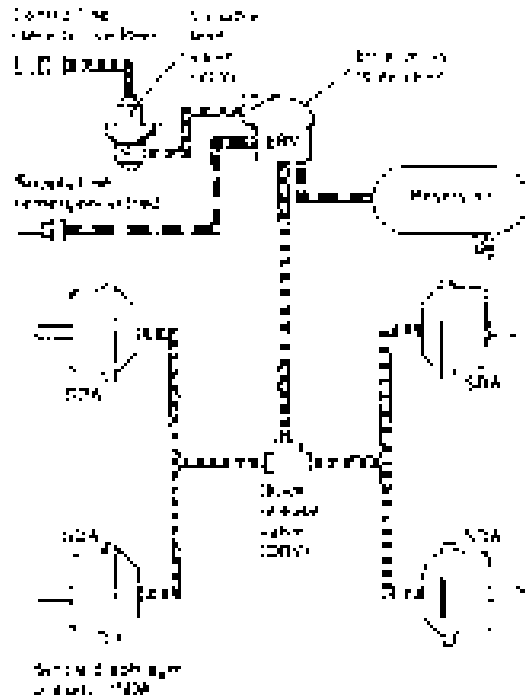
Fig. 12.6 Trailer two line brake system

portion of the emergency relay valve to deliver air pressure from the trailer reservoir to each of the single diaphragm actuator chambers. In order to provide the appropriate braking power according to the trailer payload, a variable load sensing valve is installed in the control line ahead of the emergency relay valve. This valve modifies the control line signal pressure so that the emergency relay valve only supplies the brake actuators with sufficient air pressure to retard the vehicle but not to lock the wheels. A quick-release valve may be included in the brake actuator feed line to speed up the emptying of the actuator chambers to release the brakes but usually the emergency relay valve exhaust valve provides this function adequately. If the supply (emergency) line pressure

drops below a predetermined value, then the emergency portion of the emergency relay valve automatically passes air from the trailer reservoir to the brake actuators to stop the vehicle.

12.3 Air operated power brake equipment

12.3.1 Air dryer (Bendix) (Fig. 12.7(a and b))

Generally, atmospheric air contains water vapour which will precipitate if the temperature falls low enough. The amount of water vapour content of the air is measured in terms of relative humidity. A relative humidity of 100% implies that the air is saturated so that there will be a tendency for the air to condensate. The air temperature and pressure
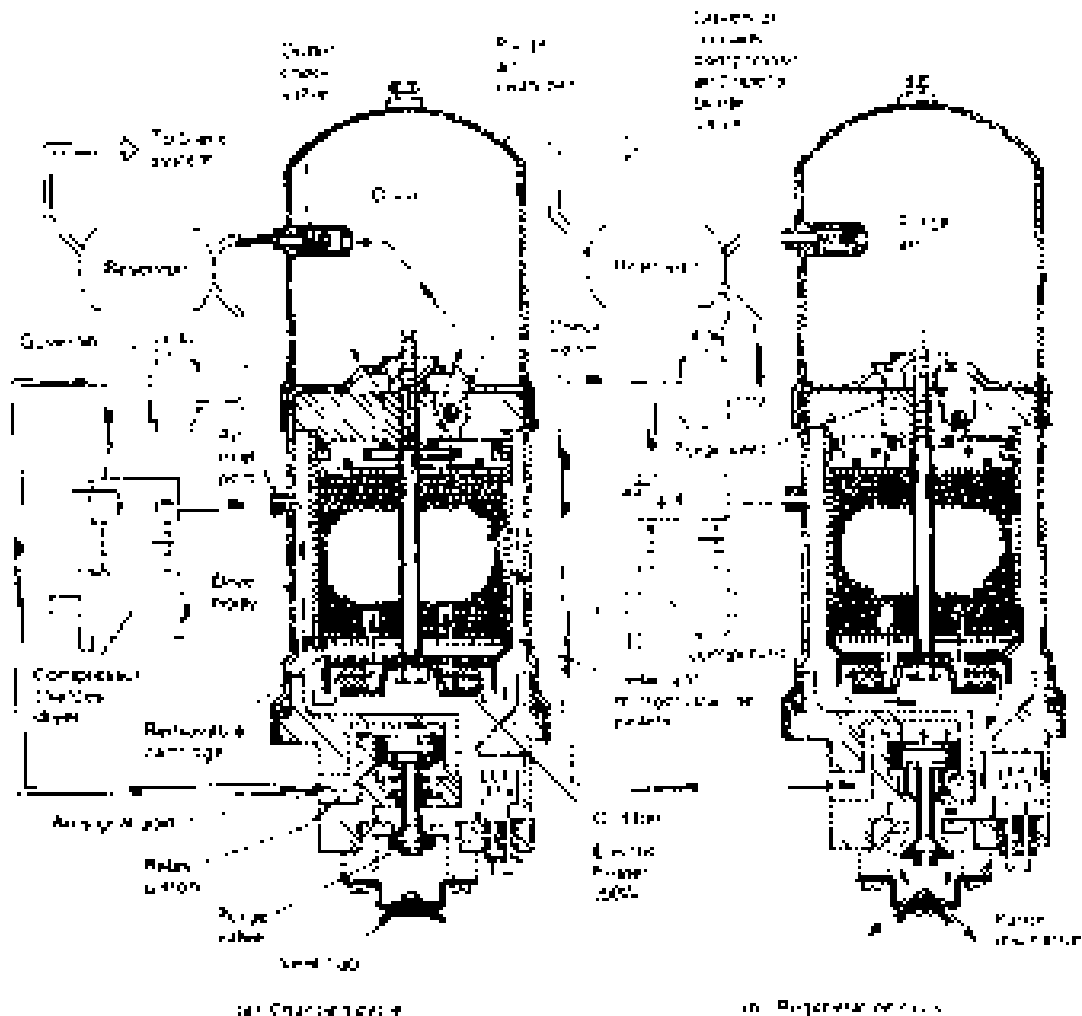
Fig. 12.7 (a and b)   Air dryer (Bendix)

516

determines the proportion of water vapour retained in the air and the amount which condenses.

If the saturation of air at atmospheric pressure occurs when the relative humidity is 100% and the output air pressure from the compressor is 8 bar, that is eight times atmospheric pressure (a typical working pressure), then the compressed air will have a much lower saturation relative humidity equal to $\frac{100}{8} = 12\ 5\%$.

Comparing this 12.5% saturation relative humidity, when the air has been compressed, to the normal midday humidity, which can range from 60% in the summer to over 90% in the winter, it can be seen that the air will be in a state of permanent saturation.

However, the increase in air temperature which will take place when the air pressure rises will raise the relative humidity somewhat before the air actually becomes saturated, but not sufficiently to counteract the lowering of the saturation relative humidity when air is compressed.

The compressed air output from the compressor will always be saturated with water vapour. A safeguard against water condensate damaging the air brake equipment is obtained by installing an air dryer between the compressor and the first reservoir.

The air dryer unit cools, filters and dries all the air supplied to the braking system. The drying process takes place inside a desiccant cartridge which consists of many thousands of small microcrystalline pellets. The water vapour is collected in the pores of these pellets. This process is known as *absorption* There is no chemical change as the pellets absorb and release water so that, provided that the pores do not become clogged with oil or other foreign matter, the pellets have an unlimited life. The total surface area of the pellets is about $464\,000\,\mathrm{m}^2$. This is because each pellet has many minute pores which considerably increase the total surface area of these pellets.

Dry, clean air is advantageous because:

1 the absence of moisture prevents any lubricant in the air valves and actuators from being washed away,
2 the absence of moisture reduces the risk of the brake system freezing,
3 the absence of oil vapour in the airstream caused by the compressor's pumping action extends the life of components such as rubber diaphragms, hoses and 'O' rings,
4 the absence of water and oil vapour prevents sludge forming and material accumulating in the pipe line and restricting the air flow.

Charge cycle (Fig. 12.7(a)) Air from the compressor is pumped to the air dryer inlet port where it flows downwards between the dryer body and the cartridge wall containing the desiccant. This cools the widely but thinly spread air, causing it to condense onto the steel walls and drip to the bottom of the dryer as a mixture of water and oil (lubricating oil from the compressor cylinder walls). Any carbon and foreign matter will also settle out in this phase. The cooled air charge now changes its direction and rises, passing through the oil filter and leaving behind most of the water droplets and oil which were still suspended in the air. Any carbon and dirt which has remained with the air is now separated by the filter.

The air will now pass through the desiccant so that any water vapour present in the air is progressively absorbed into the microcrystalline pellet matrix. The dried air then flows up through both the check valve and purge vent into the purge air chamber. The dryness of the air at this stage will permit the air to be cooled at least 17°C before any more condensation is produced. Finally the air now filling the purge chamber passes out to the check valve and outlet port on its way to the brake system's reservoirs.

Regeneration cycle (Fig. 12.7(b)) Eventually the accumulated moisture will saturate the desiccant, rendering it useless unless the microcrystalline pellets are dried. Therefore, to enable the pellets to be continuously regenerated, a reverse flow of dry air from the purge air chamber is made to occur periodically by the cut-out and cut-in pressure cycle provided by the governor action.

When the reservoir air pressure reaches the maximum cut-out pressure, the governor inlet valve opens, allowing pressurized air to be transferred to the unloader plunger in the compressor cylinder head. At the same time, this pressure signal is transmitted to the purge valve relay piston which immediately opens the purge valve. The accumulated condensation and dirt in the base of the dryer is then rapidly expelled due to the existing air pressure in the lower part of the dryer. The sudden drop in air pressure in the desiccant cartridge chamber allows the upper purge chamber to discharge dry air back through the purge vent into the desiccant cartridge, downwards through the oil filter, finally escaping through the open purge valve into the atmosphere.

During the reverse air flow process, the expanding dry air moves through the desiccant and effectively absorbs the moisture from the crystals on its

way out into the atmosphere. Once the dryer has been purged of condensation and moisture, the purge valve will remain open until the cylinder head unloader air circuit is permitted to exhaust and the compressor begins to recharge the reservoir. At this point the trapped air above the purge relay piston also exhausts, allowing the purge valve to close. Thus with the continuous rise and fall of air pressure the charge and regeneration cycles will be similarly repeated.

A 60 W electric heater is installed in the base of the dryer to prevent the condensation freezing during cold weather.

12.3.2 Reciprocating air compressors

The source of air pressure energy for an air brake system is provided by a reciprocating compressor driven by the engine by either belt, gear or shaft-drive at half engine speed. The compressor is usually base- or flange-mounted to the engine.

To prevent an excessively high air working temperature, the cast iron cylinder barrel is normally air cooled via the enlarged external surface area provided by the integrally cast fins surrounding the upper region of the cylinder barrel. For low to moderate duty, the cylinder head may also be air cooled, but for moderate to heavy-duty high speed applications, liquid coolant is circulated through the internal passages cast in the aluminium alloy cylinder head. The heat absorbed by the coolant is then dissipated via a hose to the engine's own cooling system. The air delivery temperature should not exceed 220°C.

Lubrication of the crankshaft plain main and big-end bearings is through drillings in the crankshaft, the pressurized oil supply being provided by the engine's lubrication system, whereas the piston and rings and other internal surfaces are lubricated by splash and oil mist. Surplus oil is permitted to drain via the compressor's crankcase back to the engine's sump. The total cylinder swept volume capacity needed for an air brake system with possibly auxiliary equipment for light, medium and heavy commercial vehicles ranges from about $150 \, cm^3$ to $500 \, cm^3$, which is provided by either single or twin cylinder reciprocating compressor. The maximum crankshaft speed of these compressors is anything from 1500 to 3000 rev/min depending upon maximum delivery air pressure and application. The maximum air pressure a compressor can discharge continuously varies from 7 to 11 bar. A more typical maximum pressure value would be 9 bar.

The quantity of air which can be delivered at maximum speed by these compressors ranges from 150 L/min to 500 L/min for a small to large size compressor. This corresponds to a power loss of something like 1.5 kW to 6 kW respectively.

Compressor operation When the crankshaft rotates, the piston is displaced up and down causing air to be drawn through the inlet port into the cylinder on the down stroke and the same air to be pushed out on the upward stroke through the delivery port. The unidirectional flow of the air supply is provided by the inlet and delivery valves. The suction and delivery action of the compressor may be controlled by either spring loaded disc valves (Fig. 12.9) or leaf spring (reed) valves (Fig. 12.8). For high speed compressors the reed type valve arrangements tend to be more efficient.

On the downward piston stroke the delivery valve leaf flattens and closes, thus preventing the discharged air flow reversing back into the cylinder (Fig. 12.8). At the same time the inlet valve is drawn away from its seat so that fresh air flows through the valve passage in its endeavour to fill the expanding cylinder space.

On the upward piston stroke the inlet valve leaf is pushed up against the inlet passage exit closing the valve. Consequently the trapped pressurized air is forced to open the delivery valve so that the air charge is expelled through the delivery port to the reservoir.

The sequence of events is continuous with a corresponding increase in the quantity of air delivered and the pressure generated.

The working pressure range of a compressor may be regulated by either an air delivery line mounted unloader valve (Figs 12.10 and 12.11) or an integral compressor unloader mechanism controlled by an external governor valve (Fig. 12.9). A further feature which is offered for some applications is a multiplate clutch drive which reduces pumping and frictional losses when the compressor is running light (Fig. 12.8).

Clutch operation (ig. 12.8) With the combined clutch drive compressor unit, the compressor's crankshaft can be disconnected from the engine drive once the primary reservoir has reached its maximum working pressure and the compressor is running light to reduce the wear of the rotary bearings and reciprocating piston and rings and to eliminate the power consumed in driving the compressor.

The clutch operates by compressed air and is automatically controlled by a governor valve similar to that shown in Fig. 12.9.
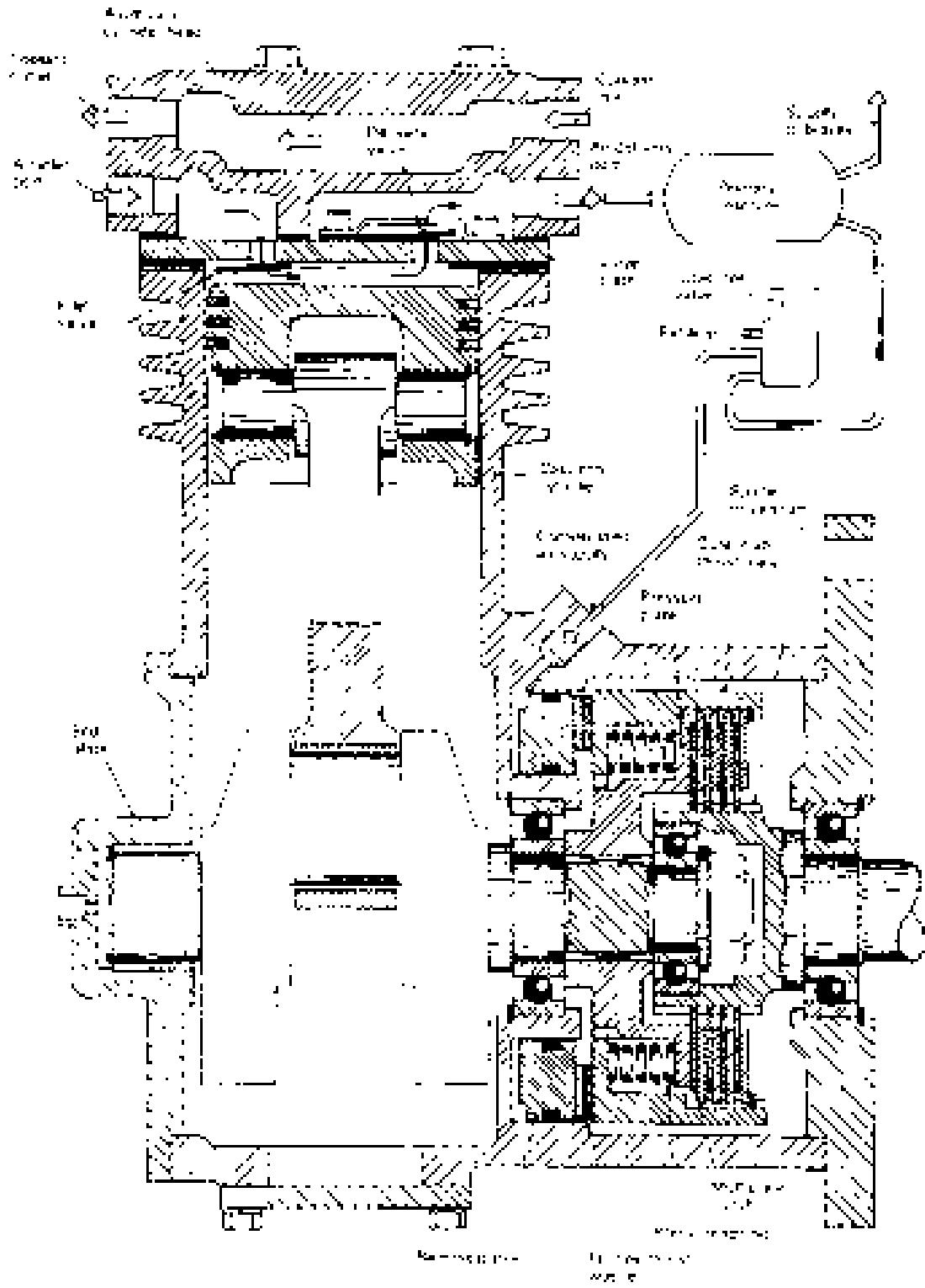
Fig. 12.8 Single cylinder air compressor with clutch drive

The multiplate clutch consists of four internally splined sintered bronze drive plates sandwiched between a pressure plate and four externally splined steel driven plates (Fig. 12.8). The driven plates fit over the enlarged end of the splined input shaft, whereas the driven plates are located inside the internally splined clutch outer hub thrust plate. The friction plate pack is clamped together by twelve circumferentially evenly spaced compression springs which react between the pressure plate and the outer hub thrust plate. Situated between the air release piston and the outer hub thrust plate are a pair of friction thrust washers which slip when the clutch is initially disengaged.

When the compressor air delivery has charged the primary reservoir to its preset maximum, the governor valve sends a pressure signal to the clutch air release piston chamber. Immediately the friction thrust washers push the clutch outer hub thrust plate outwards, causing the springs to become compressed so that the clamping pressure between the drive and driven plates is relaxed. As a result, the grip between the plates is removed. This then enables the crankshaft, pressure plate, outer hub thrust plate and the driven plates to rapidly come to a standstill.

As the air is consumed and exhausted by brake or air equipment application, the primary reservoir pressure drops to its lower limit. At this point the governor exhausts the air from the clutch release piston chamber and consequently the pressure springs are free to expand, enabling the drive and driven plates once again to be squeezed together. By these means the engagement and disengagement of the compressor's crankshaft drive is automatically achieved.

12.3.3 Compressor mounted unloader with separate governor (Fig. 12.9(a and b))

Purpose   The governor valve unit and the unloader plunger mechanism control the compressed air output which is transferred to the reservoir by causing the compressor pumping action to 'cut-out' when the predetermined maximum working pressure is attained. Conversely, as the stored air is consumed, the reduction in pressure is sensed by the governor which automatically causes the compressor to 'cut-in', thus restarting the delivery of compressed air to the reservoir and braking system again.

Operation

Compressor charging (Fig. 12.9(a))   During the charging phase, air from the compressor enters the reservoir, builds up pressure and then passes to the braking system (Fig. 12.9(a)). A small sample of air from the reservoir is also piped to the underside of the governor piston via the governor inlet port.

When the pressure in the reservoir is low, the piston will be in its lowest position so that there is a gap between the plunger's annular end face and the exhaust disc valve. Thus air above the unloader plunger situated in the compressor's cylinder head is able to escape into the atmosphere via the governor plunger tube central passage.

Compressor unloaded (Fig. 12.9(b))   As the reservoir pressure rises the control spring is compressed lifting the governor piston until the exhaust disc valve contacts the plunger tube, thereby closing the exhaust valve. A further air pressure increase from the reservoir will lift the piston seat clear of the inlet disc valve. Air from the reservoir now flows around the inlet disc valve and plunger tube. It then passes though passages to the unloader plunger upper chamber. This forces the unloader plunger down, thus permanently opening the inlet disc valve situated in the compressor's cylinder head (Fig. 12.9(b)). Under these conditions the compressor will draw in and discharge air from the cylinder head inlet port, thereby preventing the compressor pumping and charging the reservoir any further. At the same time, air pressure acts on the annular passage area around the governor plunger stem. This increases the force pushing the piston upwards with the result that the inlet disc valve opens fully. When the brakes are used, the reservoir pressure falls and, when this pressure reduction reaches 1 bar, the control spring downward force will be sufficient to push down the governor piston and to close the inlet disc valve initially.

Instantly the reduced effective area acting on the underside of the piston allows the control spring to move the piston down even further until the control exhaust valve (tube/disc) opens. Compressed air above the unloader plunger will flow back to the governor unit, enter the open governor plunger tube and exhaust into the atmosphere. The unloader plunger return spring now lifts the plunger clear of the cylinder head inlet disc, permitting the compressor to commence charging the reservoir.

The compressor will continue to charge the system until the cut-out pressure is reached and once again the cycle will be repeated.
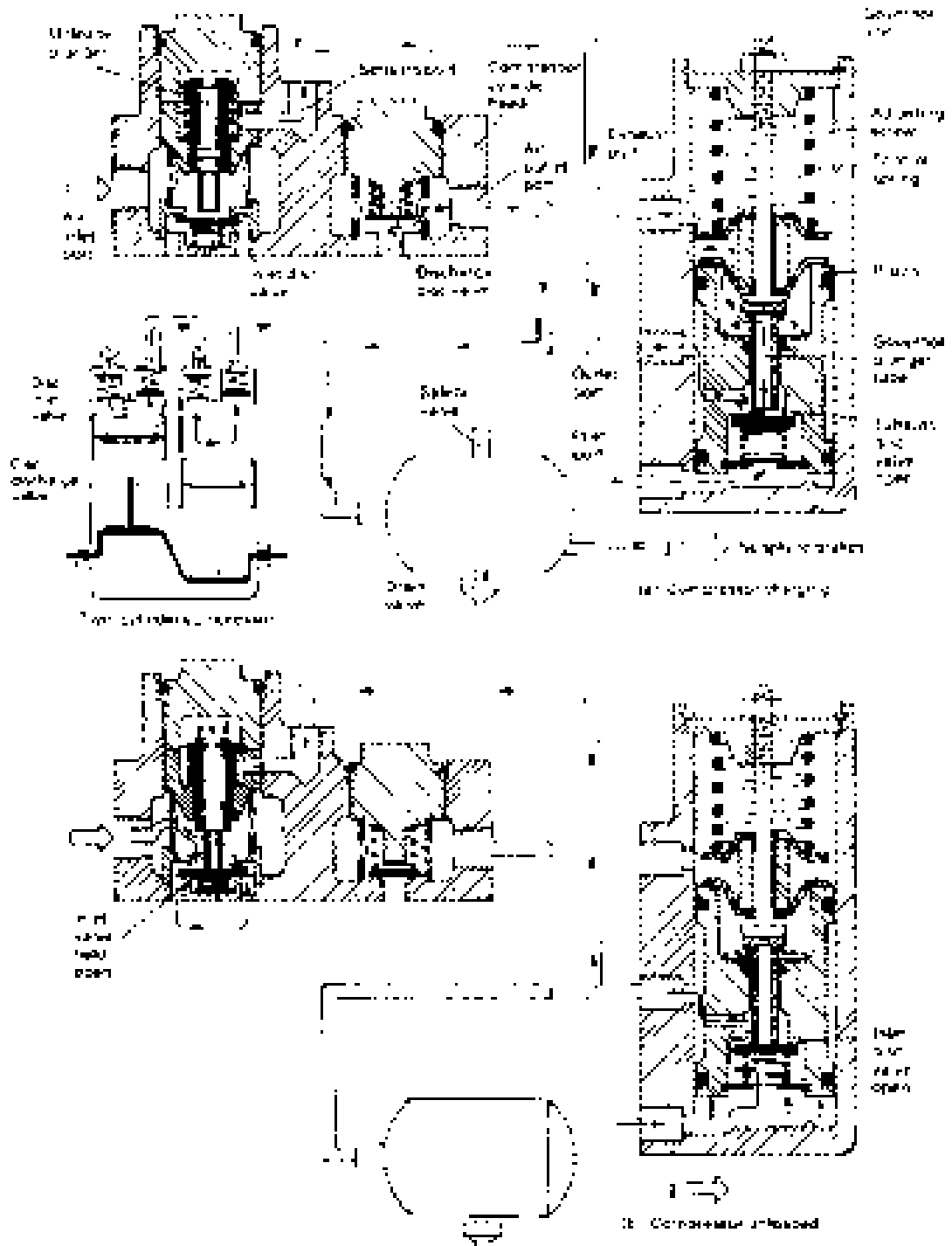
Fig. 12.9 Compressor mounted unloader with separate governor

12.3.4 Unloader valve (diaphragm type)
(Fig. 12.10(a and b))

Compressor charging (Fig. 12.10(a))   When air is initially pumped from the compressor to the reservoir, the unloader valve unit non-return valve opens and air passes from the inlet to the outlet port. At the same time, air flows between the neck of the exhaust valve and the shoulder of the relay valve piston, but since they both have the same cross-sectional area, the force in each direction is equalized. Therefore, the relay piston return spring is able to keep the exhaust valve closed. Air will also move through a passage on the reservoir side of the non-return valve to the chamber on the plunger side of the diaphragm.

Compressor unloaded (Fig. 12.10(b))   As the reservoir pressure rises, the diaphragm will move against the control spring until the governor plunger has shifted sufficiently for the exhaust valve to close (Fig. 12.10(b)). Further pressure build-up moves the diaphragm against the control spring so that the end of the plunger enters its bore and opens the inlet valve. The annular end face of the

plunger will also be exposed to the air pressure, so that the additional force produced fully opens the inlet valve. Air now passes through the centre of the plunger and is directed via a passage to the head of the relay piston.

Eventually a predetermined maximum cut-out pressure is reached, at which point the air pressure acting on the relay piston crown overcomes the relay return spring, causing the relay exhaust valve to open, expelling the compressed air into the atmosphere. This enables the compressor to operate under no-load conditions while the reservoir and braking system is sufficiently charged.

Compressor commences charging (Fig. 12.10 (a and b))   As the stored air is consumed during a braking cycle, the pressure falls until the cut-in point (minimum safe working pressure) is reached. At this point the control spring force equals and exceeds the opposing air pressure force acting on the diaphragm on the plunger side. The diaphragm and plunger will therefore tend to move away from the control spring until the plunger stem closes the inlet valve. Further plunger movement pushes the exhaust valve open so that trapped air in the relay
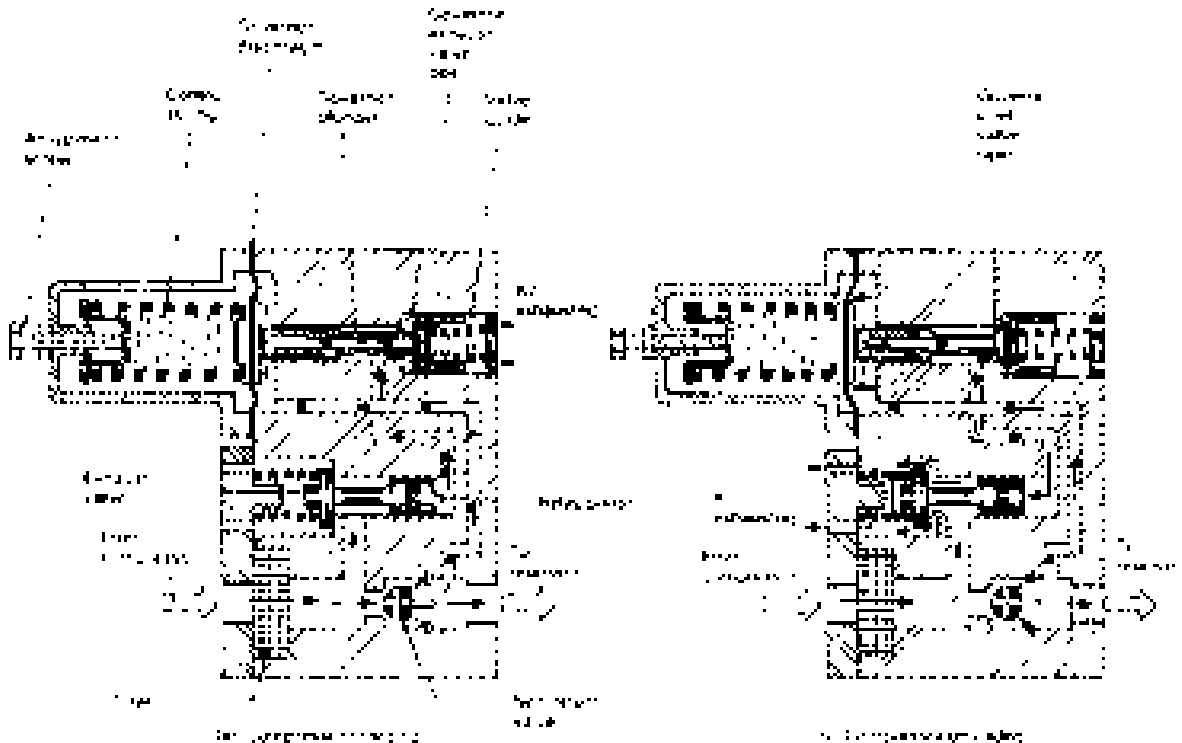
Fig. 12.10 (a and b)   nloader valve (diaphragm type)

piston crown chamber is able to escape to the atmosphere. The relay piston return spring closes the relay exhaust valve instantly so that compression of air again commences, permitting the reservoir to recharge to the pressure cut-out setting.

12.3.5 Unloader valve (piston type)
(Fig. 12.11(a and b))

Purpose The unloader valve enables the compressor to operate under no-load conditions, once the reservoir is fully charged, by automatically discharging the compressor's output into the atmosphere, and to reconnect the compressor output to the reservoir once the air pressure in the system drops to some minimum safe working value.

Operation

Compressor charging (Fig. 12.11(a)) When the compressor starts to charge, air will flow to the reservoir by way of the horizontal passage between the inlet and outlet ports.

The chamber above the relay piston is vented to the atmosphere via the open outlet pilot valve so that the return spring below the relay piston is able to keep the exhaust valve closed, thus permitting the reservoir to become charged.

Compressor unloaded (Fig. 12.11(b)) As the reservoir pressure acting on the right hand end face of the pilot piston reaches a maximum (cut-out setting), the pilot piston pushes away from its inlet seat. A larger piston area is immediately exposed to the air pressure, causing the pilot piston to rapidly move over to its outlet seat, thereby sealing the upper relay piston chamber atmospheric vent. Air will now flow along the space made between the pilot piston and its sleeve to act on the upper face of the relay piston. Consequently, the air pressure on both sides of the relay piston will be equalized momentarily. Air pressure acting down on the exhaust valve overcomes the relay piston return spring force and opens the compressor's discharge to the atmosphere. The exhaust valve will then be held fully open by the air pressure acting on the upper face of the relay piston. Compressed air from the compressor will be pumped directly to the atmosphere and so the higher pressure on the reservoir side of the non-return valve forces it to close, thereby preventing the stored air in the reservoir escaping.

Compressor commences charging (Fig. 12.11(a and b)) As the air pressure in the reservoir is discharged and lost to the atmosphere during brake applications the reservoir pressure drops. When the pressure has been reduced by approximately one bar below the cut-out setting (maximum pressure), the control spring overcomes the air pressure acting on the right hand face of the pilot piston, making it shift towards its inlet seat. The pilot piston outlet

Fig. 12.11 (a and b) nloader valve (piston type)

523
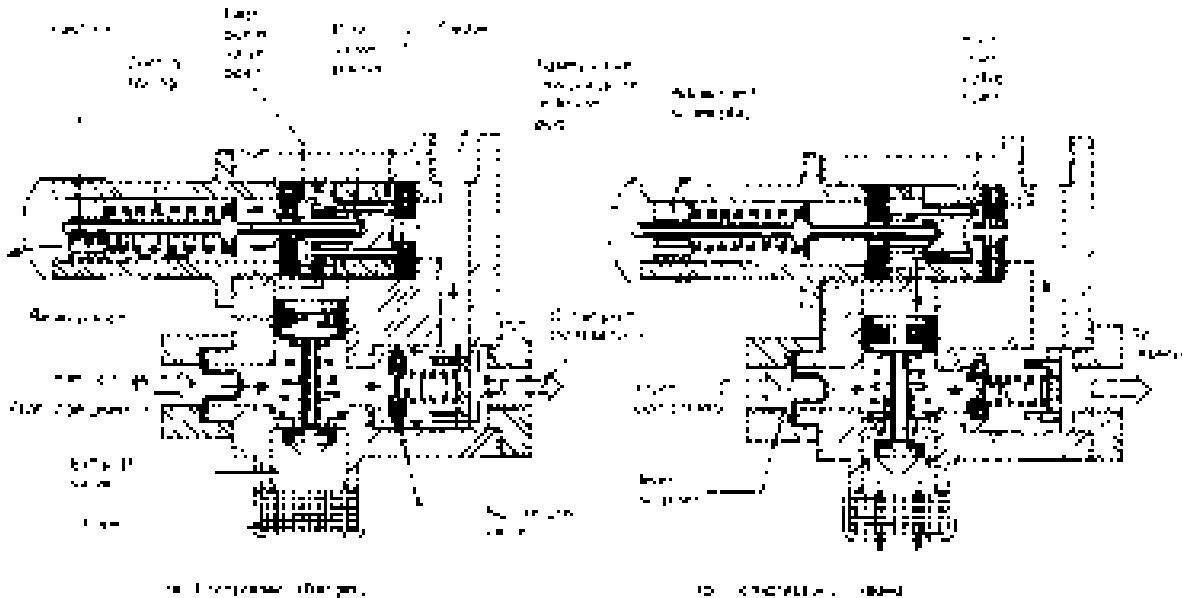
valve opens, causing the air pressure above the relay piston to escape to the atmosphere which allows the relay piston return spring to close the exhaust valve. The discharged air from the compressor will now be redirected to recharge the reservoir.

The difference between the cut-out and cut-in pressures is roughly one bar and it is not adjustable, but the maximum (cut-out) pressure can be varied over a wide pressure range by altering the adjustment screw setting.

12.3.6 Single- and multi-circuit protection valve (Fig. 12.12a)

Purpose Circuit protection valves are incorporated in the brake charging system to provide an independent method of charging a number of reservoirs to their operating minimum. Where there is a failure in one of the reservoir circuits, causing loss of air, they will isolate the affected circuit so that the remaining circuits continue to function.

Single element protection valve (Fig. 12.12(a)) When the compressor is charging, air pressure is delivered to the supply port where it increases until it is able to unseat the non-return disc valve against the closing force of the setting spring. Air will now pass between the valve disc and its seat before it enters the delivery port passage on its way to the reservoir. A larger area of the disc valve is now exposed to air pressure which forces the disc valve and piston to move further back against the already compressed setting spring. As the charging pressure in the reservoir increases, the air thrust on the disc and piston face also rises until it eventually pushes back the valve to its fully open position.

When the air pressure in the reservoir reaches its predetermined maximum, the governor or unloader valve cuts out the compressor. The light return spring around the valve stem, together with air pressure surrounding the disc, now closes the non-return valve, thereby preventing air escaping back through the valve. Under these conditions, the trapped air pressure keeps the disc valve on its seat and holds the setting spring and piston in the loaded position, away from the neck of the valve stem. As air is consumed from the reservoir, its pressure drops so that the compressor is signalled to cut in again (restarting pumping). The pressure on the compressor side of the non-return valve then builds up and opens the valve, enabling the reservoir to recharge.
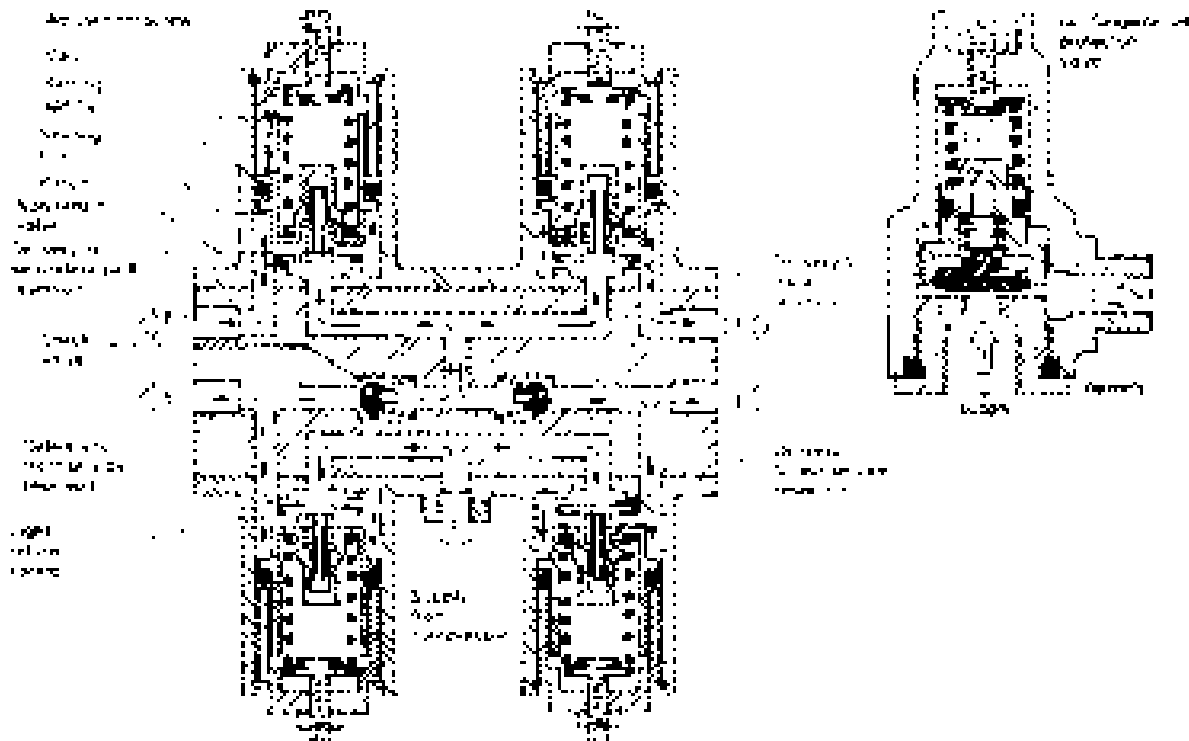
Fig. 12.12 uadruple circuit protection valve

524

Should the air pressure in one of the reservoir systems drop roughly 2.1 bar or more, the setting spring stiffness overcomes the air pressure acting on the piston so that it moves against the disc valve to close the inlet passage. The existing air pressure stored in the reservoir will still impose a thrust against the piston, but because the valve face area exposed to the charge pressure is reduced by the annular seat area and is therefore much smaller, a pressure increase of up to 1.75 bar may be required to re-open the valve.

A total loss of air from one reservoir will automatically cause the setting spring of the respective protection valve to close the piston against the non-return valve.

Multi-element protection valve (Fig. 12.12) Multi-element protection valves are available in triple and quadruple element form. Each element contains the cap, piston, setting spring and non-return valve, similar to the single element protection valve.

Charging air from the compressor enters the supply port of the multi-element protection valve, increasing the pressure on the inlet face of the first and second valve element and controlling the delivery to the front and rear service reservoirs respectively. When the predetermined setting pressure is reached, both element non-return valves open, permitting air to pass through the valve to charge both service reservoirs.

The protection valves open and close according to the governor or unloader valve cutting in or cutting out the pumping operation of the compressor.

Internal passages within the multi-element valve body, protected by two non-return valves, connect the delivery from the first and second valve elements to the inlet of the third and fourth valve elements, which control the delivery to the secondary/park and the trailer reservoir supplies respectively. Delivery to the third and fourth valve elements is fed from the reservoir connected to the first and second valve element through passages within the body.

The additional check valves located in the body of the multi-protection valve act as a safeguard against cross-leakage between the front and rear service reservoirs. Failure of the front reservoir or circuit still permits the rear service reservoir to supply the third and fourth element valve. Alternatively, if the rear service reservoir should fail, the front service reservoir can cope adequately with delivering air charge to the third and fourth reservoir.

12.3.7 Pressure reducing valve (piston type) (Fig. 12.13(a, b and c))

Various parts of an air brake system may need to operate at lower pressures than the output pressure delivered to the reservoirs. It is therefore the function of the pressure reducing valve to decrease, adjust and maintain the air line pressure within some predetermined tolerance.
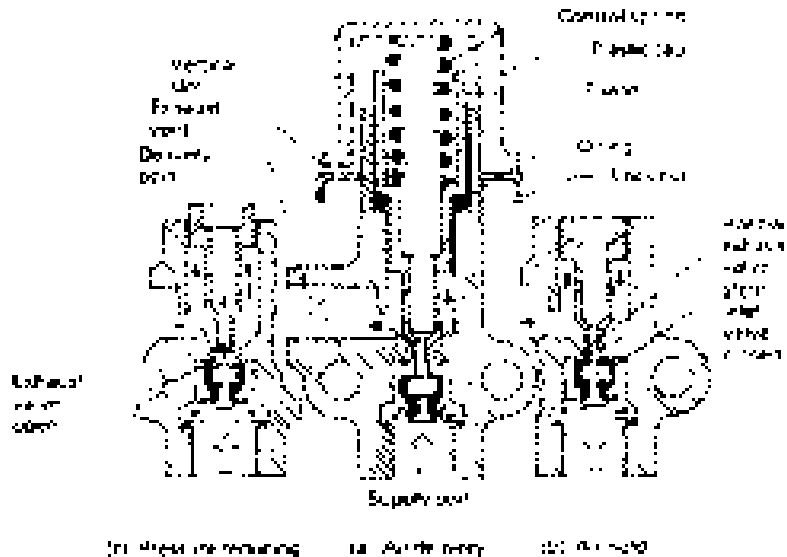
Fig. 12.13 (a–c) Pressure reducing valve (piston type)

Operation When the vehicle is about to start a journey, the compressor charges the reservoirs and air will flow through the system to the various components. Initially, air flows through to the pressure reducing valve supply port through the open inlet valve and out to the delivery port (Fig. 12.13(a)). As the air line pressure approaches its designed working value, the air pressure underneath the piston overcomes the stiffness of the control spring and lifts the piston sufficiently to close the inlet valve and cut off the supply of air passing to the brake circuit it supplies (Fig. 12.13(b)).

If the pressure in the delivery line exceeds the predetermined pressure setting of the valve spring, the extra pressure will lift the piston still further until the hollow exhaust stem tip is lifted clear of its seat. The surplus of air will now escape through the central exhaust valve stem into the hollow piston chamber where it passes out into the atmosphere via the vertical slot on the inside of the adjustable pressure cap (Fig. 12.13(c)). Delivery line air will continue to exhaust until it can no longer support the control spring. At this point, the spring pushes the piston down and closes the exhaust valve. After a few brake applications, the delivery line pressure will drop so that the control spring is able to expand further, thereby unseating the inlet valve. Hence the system is able to be recharged.

12.3.8 Non-return (check) valve (Fig. 12.14(a))

Purpose A non-return valve, sometimes known as a *check valve* is situated in an air line system where it is necessary for the air to flow in one direction only. It is the valve's function therefore not to restrict the air flow in the forward direction, but to prevent any air movement in the reverse or opposite direction.

Operation (Fig. 12.14(a)) When compressed air is delivered to a part of the braking system via the non-return valve, the air pressure forces the spherical valve (sometimes disc) head of its seat against the resistance of the return spring. Air is then permitted to flow almost unrestricted through the valve. Should the air flow in the forward direction cease or even reverse, the return spring quickly closes to prevent air movement in the opposite direction occurring.

12.3.9 Safety valve (Fig. 12.14(b))

Purpose To protect the charging circuit of an air braking system from excessive air pressure, safety valves are incorporated and mounted at various positions in the system, such as on the compressor cylinder head, on the charging reservoir or in the pipe line between the compressor and reservoir.

Operation (Fig. 12.14(b)) If an abnormal pressure surge occurs in the charging system, the rise in air pressure will be sufficient to push the ball valve back against the regulating spring. The unseated ball now permits the excess air pressure to escape into the atmosphere. Air will exhaust to the atmosphere until the pressure in the charging system has been reduced to the blow-off setting determined by the initial spring adjustment. The regulating spring then forces the ball valve to re-seat so that no more air is lost from the charging system.
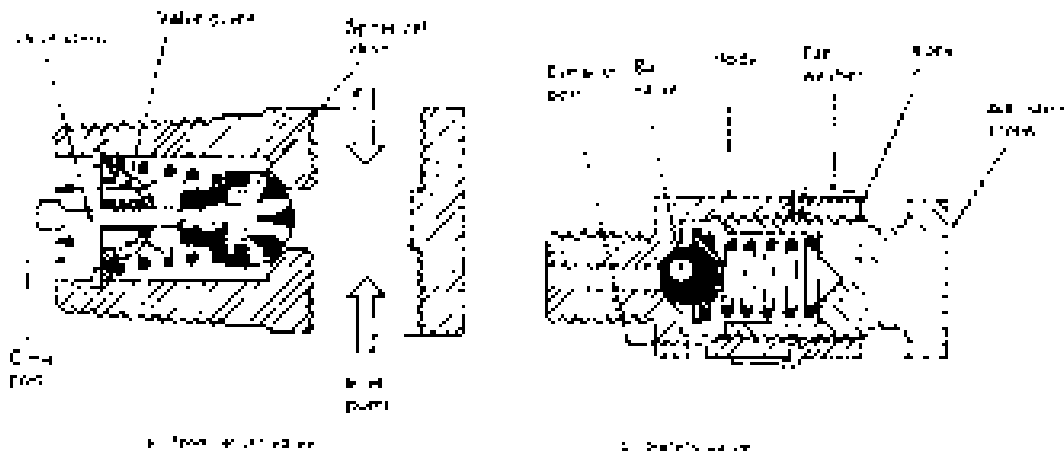
Fig. 12.14 (a and b) Non-return and safety valves

12.3.10 Dual concentric foot control valve
(Fig. 12.15(a and b))

Purpose The foot control valve regulates the air pressure passing to the brake system from the reservoir according to the amount the foot treadle is depressed. It also imparts a proportional reaction to the movement of the treadle so that the driver experiences a degree of brake application.

Operation

Applying brakes (Fig. 12.15(a)) Depressing the foot treadle applies a force through the graduating springs to the pistons, causing the exhaust hollow stem seats for both pistons to close the inlet/exhaust valves. With further depression of the foot pedal, the piston simultaneously unseats the inlet/exhaust valves and compressed air from the reservoirs passes through the upper and lower valves to the front and rear brake actuators respectively (or to the tractor and trailer brake actuators respectively).

Balancing (Fig. 12.15(a and b)) With the compressed air passing to the brake actuator chambers, air pressure is built up beneath the upper and lower pistons. Eventually the upthrust created by this air pressure equals the downward spring force; the pistons and valve carrier lift and the inlet valves close, thus interrupting the compressed air supply to the brake actuators. At the same time, the exhaust valves remain closed. The valves are then in a balanced condition with equal force above and
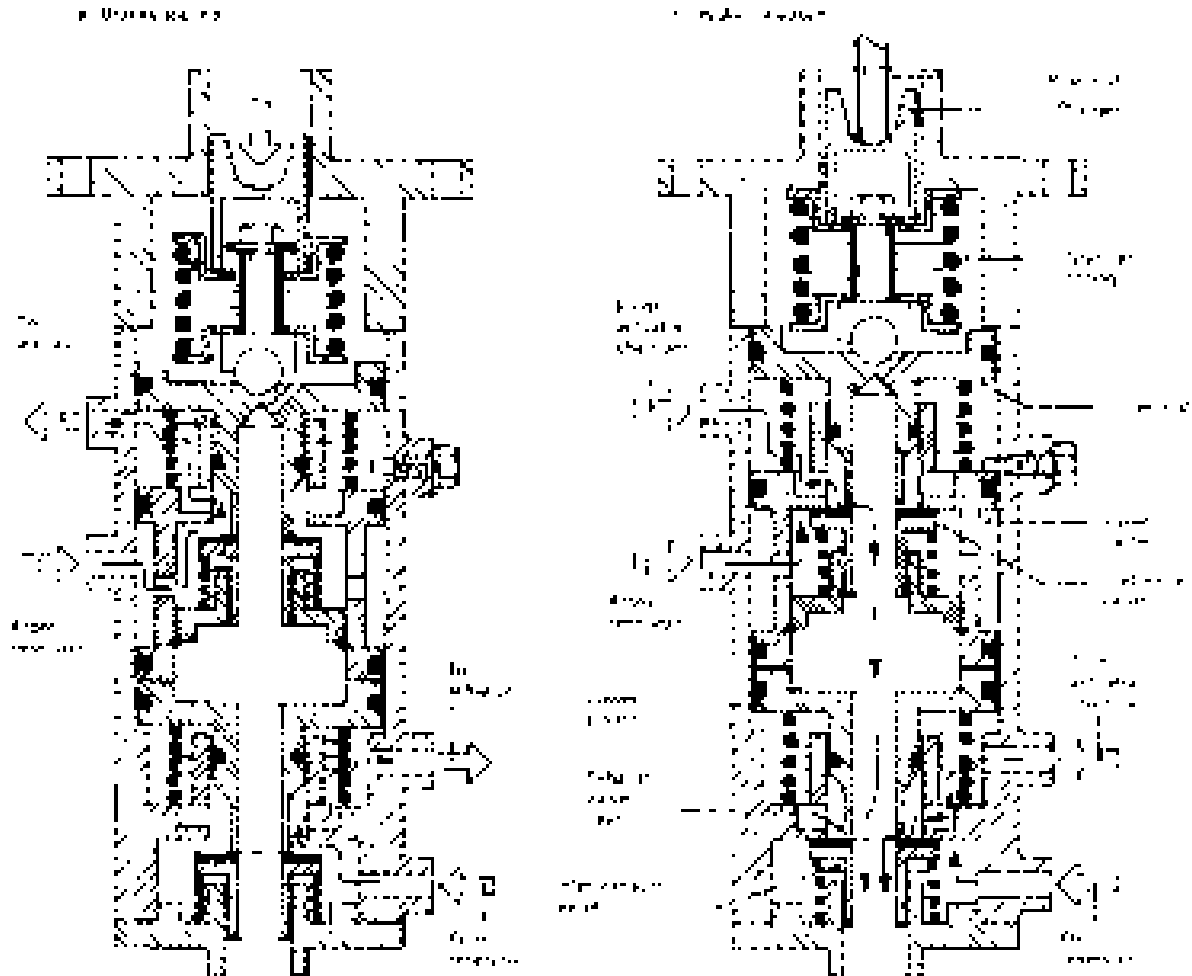
Fig. 12.15 (a and b) Dual concentric foot valve

527

below the upper piston and with equal air pressure being held in both halves of the brake line circuits.

Pushing the treadle down still further applies an additional force on top of the graduating spring. There will be a corresponding increase in the air pressure delivered and a new point of balance will be reached.

Removing some of the effort on the foot treadle reduces the force on top of the graduating spring. The pistons and valve carrier will then lift due to the air pressure and piston return springs. When this occurs the inlet valves remain closed and the exhaust valves open to exhausting air pressure from the brake actuators until a state of balance is obtained at lower pressure.

Releasing brakes (Fig. 12.15(b)) Removing the driver's force from the treadle allows the upper and lower piston and the valve carrier to rise to the highest position. This initially causes the inlet/exhaust valves to close their inlet seats, but with further upward movement of the pistons and valve

assembly both exhaust valves open. Air from both brake circuits will therefore quickly escape to the atmosphere thus fully releasing the brakes.

12.3.11 Dual delta series foot control valve
(Fig. 12.16)

Purpose The delta series of dual foot valves provide the braking system with two entirely separate foot controlled air valve circuits but which operate simultaneously with each other. Thus, if one half of the dual foot valve unit should develop a fault then the balance beam movement will automatically ensure that the other half of the twin valve unit continues to function.

Operation

Brakes released (12.16(a)) When the brakes are released, the return springs push up the piston, graduating spring and plunger assemblies for each half valve unit. Consequently the inlet disc valves close and the control tube shaped exhaust valves
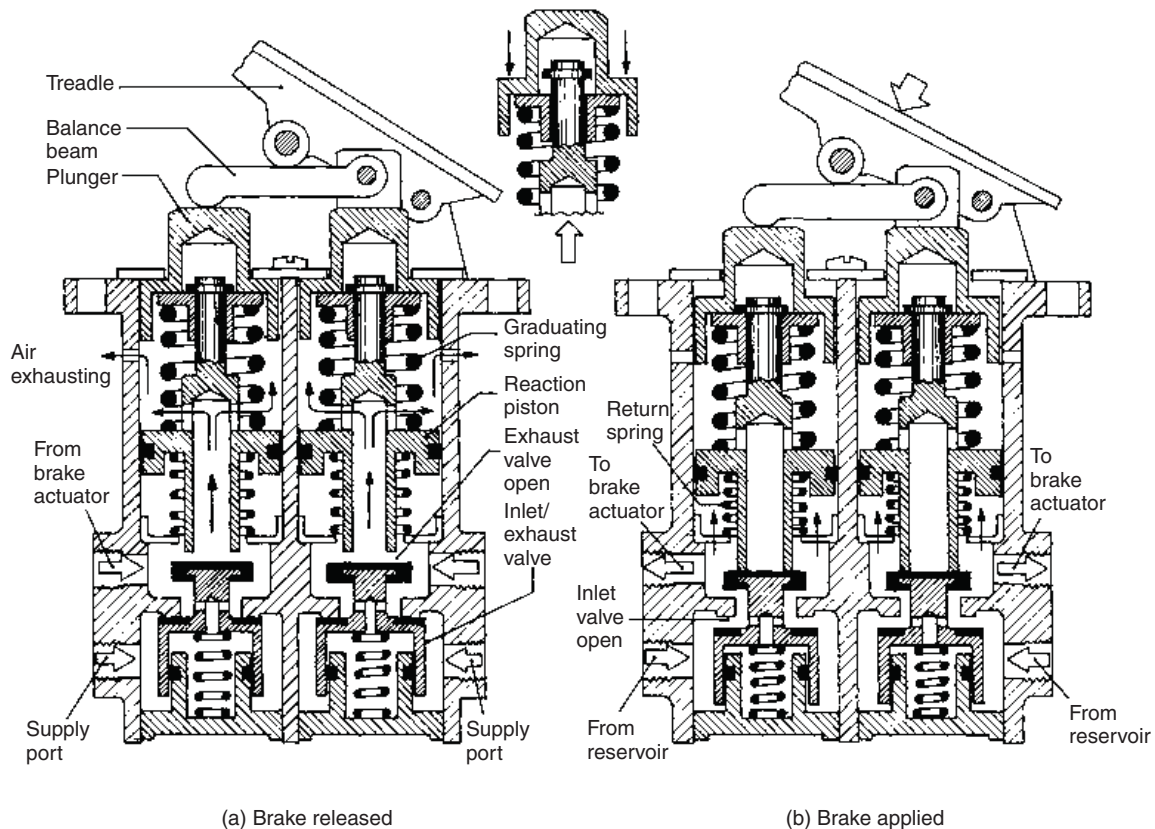
Fig. 12.16 (a and b)   Dual delta foot control valve

528

open. This permits air to exhaust through the centre of the piston tube, upper piston chamber and out to the atmosphere.

Brakes applied (12.16(b))   When the foot treadle is depressed, a force is applied centrally to the balance beam which then shares the load between both plunger spring and piston assemblies. The downward plunger load initially pushes the piston tubular stem on its seat, closing the exhaust disc valve, and with further downward movement unseats and opens the inlet disc valve. Air from the reservoirs will now enter the lower piston chambers on its way to the brake actuators via the delivery ports.

As the air pressure builds up in the lower piston chambers it will oppose and compress the graduating springs until it eventually closes the inlet valve. The valve assembly is then in a lapped or balanced position where both exhaust and inlet valves are closed. Only when the driver applies an additional effort to the treadle will the inlet valve again open to allow a corresponding increase in pressure to pass through to the brake actuator.

The amount the inlet valve opens will be proportional to the graduating spring load, and the pressure reaching the brake actuator will likewise depend upon the effective opening area of the inlet valve. Immediately the braking effort to the foot treadle is charged, a new state of valve lap will exist so that the braking power caused by the air operating on the wheel brake actuator will be progressive and can be sensed by the driver by the amount of force being applied to the treadle. When the driver reduces the foot treadle load, the inlet valve closes and to some extent the exhaust valve will open, permitting some air to escape from the actuator to the atmosphere via central tube passages in the dual piston tubes. Thus the graduating spring driver-controlled downthrust and the reaction piston air-controlled upthrust will create a new state of valve lap and a corresponding charge to the braking power.

12.3.12 *Hand control valve* (Fig. 12.17(a and b))

Purpose   These valves are used to regulate the secondary brake system on both the towing tractor
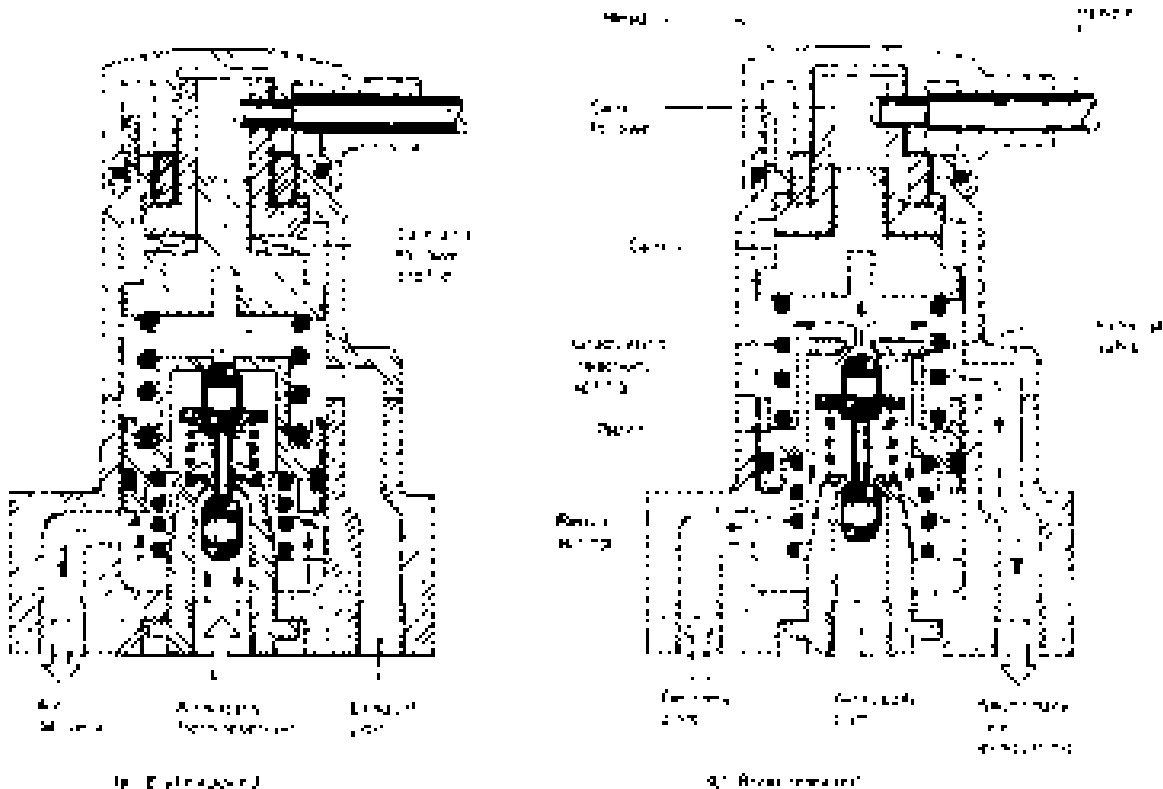
Fig. 12.17 (a and b)   Hand control valve

and on the trailer. Usually only the tractor front axle has secondary braking to reduce the risk of a jack-knife during heavy emergency braking.

Operation

Applying brakes (Fig. 12.17(a)) Swivelling the handle from the released position enables the cam follower to slide over the matching inclined cam profile, thereby forcing the cam plate downwards against the graduating (reaction) spring. The stiffening of the reaction spring forces the piston to move downwards until the exhaust valve passage is closed. Further downward movement of the piston unseats the inlet valve, permitting compressed air from the reservoir to flow through the valve underneath the piston and out of the delivery port, to the front brake actuator and to the trailer brake actuator via the secondary line (blue) coupling to operate the brakes.

Balancing (Fig. 12.17(a and b)) The air supply passing through the valve gradually builds up an opposing upthrust on the underside of the piston until it eventually overcomes the downward force caused by the compressed reaction spring. Subsequently the piston lifts, causing the inlet valve to close so that the compressed air supply to the brake actuators is interrupted. The exhaust valve during this phase still remains seated, thereby preventing air exhaustion. With both inlet and exhaust valves closed, the system is in a balanced condition, thus the downward thrust of the spring is equal to the upthrust of the air supply and the predetermined air pressure established in the brake actuators.

Rotating the handle so that the reaction spring is further compressed, opens the inlet valve and admits more air at higher pressure, producing a new point of balance.

Partially rotating the handle back to the released position reduces some of the reaction spring downward thrust so that the existing air pressure is able to raise the piston slightly. The raised piston results in the inlet valve remaining seated, but the exhaust valve opens, permitting a portion of the trapped air inside the brake actuator to escape into the atmosphere. Therefore the pressure underneath the piston will decrease until the piston upthrust caused by the air pressure has decreased to the spring downthrust acting above the piston. Thus a new state of balance again is reached.

Releasing brakes (Fig. 12.17(b)) Returning the handle to the released position reduces the downward load of the reaction spring to fully raise the piston. As a result, the inlet valve closes and the exhaust valve is unseated, so that the air pressure in the brake actuator chambers collapses as the air is permitted to escape to the atmosphere.

12.3.13 Spring brake hand control valve
(Fig. 12.18(a, b and c))

Purpose This hand control valve unit has two valve assemblies which, due to the cam profile design, is able to simultaneously deliver an 'upright' and an 'inverse' pressure. The valve unit is designed to provide pressure signals via the delivery of small volumes of air to the tractor spring brakes and the trailer's conventional diaphragm actuators. The required full volume of air is then able to pass from the secondary/park reservoir to the brake actuators via the relay valves to apply or release the brakes.

Operation

Spring brake release (Fig. 12.18(a)) When the vehicle is in motion with the brakes released, the upright valve assembly inlet valve is closed and the exhaust valve is unseated, permitting all the air in the trailer brake actuators to be expelled. Conversely the inverse valve assembly delivers a signal pressure to the spring brake relay valve. This results in the line from the secondary/park reservoir to the tractor spring brake actuators to be open. Thus a large volume of air will be delivered to the air chambers controlling the compression of the power springs and the releasing of the tractor brakes.

Secondary brake application (Fig. 12.18(b)) As the handle is moved across the gate to make a secondary brake application it rotates the cam, depressing the upright plunger. The exhaust valve closes and the inlet valve is unseated, causing compressed air to pass to the trailer brake actuator chambers. As the pressure in the brake actuators increases, the air pressure acting on top of the upright piston causes it to move down against the upthrust exerted by the graduating spring, closing the inlet valve. This procedure is repeated for further handle movement until the full secondary brake position is reached when the air pressure delivered to the trailer brake chamber is at a maximum.

During this operation the inverse valve assembly, which was delivering maximum pressure when the handle was in the 'off' position, is exhausting
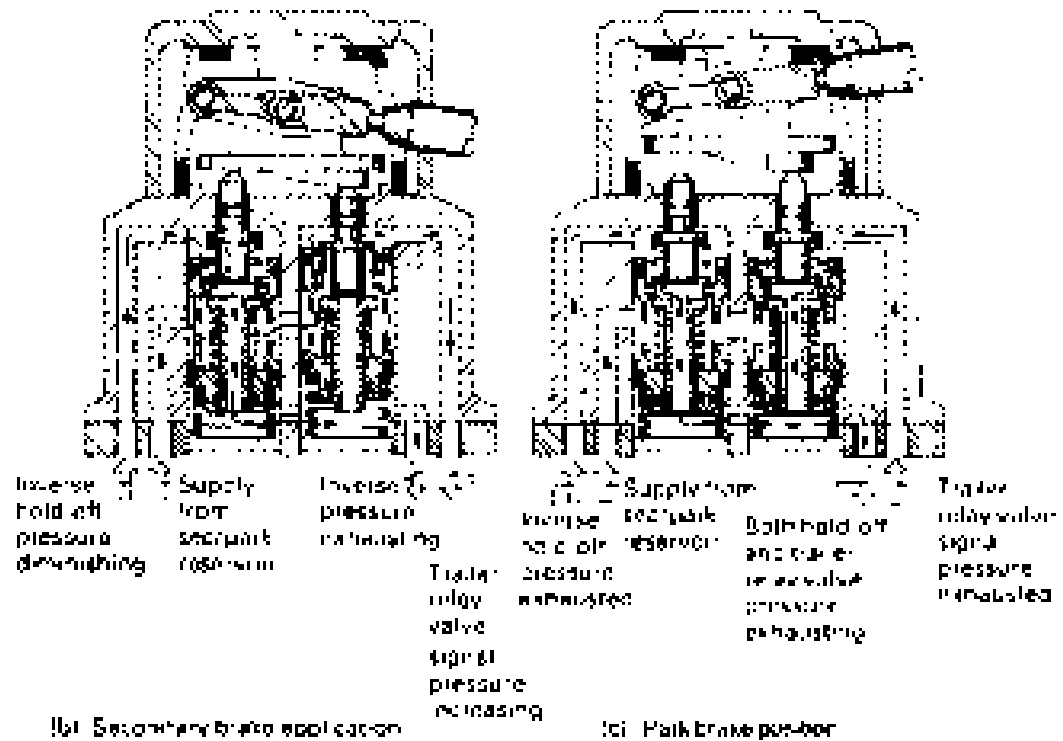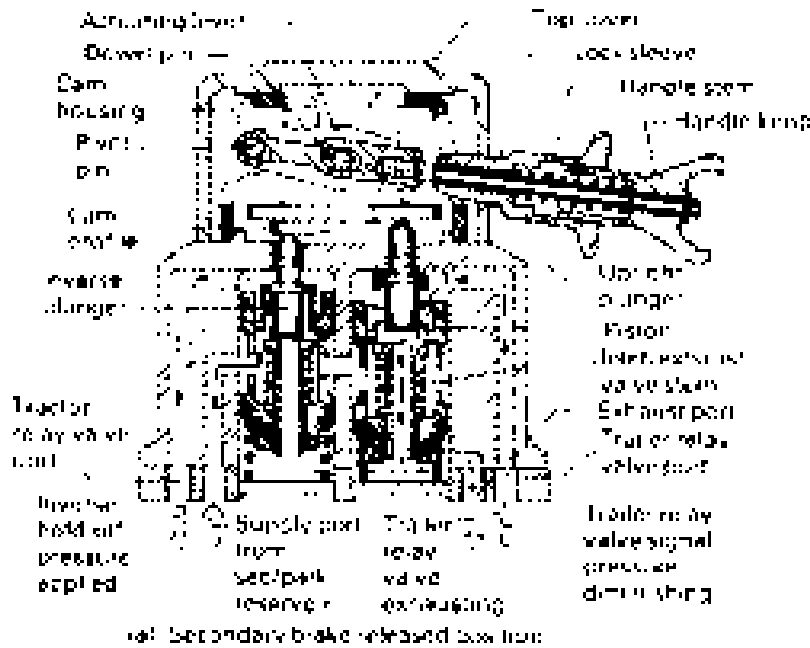
Fig. 12.18 (a–c)   Spring brake hand control valve

until with the secondary brake position the delivered pressure is zero.

In other words, the upright valve delivers a gradually increasing pressure to the trailer brake actuators and, at the same time, the inverse valve assembly allows the air pressure on the tractor spring brake actuators to be gradually released.

ark brake application (Fig. 12.18(c)) When the handle is moved from the secondary brake position to the park position, the cam lifted by the leverage of the handle about its pivot allows the upright plunger and the inverse plunger to be raised. The air pressure in both tractor and trailer brake actuators then exhaust into the atmosphere. The tractor brakes are now applied in the park position by the mechanical force exerted by the spring actuators.

12.3.14 Relay valve (piston type) (Bendix)
(Fig. 12.19(a and b))

Purpose The relay valve is used to rapidly operate a part of a braking system when signalled by either a foot or hand control valve. This is achieved by a small bore signal line feeding into the relay valve which then controls the air delivery to a large bore output service line. As a result, a small variation in signal pressure from the foot or hand valve will produce an instant response by the relay valve to admit air from the service reservoir directly to the service line brake system.

Operation

Brakes applied (Fig. 12.19(a)) When the brakes are applied, a signal pressure from the foot control valve (or hand control valve) reacts on the large control piston which responds by moving downwards rapidly until the centre stem of the piston closes the exhaust passage. The downwards movement of the piston pushes open the inlet valve. Air will now be admitted to the underside of the piston as it flows through to the service line and brake actuator. Movement of air from the service reservoir to the service line continues until the combined upthrust of both piston and valve springs and the air pressure balances the air signal pressure force, pushing the piston downwards. The piston now rises, closing the inlet valve so that both inlet and exhaust valves are in the *lapped* condition.

Brakes hold (Fig. 12.19(a and b)) A reduction in signal pressure now produces a greater force, pushing the piston upwards rather than downwards. The piston rises, closing the inlet valve, followed by the opening of the exhaust valve. The trapped air in the service line and actuator will now exhaust through the hollow valve stem to the atmosphere. The exhaustion of the service line air continues until the upward piston force balances the downward force caused by signal pressure. Both inlet and exhaust valves will subsequently close. These cycles of events are repeated the instant there is a change in signal pressure, be it an increasing or decreasing one, the valve being self-lapping under all conditions.

Brakes released (Fig. 12.19(b)) When the brakes are released, the signal pressure collapses, permitting the piston return spring to raise the piston; first closing the inlet valve, and then opening the exhaust valve. Air in the service line then escapes
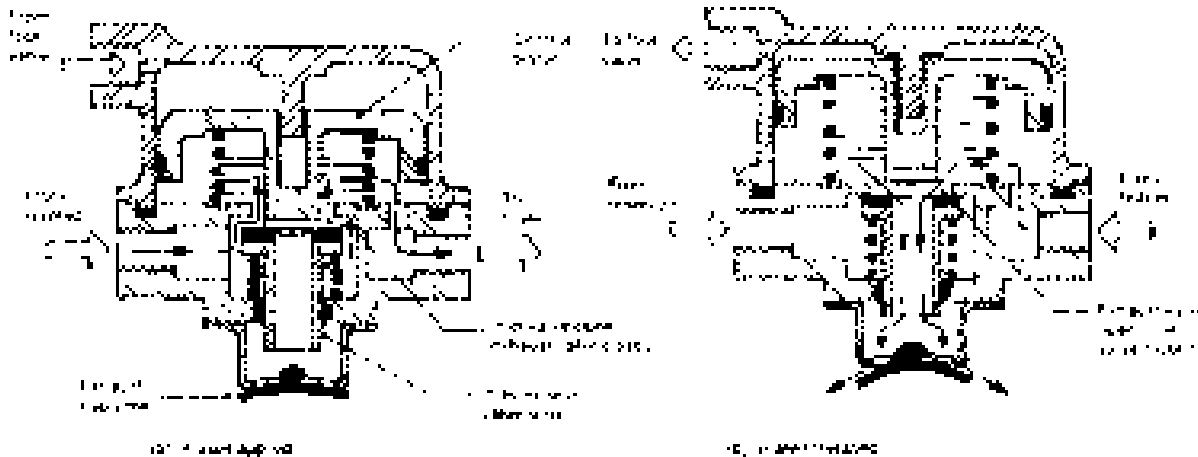
Fig. 12.19 (a and b) Relay valve

through the lower piston chamber and out into the atmosphere through the hollow valve stem.

12.3.15 uick release valve
(Fig. 12.20(a, b and c))

Purpose The quick release valve (RV) shortens the brake release time by speeding up the exhaustion of air from the brake actuator chambers, particularly if the actuators are some distance from the foot, hand or relay valve.

Operation

Applied position (Fig. 12.20(a))   When the brakes are applied, the air pressure from the foot or hand control valve enters the upper diaphragm chamber, forcing the diaphragm and its central stem down onto the exhaust port seat. The air pressure build-up then deflects downwards the circumferential diaphragm rim, thereby admitting air to the brake actuators via the pipe lines.

Hold position (Fig. 12.20(b)) Movement of air from the inlet port to the outlet ports permits air to occupy the underside of the diaphragm. Once the air pressure above and below the diaphragm has equalized, the diaphragm return spring upthrust pushes the outer diaphragm rim up onto its seat whilst the centre of the diaphragm and stem still seal off the exhaust port. Under these conditions, both inlet and exhaust passages are closed, preventing any additional air flow to occur to or from the brake actuators. The diaphragm is therefore in a state of 'hold'.

Released position (Fig. 12.20(c))   Releasing the air pressure above the diaphragm allows the trapped and pressurized air below the diaphragm to raise the central region of the diaphragm and stem. The trapped air in the brake lines and actuator chambers escape into this atmosphere.

Reducing the brake load slightly decreases the air pressure above the diaphragm, so that some of the air in the brake lines is allowed to escape before the pressure on both sides of the diaphragm balances again. The central region of the diaphragm moves down to close the exhaust port which moves the diaphragm into its 'hold' condition again.

The quick release valve therefore transfers any increased foot or hand valve control pressure through it to the brake actuators and quickly releases the air pressure from the brake actuators when the brake control valve pressure is reduced.
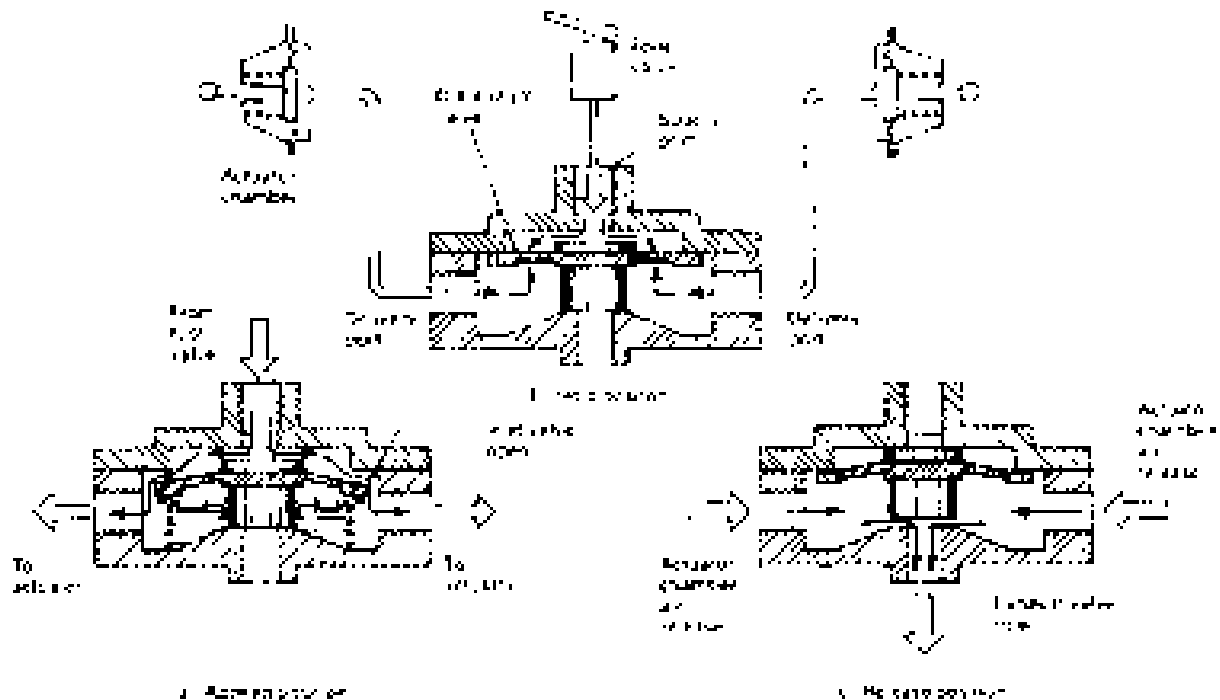
Fig. 12.20 (a–c)   uick release valve

By these means the air pressure in the brake actuators will always be similar to the delivery air pressure from the brake control valve.

12.3.16 Relay emergency valve
(Fig. 12.21(a–d))

Charging (Fig. 12.21(a)) Air delivery from the emergency line (red) enters the inlet port and strainer. The compressed air then opens the check valve, permitting air to flow across to and around the emergency piston, whence it passes to the outlet port leading to the trailer reservoir, enabling it to become charged.

If the reservoir is completely empty, both the relay piston and the emergency piston will be in their uppermost position. Under these conditions, the exhaust valve will be closed and the inlet valve open. Therefore some of the air flowing to the trailer reservoir will be diverted through the inlet valve to the brake actuator chambers, thereby operating the brakes. When the trailer reservoir charge pressure reaches 3.5 bar, air fed through a hole from the strainer pushes down on the annular area of the emergency piston causing the inlet to close. As the reservoir stored pressure rises to 4.2 bar, the downward air pressure force on the emergency piston moves the inlet/exhaust valve stem away from its exhaust seat, enabling the trapped air in the brake actuator chambers to escape to the atmosphere. The brakes will then be released.

Applying brakes (Fig. 12.21(b)) When the brakes are applied, a signal pressure is passed through the service line (yellow) to the upper relay piston chamber, forcing the piston downwards. The lowering of the relay piston and its central exhaust seat stem first closes the exhaust valve. It then opens the inlet valve which immediately admits compressed air from both the emergency line via the check valve (non-return valve) and the trailer reservoir through the central inlet valve, underneath the relay piston and out to the brake actuator chambers. The expanding brake actuator chambers subsequently press the brake shoes into contact with the drums.

Balancing brakes (Fig. 12.21(b and c)) As the air pressure in the actuator chambers builds up, the pressure underneath the relay piston increases its upthrust on the piston until it eventually equals the downward relay piston force created by the service line pressure. At this point the inlet valve also closes, so that both valves are now in a balanced state. Until a larger service line pressure is applied to the relay piston, the central stem will not move further down to open the inlet valve again and permit more air to pass to the brake actuator chambers. Conversely, if the foot brake is slightly released, initially the relay piston is permitted to rise, closing the inlet valve, followed by opening of the exhaust valve to release some of the air pressure acting on the brake actuator chambers.

Releasing brakes (Fig. 12.21(c)) Removing the load on the foot control valve first closes off the air supply to the service line and then releases the remaining air in the service line to the atmosphere. The collapse of service line pressure allows the relay piston to rise due to the existing brake actuator pressure acting upwards against the relay piston. The hollow valve stem immediately closes the inlet valve passage, followed by the relay piston centre stem exhaust seat lifting clear of the exhaust valve. Air is now free to escape underneath the relay piston through the central hollow inlet/exhaust valve inlet stem and out to the exhaust vent flap to the atmosphere. The brake actuators now move to the 'off' position, permitting the 'S' cam expanders to release the brake shoes from their drums.

mergency position (Fig. 12.21(d)) If the air pressure in the emergency line (red) should drop below a predetermined minimum (normally 2 bar), due to air leakage or trailer breakaway, then the air pressure around the upper shoulder of the emergency piston will collapse, causing the emergency piston return spring to rapidly raise the piston. As the emergency piston rises, the hollow inlet/exhaust valve stem contacts and closes the relay piston exhaust stem seat. Further piston lift then opens the inlet valve. Air from the trailer reservoir is now admitted through the control inlet valve to the underside of the relay piston where it then passes out to the trailer brake actuator chambers. The trailer brakes are then applied automatically and independently to the demands of the driver.

A trailer which has been braked to a standstill, caused by a failure in the emergency line pressure, can be temporarily moved by opening the trailer's reservoir drain cock to exhaust the trailer brake actuators of pressurized air.

12.3.17 Differential protection valve
(Fig. 12.22(a, b and c))

Purpose The differential protection valve prevents both service brakes and secondary brakes

applying their full braking force at any one time. The valve is designed to supply secondary line pressure to the spring brake release chambers when the service brakes are operating or to allow the service line pressure supplying the service brake chambers to decrease as the spring brakes are applied. By these means the spring and diaphragm actuator forces are prevented from compounding and overloading the combined spring and diaphragm actuator units and the foundation brakes which absorb the braking loads.

Operation

Brakes in off position (Fig. 12.22(a)) Releasing both the foot and hand brakes exhausts air from the service line. Air from the secondary line enters the secondary inlet port of the valve and flows between the outer piston and the casing to the spring brake output ports. It then passes to the actuator air chambers. The compressed air now holds the secondary springs in compression, thereby releasing the brake shoes from the drums.
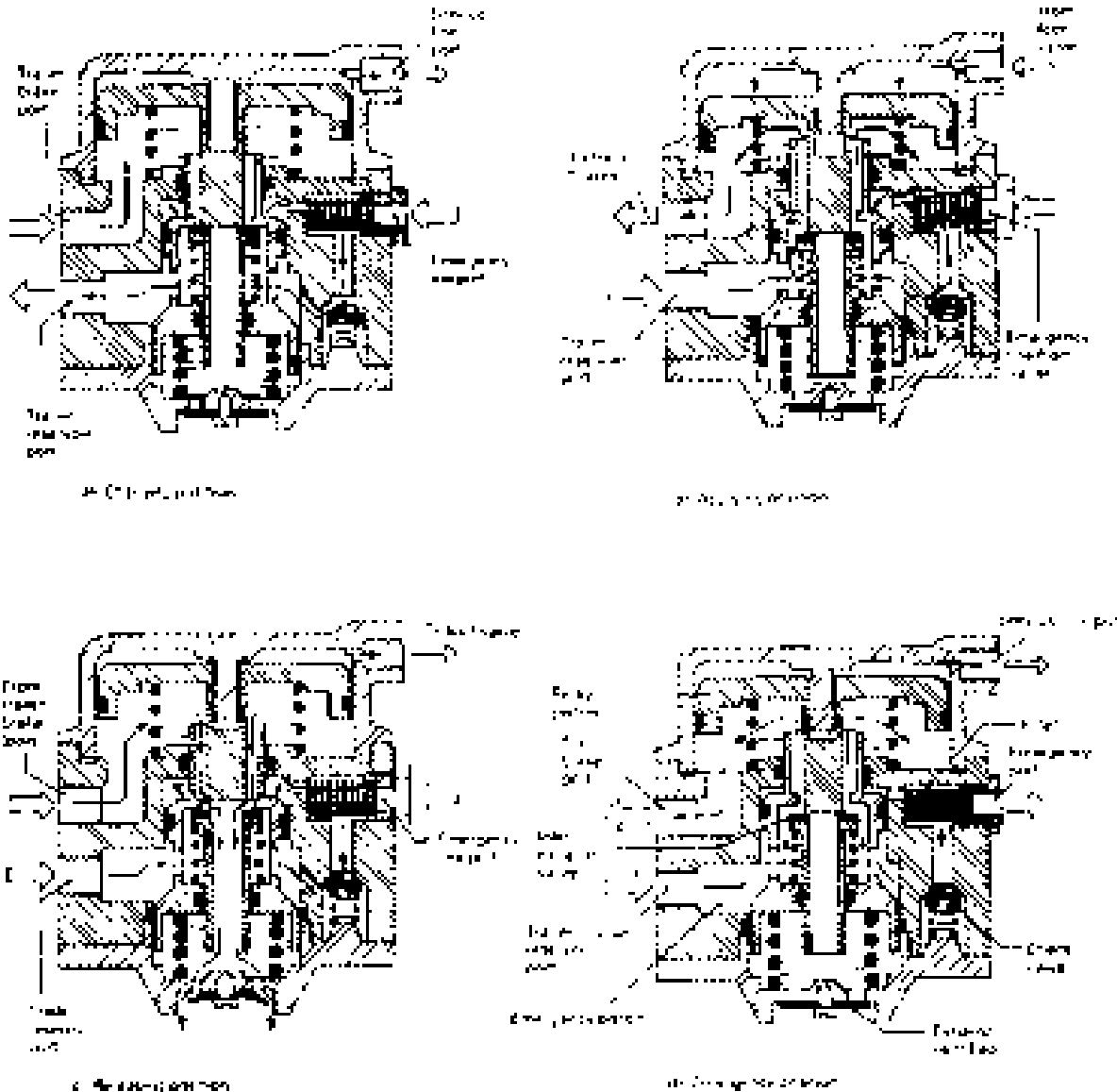
Fig. 12.21 (a–d) Relay emergency valve

Secondary (spring) brake application (Fig. 12.22(b)) When the secondary (spring) brakes are applied, following the initial application and holding of the service (foot) brakes, the compressed air in the spring actuator chambers and in the secondary line is exhausted via the differential protection valve to the atmosphere through the hand control valve. As the secondary line pressure reduces, the pressure trapped in the service line due to the previous foot brake application becomes greater than the decreasing pressure in the secondary line. It therefore causes the inner piston to be pushed across to block the secondary port air exit. Immediately afterwards, the outer piston is unseated so that service line air now flows through the valve from the service line inlet port to the spring delivery ports and from there to the spring actuator chambers. The service line air which has entered the secondary line now holds the springs so that they are not applied whilst the driver is still applying the foot brake.

As the driver reduces the foot pedal pressure, the corresponding reduction in service line pressure permits the outer piston, followed by the inner piston, to move away from the secondary line inlet port, closing the service line inlet port and opening the secondary inlet port. The compressed air occupying the spring brake actuator chambers is now permitted to fully exhaust so that the expanding springs re-apply the brakes simultaneously as the service (foot) brakes are being released.

Service (foot) brake application (Fig. 12.22(c)) When the service (foot) brakes are applied after a spring brake application, the secondary line will be exhausted of compressed air, which was essential for the spring brakes to operate. Therefore, as the service line pressure rises, it pushes the inner piston against its seat, closing the secondary line inlet port. With a further increase in service line pressure, the outer piston becomes unseated so that service line pressure can now flow through the valve and pass on to the spring brake actuators. This withdraws the spring brake force, thereby preventing the compounding of both spring and service chamber forces.

While the differential protection valve is in operation, an approximate 2.1 bar pressure differential between the service pressures and the delivered effective anti-compounding pressure will be maintained across the valve.

12.3.18 Double check valve (Fig. 12.23)

Purpose When two sources of charging a pipe line are incorporated in a braking system such as the service (foot) line and secondary (hand) line circuits, a double check valve is sometimes utilized to connect whichever charging system is being used to supply the single output circuit and to isolate (disconnect) the charging circuit which is not being operated at that time.

Operation (Fig. 12.23(a and b)) The two separate charging circuits (service and secondary lines) are joined together by the end inlet ports of the double check valve. When one of the brake systems is applied, air charge will be delivered to its double
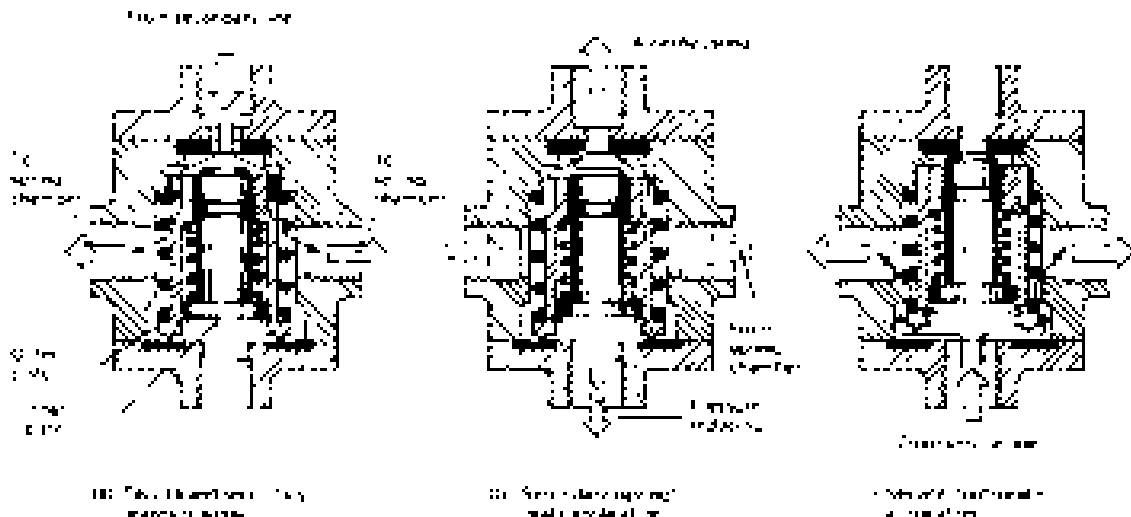
Fig. 12.22 (a–c) Differential protection valve
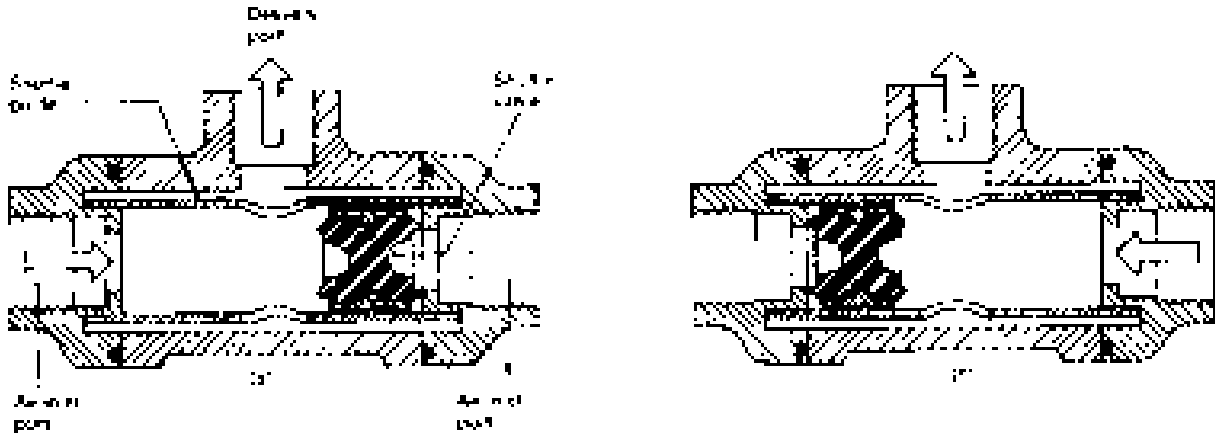
Fig. 12.23 (a and b) Double check valve

check valve inlet port, pushing the shuttle valve to the opposite end, thereby sealing off the inoperative charging system. Air from the active charging system will now flow from its inlet port through to the delivery port where it then charges the brake actuator chambers. If the charge source is switched, say from the hand control to foot application, the shuttle valve shifts against the non-pressurized end inlet port, causing it to close. Air from the foot control circuit will now pass through the double check valve on its way to the brake actuators.

12.3.19 *Variable load valve* (Fig. 12.24)

Purpose This valve is designed to sense the vertical load imposed on a particular axle by monitoring the charge in suspension height and to regulate the braking force applied to the axle's brakes in proportion to this loading. The valve therefore controls the brake actuator chamber air pressure in accordance with the load supported by the axle and the service line pressure.

Operation (Fig. 12.24(a, b and c)) The valve is mounted on the vehicle's chassis and its control lever is connected to the axle through a vertical adjustable link rod. The valve control lever is in its lowest position with the axle unladen, moving to its highest position as the axle load is increased to fully laden.

Brakes released (Fig. 12.24(a)) When the brakes are released, the service line pressure collapses, permitting the control piston to rise to its highest position. Because the valve stem rests on the ball pin, the inlet valve closes whereas the exhaust valve is

unseated. Pressurized air in the brake actuator chambers and pipe line will subsequently flow underneath the diaphragm, up and around the hollow valve stem, past the exhaust valve and its seat into the atmosphere via the control exhaust passage.

Brakes applied (Fig. 12.24(b and c)) When the brakes are applied, service line pressure enters the upper piston chamber, pushing the control piston downwards. At the same time, some of the air is transferred through the external pipe to the lower clamp plunger, which is then forced upwards against the ball pin. As the control piston moves downwards, the exhaust/inlet valve stem closes the central exhaust passage and then uncovers the inlet valve passage. Air from the service line inlet port now passes through the inlet valve to the lower diaphragm chamber and from there it continues on its way to the brake actuator chambers.

If the axle is laden, the control lever ball pin will be in a high position so that the control piston does not move very far down before the exhaust valve is closed and the inlet valve is opened. Conversely, if the axle is unladen the control lever and ball pin will be in a much lower position so that the control piston has to move much further downwards.

When the brakes are released, the clamp plunger chamber is exhausted of air so that the valve stem assembly will not be rigidly attached to the ball pin and only becomes active during brake application. Hence unnecessary wear is avoided.

Brakes applied with heavy load (Fig. 12.24(b)) When the axle is laden, the ball pin will hold the valve stem in the high position, therefore the control piston will also be in the upper position. Under

537

Fig. 12.24 (a–c) Variable load valve

these conditions the underside of the diaphragm reacts against the fixed fins and only a small portion of the diaphragm area is supported by the moving fins attached to the piston. This means that very little piston upthrust is provided, which therefore permits the inlet valve to open wide and to admit a large air delivery pressure to the brake actuators. As the air supply flows through the valve, the pressure under the diaphragm increases until the upthrust acting on the varying effective area of the diaphragm equals that produced by the

service line pressure acting on top of the control piston. The valve assembly now moves into a lapped condition whilst the forces imposed on the piston are in a state of balance.

Brakes applied with light load (Fig. 12.24(c)) When the axle is unladen, the ball pin will hold the valve stem in a lower position so that the control piston will be forced by the service line air pressure to move further down. Under these new conditions the underside of the diaphragm reacts against the

538

moving fins more than the fixed ones. Consequently there will be a much larger diaphragm upthrust, tending to partially close the inlet valve whilst air pressure is being delivered to the brake actuator chambers. As a result, the piston will move to a new position of balance and the valve assembly again moves into a lapped condition.

It can be seen that the variable load valve automatically regulates the output air pressure delivered to the axle brake actuators in proportion to the laden weight imposed on the axle.

12.3.20 ulti-relay (triple) valve
(Fig. 12.25(a–d))

Purpose With a two line braking system the trailer has no secondary braking system. Therefore the tractor foot control valve and the hand control valve must each be able to apply the single trailer brake system independently. This is made possible by the utilization of a multi-relay (triple) valve which is very similar to the conventional single relay valve except that it has three signal sensing relay pistons placed one above the other.

Operation

Brakes released (Fig. 12.25(a)) If the brakes are released, all three relay pistons will rise to their uppermost positions due to the return spring upthrust. Consequently, the inlet valve closes and the exhaust valve will be unseated. This ensures that the trailer brake actuators are cleared of compressed air, so releasing the brakes.

Secondary line brake application (Fig. 12.25(b)) Applying the hand control valve handle sends a pressure signal to the lower relay piston (3). The lower relay piston will move downwards, initially closing the exhaust valve and then opening the inlet valve. A pressure signal will then pass from the trailer reservoir mounted on the tractor to the upper part of the trailer's emergency relay valve. As a result, air pressure from the supply line (red) now flows to the trailer brake actuators.

Service line brake application (Fig. 12.25(c and d)) When the foot control valve is depressed a signal pressure from the both halves of the foot valve is transmitted to the upper (1) and middle (2) relay valve pistons. Both relay pistons react immediately by moving down until the three relay pistons are pressed together. Further downward movement will close the exhaust valve and open the inlet

valve. Air from the trailer reservoir mounted on the tractor will now pass to the emergency relay valve, permitting air from the supply line (red) to pass directly to the trailer brake actuators via the now opened passage passing through the emergency valve.

Should half of the dual foot valve service line circuit develop a fault, the other half service line circuit will still be effective and be able to operate the multi-relay valve.

12.3.21 Supply dump valve
(Fig. 12.26(a, b and c))

Purpose The supply dump valve has been designed to meet one of the C Brake Safety irective for Trailers which requires that if there is an imbalance of air pressure between the tractor service line and the trailer service line due to leakage or decoupling, then within two seconds of the next full service brake application the compressed air in the trailer supply (emergency) line will be dumped to the atmosphere, reducing the pressure to 1.5 bar. The result of the service line pressure collapse signals the trailer emergency valve to transfer compressed air stored in the trailer reservoir to the trailer brake actuators, so causing the brakes to be applied.

Operation

Brakes released (Fig. 12.26(a)) When the brake pedal is released, compressed air exhausts from the supply dump valve tractor and trailer service line sensing chambers. Under these conditions, the piston spring forces the piston and exhaust valve stem down and unseats the inlet valve. Air from the tractor emergency (supply) line is therefore free to flow through the supply dump valve to the trailer's emergency (supply) line to charge the trailer's reservoir.

Service brakes applied (Fig. 12.26(b)) When the foot pedal is depressed the tractor service line output from the foot control valve and the multi-relay valve output to the trailer service line both send a pressure signal. The air pressure in both the upper and lower chambers will therefore be approximately equal. Because the piston's upper surface area is greater than its underside area and the piston spring applies a downward thrust onto the piston, the piston will be forced to move to its lowest position. This lowering of the piston closes the exhaust valve and opens the inlet valve. Air is now able to flow from the tractor emergency
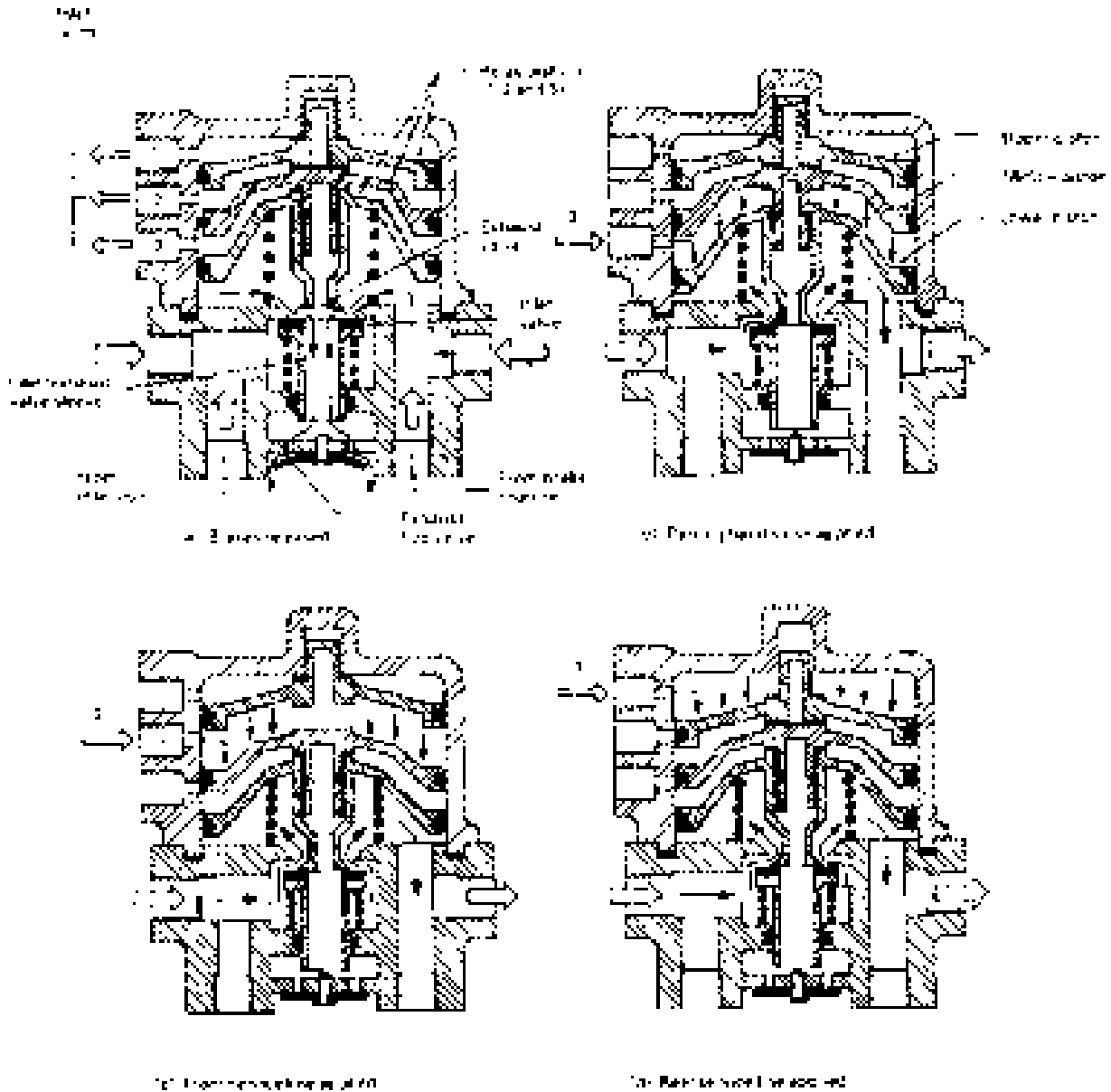
Fig. 12.25 (a–d) Multi-relay valve application

(supply) line to the trailer's emergency line via the open inlet valve mounted in the lower part of the dump valve. As a result, when the service line pressure signal is delivered to the emergency relay valve, the emergency (supply) line passes compressed air to the brake actuators on the trailer, thereby engaging the brakes.

Failure of service line pressure (Fig. 12.26(c)) Should the trailer service line be at fault, causing the piping or coupling to leak, the air pressure in the upper trailer service line sensing chamber will be lower than that in the tractor service line sensing chamber. Consequently the piston will lift, causing the inlet valve to close so that no more compressed air passes to the trailer emergency line and the exhaust valve becomes unseated. Air trapped in the trailer emergency line will immediately discharge through the centre hollow exhaust valve stem to the atmosphere. Once the trailer emergency

540
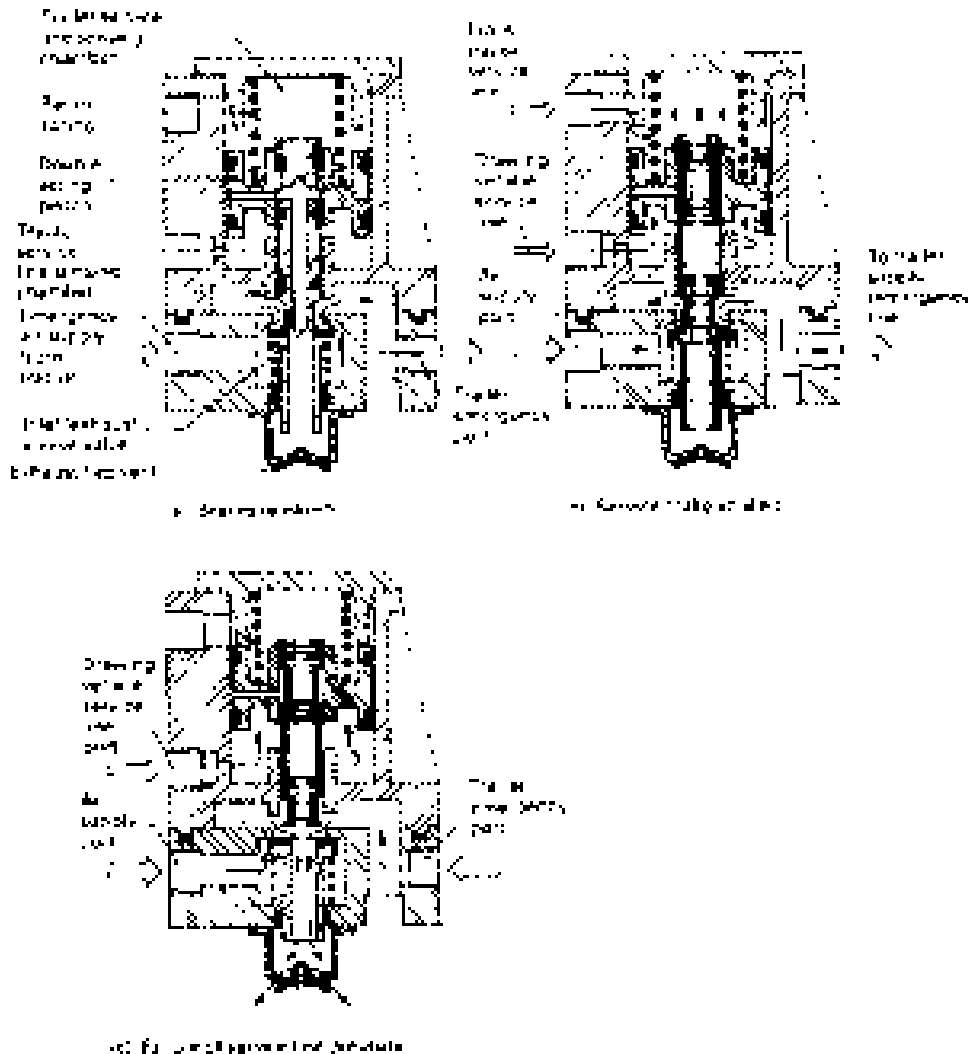
Fig. 12.26 (a–c) Supply dump valve (Bendix)

line pressure has dropped below 2 bar, the emergency relay valve inlet passage opens, permitting the compressed air stored in the trailer reservoir to discharge into the trailer brake actuators. The towing and towed vehicles are therefore braked to a standstill.

12.3.22 Automatic reservoir drain valve
(Fig. 12.27(a–d))
Discharged air from the compressor entering the reservoir goes through a cycle of compression and expansion as it is exhausted during brake on/off applications. The consequence of the changing air density is the moisture, which is always present in the air, condenses against the cold walls of the

reservoir, trickles down to the base of the chamber and thereby forms a common water pool. Permitting water to accumulate may result in the corroding of certain brake components and in cold weather this water may freeze thereby preventing the various braking valves from functioning correctly.

The object of the automatic reservoir drain valve is to constantly expel all the condensed unwanted water into the atmosphere from any container it is attached to.

Operation (Fig. 12.27(a–d))   If there is no air pressure in the braking system, both the inlet and exhaust valves will be in the closed position (Fig. 12.27(a)). Initially, as the compressor commences

541
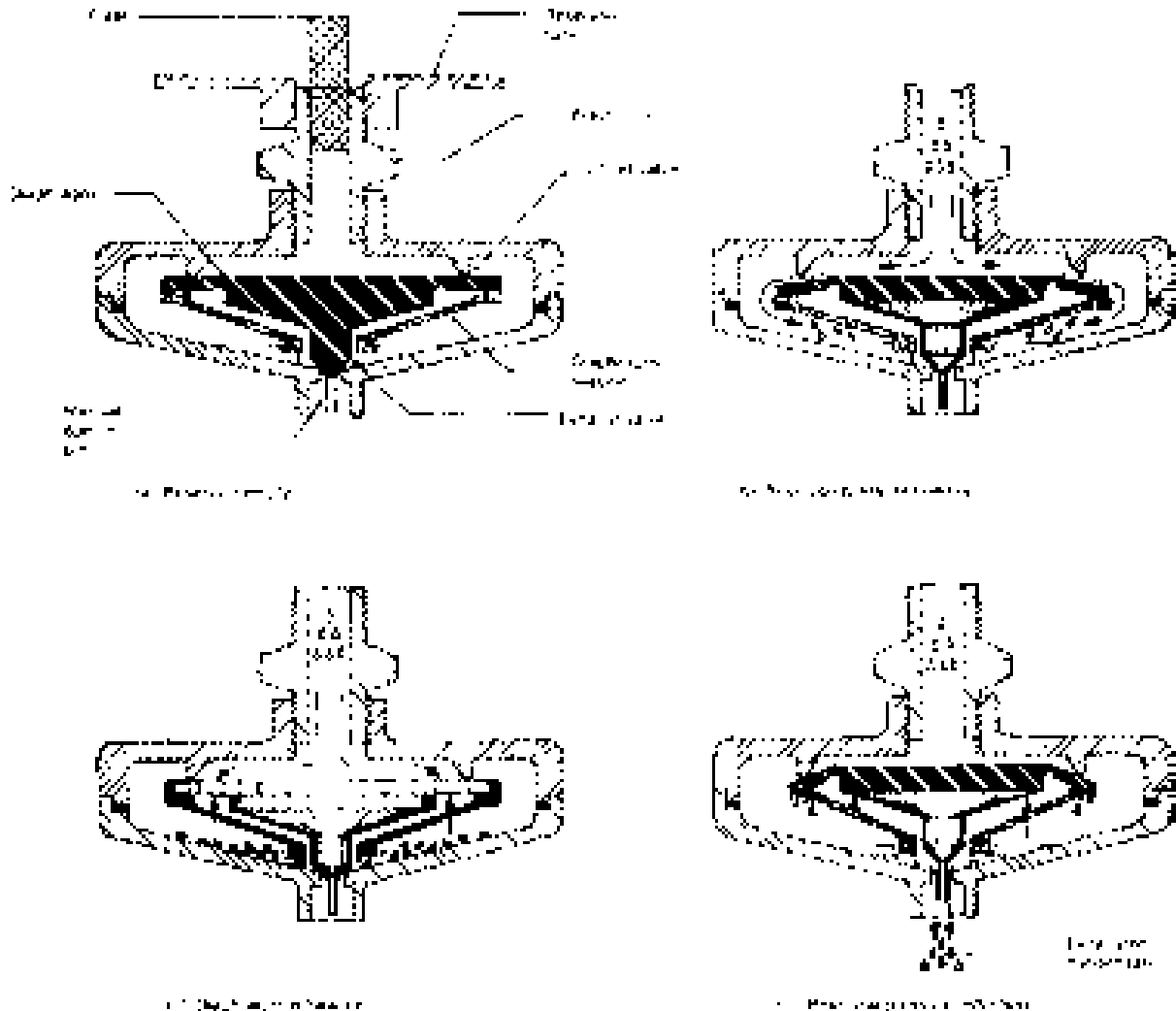
Fig. 12.27 (a–d) Automatic reservoir drain valve

to charge the reservoir, the air pressure rises and pushes open the inlet diaphragm valve. Condensed water collected above the diaphragm will now gravitate to the lower conical sump of the valve during the time the pressure is rising (Fig. 12.27(b)).

When peak governor pressure (cut-out pressure) is reached, the compressor is unloaded, cutting off further reservoir air supply so that the pressure above and below the valve diaphragm equalize. As a result, the diaphragm support below the diaphragm closes the inlet valve (Fig. 12.27(c)). Air is consumed due to the application of the brakes. There will be a reduction in reservoir air pressure so that the trapped air pressure below the diaphragm will be slightly higher than that above.

Consequently the pressure difference between the two sides of the diaphragm will be sufficient to lift the middle portion of the diaphragm and the conical exhaust valve clear of its seat (Fig. 12.27(d)). Thus the air trapped underneath the diaphragm forcibly expels any condensate or foreign matter which has collected into the atmosphere.

Manual draining of the reservoir with the automatic valve is obtained by pushing up the vertical pin situated in the base of the exhaust port.

This type of automatic drain valve is designed for large reservoirs positioned away from the compressor. It is unsuitable for small sensing tanks or small volume condensers mounted near to the compressor.

542

12.3.23 Brake actuator chambers

Purpose Brake actuator chambers convert the energy of compressed air into the mechanical force and motion necessary to operate the foundation brakes which are attached to the ends of each axle casing or steering stub axles.

Brake actuators are designed for braking systems which must meet various application requirements. Broadly, brake actuators come into the following classifications:

1 single diaphragm chamber type,
2 double diaphragm chamber type,
3 triple diaphragm chamber type,
4 diaphragm/piston chamber push or pull type,
5 spring brake diaphragm/piston chamber type,
6 lock wedge actuator (now obsolete and will not therefore be described),
7 remote spring brakes.

Single diaphragm chamber type actuator (Fig. 12.28(a)) These actuators are normally used on single line trailers incorporating 'S' cam brake shoe expanders and slack adjuster levers.

The unit consists of two half chamber pressings supporting a flexible rubber diaphragm positioned inbetween. The sealed chamber subjected to compressed air is known as the *pressure plate chamber* whereas the chamber exposed to the atmosphere and housing the return spring is known as the *non pressure plate chamber* Holding the two half plates and diaphragm together is a clamp ring which, when tightened, wedges and seals the central diaphragm. The non-pressure plate also has studs attached, permitting the actuator to be bolted to the axle casing mounting bracket. A rigid disc and push rod transfer the air pressure thrust acting on the diaphragm when the brakes are applied to the slack adjuster lever.

Service line application When the brakes are applied, the foot brake valve is depressed, admitting compressed air into the pressure plate chamber. The build-up of pressure inside the chamber forces the diaphragm and push rod assembly outwards to apply the brakes. When the foot brake valve is released, the compressed air in the pressure plate chamber is exhausted to the atmosphere via the foot valve, permitting the return spring to move the diaphragm and push rod assembly back against the pressure plate wall in the off position.

ouble diaphragm chamber type actuator (Fig. 12.28(b)) The double diaphragm chamber actuators are designed to be used when there are two separate air delivery circuits, known as the *service line (foot)* and the *secondary (hand) line* systems operating on each foundation brake.

Service line application When the service (foot) brake is applied compressed air enters the service chamber via the intake ring. As the air pressure rises, the service diaphragm and push rod are forced outwards, applying the leverage to the slack adjuster.

Secondary application When the secondary (hand) brake is applied, compressed air enters the secondary chamber via the central pressure plate port. Rising air pressure forces both diaphragm and push rod outwards again, applying leverage to the slack adjuster and expanding the shoes against the brake drum.

Service and secondary brake line failure Should the service diaphragm puncture, the hand control valve secondary line can be used to operate the secondary diaphragm and push rod independently to apply the brakes.

Should the secondary diaphragm fail, the foot control valve service line will provide an alternative braking system. The air pressure in the service chamber automatically pushes the secondary diaphragm back against the central inlet port seal, preventing service line pressure escaping back through the secondary line due to a damaged or leaking secondary diaphragm.

If the secondary diaphragm and seal should fail together the service brake will still operate, provided the hand control valve is moved to the fully applied position because in this condition the exhaust valve exit for the secondary line is closed.

riple diaphragm chamber type actuator (Fig. 12.28(c)) The triple diaphragm chamber actuator functions similarly to the double diaphragm actuator. Both types of actuators are designed to accommodate and merge both service and secondary braking systems into one integral wheel brake actuator, hence permitting both brake systems to operate independently of each other but applying their braking force and movement to one common slack adjuster lever.

Service line application When the service (foot) brake is depressed, compressed air passes through the inlet ring port between the central diaphragm and the service diaphragm nearest the push rod assembly. With increased air pressure, the secondary
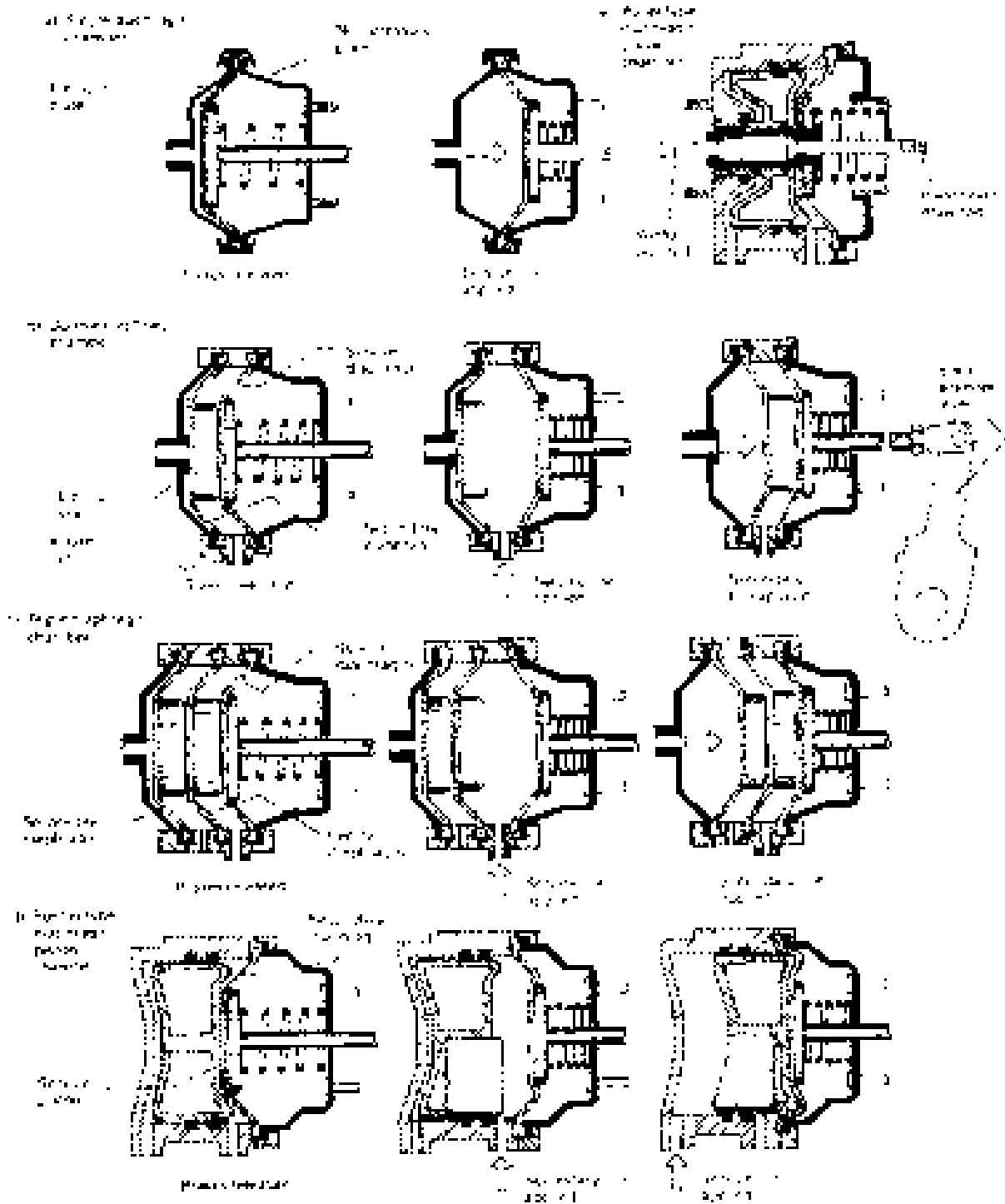
Fig. 12.28 (a–d) Air brake actuator chambers

544

and central diaphragm react against the pressure plate chamber and the service diaphragm forces the push rod assembly outwards. This results in the rotation of the slack adjuster lever, camshaft and cam expander, causing the brake shoes to grip the drums and apply the brakes.

Secondary line application When the secondary (hand) brake is operated, compressed air is admitted through the central end port between the pressure plate and the secondary diaphragm, causing the diaphragm and push rod assembly to expand outwards, again applying the brakes similarly to the service brake application. Releasing the hand operated lever permits the compressed air to exhaust out of the chamber through the hand control valve into the atmosphere. The return spring is now able to move all three diaphragms in the opposite direction towards the pressure plate chamber.

Service and secondary brake line failure Should the service diaphragm or central diaphragm rupture, there will be no interference with operation of the secondary diaphragm chamber. Likewise, if the secondary diaphragm fails, the service line air pressure operates only between the central and service diaphragm so that they will apply the brakes as normal.

 Note that with the triple diaphragm arrangement, the central diaphragm performs the same function as the central inlet port seal in the secondary diaphragm chamber so that there is no possibility of a back leakage problem as with the double diaphragm actuator.

 iaphragm/piston chamber type actuator (Fig. 12.28) This diaphragm/piston type actuator operates similarly to the double and triple diaphragm type actuators, but because the whole cross-sectional area of the piston is effective in applying air pressure thrust (as opposed to a variable effective diaphragm area) the service chamber has a high mechanical efficiency.

 usher type piston diaphragm chamber (Fig. 12.28(d))

Service line application When the brake pedal is depressed and the foot valve opens, air is supplied to the service port of the brake chamber. Air pressure is then applied between the rear of the chamber and the piston face. The thrust produced on the piston moves the piston, diaphragm and push rod

outwards against the resistance of the return spring. The force and movement of the push rod actuates the wedge type of brake shoe expander, normally used with this type of brake actuator, to apply the brakes. Releasing the foot pedal permits the compressed air in the chambers to exhaust through the foot control valve into the atmosphere.

Secondary line application When the secondary hand control valve is operated, air is delivered to the brake chamber secondary port where it enters the space made between the piston and the diaphragm. As the air pressure increases, the piston reacts against the brake chamber's rear wall. The diaphragm and push rod are forced outwards again to force the wedge between the brake shoe expander to apply the brakes.

 If air failure occurs in either the service or secondary chambers the other braking system can be operated independently to apply the brakes.

 uller type piston diaphragm chamber (Fig. 12.28(e))

Service line application When air is delivered to the service chamber port, the piston, diaphragm, central sleeve and the wedge expander rod are all pulled outwards from the rear wall of the brake chamber, actuating the wedges of the brake shoe expander to apply the brakes.

Secondary line application Similarly when the secondary hand control is operated, air passes through the secondary chamber port to the space between the piston and diaphragm, so that the diaphragm and the central sleeve assembly are moved outwards from the chamber rear wall. As before, the brakes are applied.

 When either the service or secondary line air pressure operates the brake chamber actuator, the hand brake draw rod remains stationary. When the hand brake is operated the draw rod connected to the hand brake cable (not shown) operates the sleeve and pull rod applying the brakes.

 Spring brake actuator (Fig. 12.29(a–d)) Spring brake actuators are designed for use in air pressure braking systems to produce the force and movement necessary to operate the foundation brakes. The actuator uses air pressure to respond and control both service and secondary braking. The conversion of air pressure into mechanical force and travel for the service braking systems is obtained by a conventional diaphragm-operated

servo chamber. In contrast, secondary braking and parking is achieved through the strain energy stored in a powerful coil spring which, when permitted to expand, applies mechanical effort and displacement to the foundation brake shoe expander.

peration

All brakes released (Fig. 12.29(a)) Under normal driving conditions both service and secondary/park brake systems must be in the 'off' position. To release both braking systems, air pressure (above the low signal pressure) is supplied to the

spring piston chamber which compresses the power spring and holds off the foundation brakes. At the same time air pressure is exhausted from the service line diaphragm chamber.

Service line applied (Fig. 12.29(b)) Air pressure controlled progressively by the foot valve is supplied to the service diaphragm chamber port. The cross-sectional air of the diaphragm exposed to the air pressure is subjected to a thrust forcing the push rod outwards. The combined movement and force is applied to the slack adjuster which then relays it to the camshaft, expander cam and shoes of the

Fig. 12.29 (a–d) Spring brake actuator

546
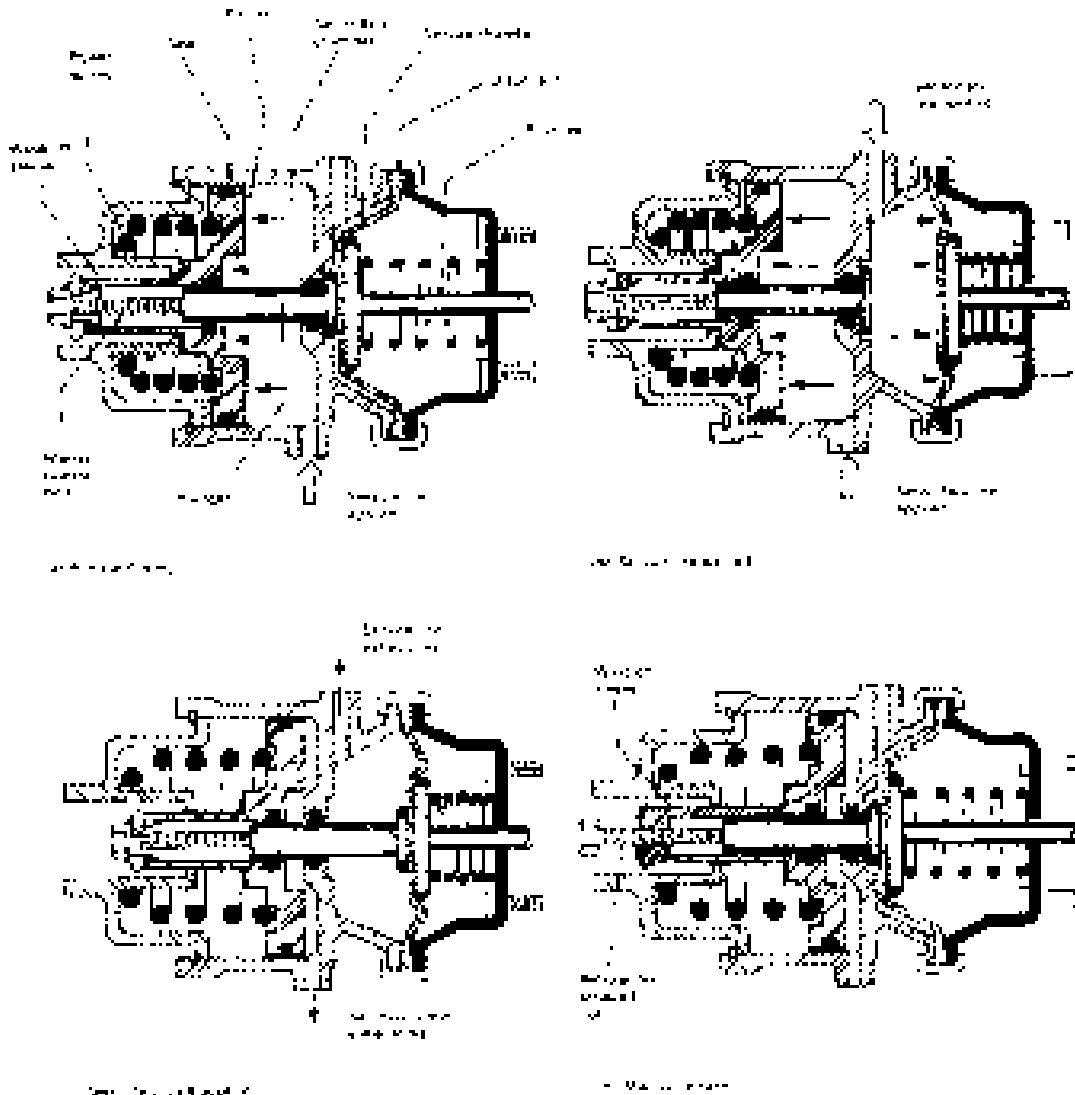
foundation brakes. During service brake operation the power spring is held in compression by the secondary line air pressure so that it does not compound the service brake operational force.

Secondary parking applied (Fig. 12.29(c)) Air pressure, controlled progressively by the hand valve, is released from the piston spring chamber, permitting the power spring to expand and push the plunger and push rod outwards against the resistance of the brake cam which is expanding the brake shoes up to the drum. When the vehicle is at a standstill, the hand lever is moved from secondary to park position, where any remaining air pressure in the secondary line and piston spring chamber is exhausted via the open hand control valve to the atmosphere. The brakes are then held on purely by the strain energy of the extended power spring and therefore are not dependent upon air pressure with its inherent leakage problem.

anual release applied (Fig. 12.29(d)) Should there be a secondary line air supply failure, then the foundation brakes may be released for towing purposes or removal of the spring brake actuator by the readily accessible release bolt. Winding out the release bolt permits the plunger to move into the wind-off sleeve so that the push rod is able to return to the 'off' position, thereby releasing the brake shoes.

12.4 Vehicle retarders

12.4.1 Engine overrun vehicle retardation
(Fig. 12.30)
When an engine is being overrun by the transmission system, the accelerator pedal is released and no fuel is available for combustion. Therefore the normal expansion in the power stroke by the burning products of combustion is replaced by the expansion of the compressed air alone. Consequently, the energy used in compressing the air on its upstroke is only partially given back on the downstroke, due to the friction and heat losses, so that additional work must be done to rotate the crankshaft (Fig. 12.30).

Thus an external energy source is necessary to keep this crankshaft rotating when no power is produced in the cylinder. This is conveniently utilized when the vehicle is being propelled by its own kinetic energy: the work put into overcoming the pumping, friction and heat losses while driving the

transmission and engine reduces the vehicle kinetic energy, causing the vehicle to slow down. The retarding torque required on the road driving wheels to rotate the transmission and engine consists of the torque necessary to overcome the engine resistances when there is no power produced multiplied by the overall gear ratio of the transmission system.

12.4.2 Exhaust compression vehicle retardation
(Fig. 12.30)

Principle of e haust compression retarder The object of the exhaust brake (retarded) is to convert the exhaust gas expelling stroke into an air compressing power, absorbing upstroke when the accelerator pedal is released and the vehicle possesses a surplus of kinetic energy. This happens on overrun and tends to propel the vehicle forward, thus causing the transmission to rotate the engine. To achieve this negative work, a butterfly valve or slide valve is incorporated into the exhaust pipe as near to the manifold as possible, so that during the exhaust stroke the volume of air in the engine's cylinder will be displaced and contained within the exhaust system, causing its pressure to rise (Fig. 12.30). ust after the piston has reversed its direction and commences to move outwards on its induction stroke, the exhaust valve closes, trapping
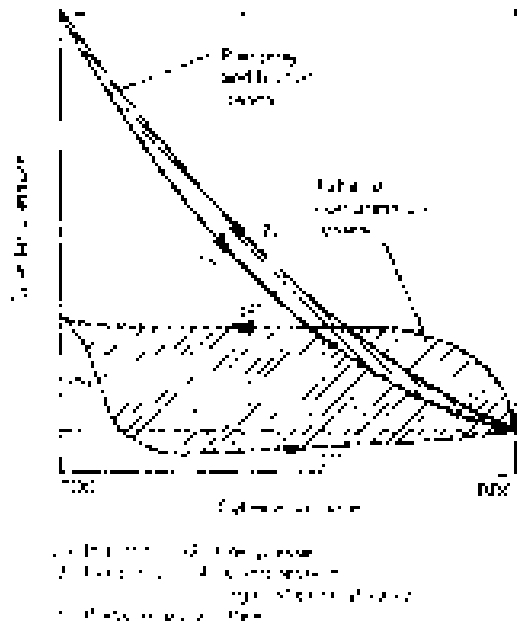
Fig. 12.30 Pressure/volume diagram for a four stroke engine installed with an exhaust compression

the air remaining between the exhaust port poppet valve and the exhaust butterfly brake valve in the exhaust system. Therefore approximately 90% of the exhaust upstroke is devoted in compressing air and absorbing energy (Fig. 12.30) before the inlet valve opens. This energy is then released to the atmosphere and cannot do useful work in pushing down the piston on the following induction stroke. Hence there is a net gain in negative work done by the engine whilst resisting the rotation of the crankshaft, which considerably helps to reduce the vehicle's speed.

Air will therefore be drawn into the cylinder on each induction stroke as normal, but on subsequent exhaust strokes the pressure in the exhaust manifold and pipe increases to a maximum of between 2.5 and 4.5 bar, depending upon the valve overlap compression ratio and engine speed.

With increased engine speed, the pressure build-up in the exhaust system eventually overcomes the exhaust valve spring closing force and unseats the exhaust valve. Pressurized air will then flow back into the cylinder and escape out of the induction port the next time the inlet valve opens. Consequently the upper limit to the pressure build-up on the exhaust upstroke is controlled by the amount of trapped compressed air stored in the exhaust system behind the exhaust valve when the exhaust valve opens and the piston commences on its modified exhaust upstroke.

With several cylinders feeding into a common exhaust manifold, pressure fluctuations are considerably reduced and a relatively high average pressure can be maintained. Subsequently a great deal of negative work can be done by all the cylinders collectively.

Operation of e haust compression brake

Retarder operating (Fig. 12.31(a))   When the foot control (on/off) valve is depressed by the driver's left foot heel, compressed air from the brake system's reservoir is delivered to both the brake butterfly valve slave cylinder and the fuel cut-off slave cylinder via the pressure regulator valve, causing both slave pistons to move outwards simultaneously.

The outward movement of the butterfly slave piston causes the butterfly operating lever to rotate about its spindle to close the exhaust passage leading to the silencer. The engine then becomes a single stage low pressure compressor driven through the transmission by the road wheels. The air pressure established in the exhaust manifold and pipe due to blocking the exhaust exit reacts against the piston movement on most of the exhaust stroke, thus producing a retarding torque on the propeller shaft.

The outward movement of the fuel cut-off slave piston rotates the shift bell crank lever, causing the vertical link rod to pull down and fold the two governor link rods. As a result, the change speed lever of the injection pump is moved to the closed or full cut-off position and at the same time the accelerator pedal is drawn towards the floorboard.

The exhaust compression brake remains operative during the whole time the foot control valve is depressed.

Retarder inoperative (Fig. 12.31(b))   When the (on/off) foot valve is released during clutching or declutching, the compressed air in both slave cylinders is exhausted through the control valve. This permits the slave cylinders' respective return springs to open the butterfly valve, and to unfold the governor link rods so that their combined extended length moves the change speed lever to the full delivery position, and at the same time raises the accelerator pedal from the floorboards. The exhaust compression brake system is then inoperative and the engine can be driven normally once again.

12.4.3 Engine compressed air type retarder (acobs) (Fig. 12.33)

A cylinder compression retarder converts a power producing diesel engine into a power absorbing air compressor.

The compressed air engine retarder consists of a hydraulic circuit supplied by the engine oil pump which uses the existing injector rocker motion to open the exhaust valve at the end of the compression stroke via a pair of master and slave pistons actuated by a solenoid valve and a piston control valve.

These compressed air brake units are made to fit on top of the cylinder head and are designed for engines which incorporate combined pump and injector units such as the *Cummins* and   *etroit iesel*

heory of operation   With a conventional engine valve timing, work is done in compressing air on the inward compression stroke. The much reduced volume of air then gives out its energy by driving outwards the piston on its expansion stroke.
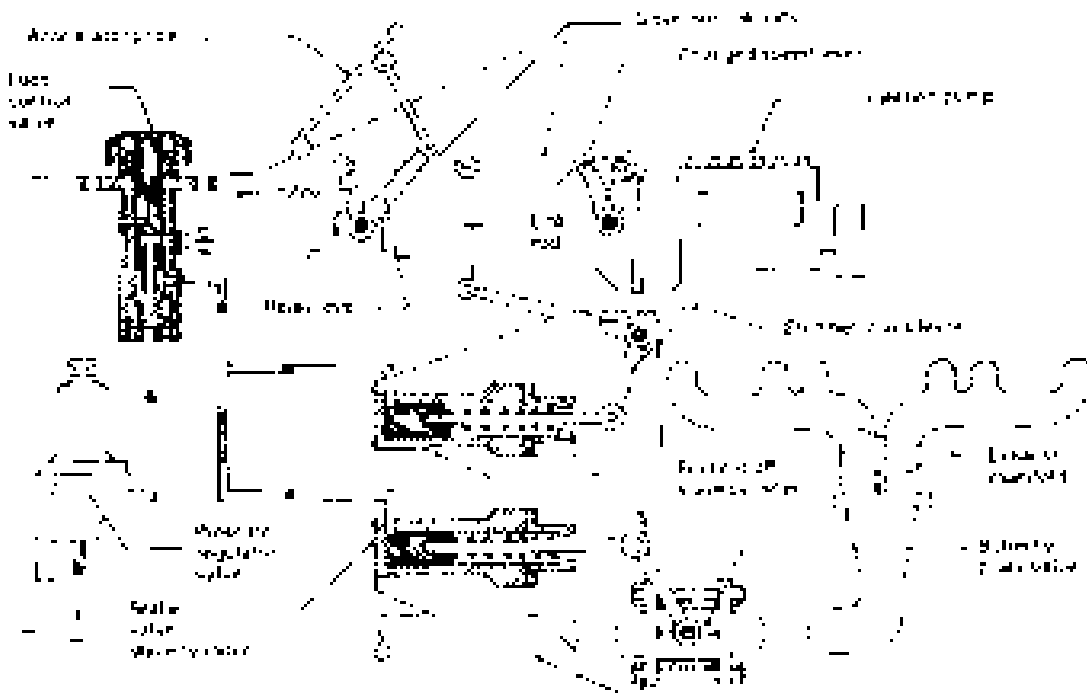
Fig. 12.31 (a and b)   Exhaust compression (brake) type retarder

Therefore, except for frictional losses, there is very little energy lost in rotating the engine on overrun.

The adoption of a mechanism which modifies the exhaust valve timing to open the exhaust valves at the beginning of the expansion stroke causes the release of air into the atmosphere via the exhaust ports (*exhaust blowdown*) (Fig. 12.32), instead of making it do useful work in expanding the piston outwards. The effect of this is a net energy loss so that considerably more effort is required to crank the engine under these conditions.

ngine retarder engaged (e haust blowdown) (Fig. 12.33(a)) When the accelerator pedal is released, the accelerator switch closes to complete the electrical circuit and thereby energizes the solenoid. The solenoid valve closes the oil return passage immediately and opens the passage leading to the control valve. Pressurized oil now pushes the control valve piston up to a point where the annular waste of the control valve piston uncovers the slave piston passage and unseats the ball. Oil at pump pressure therefore passes to the upper crown of the slave and master pistons. This forces the master piston downwards to take up the free movement between itself and the injector rocker adjustment screw. As the camshaft rotates, in the normal cycle of events, the cam lift forces the master piston upwards, causing the ball in the control valve to seat. The trapped contracting volume of oil

between the control valve and master piston now increases its pressure sufficiently to force the slave piston and valve cross head downwards. Consequently both exhaust valves are unseated approximately 1.5 mm and so as a result the exhaust valves open just before the piston reaches TDC position at the end of the compression stroke.

Whilst the engine is on exhaust blowdown, the vehicle speed is reducing and the engine is being overrun by the transmission, so that with the accelerator pedal released and the centrifugal governor weights thrown outwards, fuel is prevented from injecting into the engine cylinders.

ngine retarder disengaged (Fig. 12.33(b)) When the accelerator pedal is depressed, the solenoid circuit is interrupted, causing the de-energized solenoid valve to open the oil pump return passage. The oil pressure under the control valve piston collapses, causing the valve to move to its lowest position. The trapped oil between the master piston and slave piston therefore escapes through the passage opened by the control valve piston. The slave piston is then permitted to rise enabling, the cross head and exhaust valve to operate as normal.

Retarder control A master 'on/off' dash switch in combination with automatic accelerator and clutch switches allows the driver to operate the engine pump retarder under most conditions. Complete release of the accelerator pedal operates the retarder. Depression of the accelerator or clutch pedals opens the electrical circuit, permitting gear changes to be made during descent and prevents the engine from stalling when the vehicle comes to a halt.

12.4.4 ultiplate friction type retarder (Ferodo) (Fig. 12.34)

The retarder is an oil-cooled multiplate brake mounted against the rear end of the gearbox casing. It consists of four steel annular shaped plates with internal locating slots. Both sides of each plate are faced with sintered bronze (Fig. 12.34). These facings have two sets of parallel grooves machined in them at right angles to each other for distribution of the oil. The drive plates are aligned and supported on the slotted output hub, which is itself internally splined at one end to the input shaft and bolted to the output shaft at the other end. This provides a straight-through drive between the gearbox main shaft and the propeller shaft. Support is provided for the output hub and shaft by an inner

Fig. 12.32 Typical pressure/crank diagram for a four stroke engine installed with an engine compressor retarder

Fig. 12.33 (a and b)   Engine compressed air type retarder

551

Fig. 12.34 Multiplate friction type retarder

and outer roller and ball bearing respectively. Interleaved with the driven plates are five cast iron stationary counter plates, also of the annular form, with four outer radial lugs. Four stator pins supported at their ends by the casing are pressed through holes in these lugs to prevent the counter plates rotating, and therefore absorb the frictional reaction torque.

Between the pump housing flange and the friction plate assembly is an annular stainless steel bellows. When oil under pressure is directed into the bellows, it expands to compress and clamp the friction plate assembly to apply the retarder.

The friction level achieved at the rubbing surfaces is a function of the special oil used and the film thickness, as well as of the friction materials.

The oil flow is generated by a lobe type positive displacement pump, housed in the same inner housing that supports the stator pins. The inner member of the pump is concentric with the shaft, to which it is keyed, and drives the outer member. The pump draws oil from the pump pick-up and circulates it through a control valve. It then passes

the oil through a relief valve and a filter (both not shown) and a heat exchanger before returning it to the inlet port. The heat exchanger dissipates its heat energy into the engine cooling system at the time when the waste heat from the engine is at a minimum.

Output torque control (Fig. 12.34) When the spool control valve is in the 'off' position, part of the oil flow still circulates through the heat exchanger, so that cooling continues, but the main flow returns direct to the casing sump. The bellows are vented into the casing, releasing all pressure on the friction surfaces.

When the control valve is moved to the open position, it directs some oil into the bellows at a pressure which is governed by the amount the spool valve shifts to one side. This pressure determines the clamping force on the friction assembly. The main oil flow is now passed through the heat exchanger and into the friction assembly to lubricate and cool the friction plates.

552

12.4.5 Electro-magnetic eddy current type retarder (Telma) (Fig. 12.35(a, b and c))

The essential components are a stator, a support plate, which carries suitably arranged solenoids and is attached either to the chassis for mid-propeller shaft location or on the rear end of the gearbox (Fig. 12.35(a)), and a rotor assembly mounted on a flange hub. The stator consists of a steel dished plate mounted on a support bracket which is itself bolted to a rear gearbox flange. On the outward facing dished stator plate side are fixed eight solenoids with their axis parallel to that of the transmission. The rotor, made up of two soft steel discs facing the stator pole pieces, is bolted to a hub which is supported at the propeller shaft end by a ball bearing and at its other end by the gearbox output shaft. The drive from the gearbox output shaft is transferred to the propeller universal joint via the internally splined rotor hub sleeve. The rotor discs incorporate spiral shaped (turbine type blades) vanes to provide a large exposed area and to induce airflow sufficient to dissipate the heat generated by the current induced in the rotor and that produced in the stationary solenoid windings.

Four independent circuits are energized by the vehicle's battery through a relay box, itself controlled by a fingertip lever switch usually positioned under the steering wheel (Fig. 12.35(b and c)). These solenoid circuits are arranged in parallel as an added safety precaution because, in the event of failure of one circuit, the unit can still develop three-quarters of its normal power. The control lever has four positions beside 'off' which respectively energize two, four, six and eight poles. The solenoid circuit consumption is fairly heavy, ranging for a typical retarder from 40 to 180 amperes for a 12 volt system.

Operating principles (Fig. 12.36) If current is introduced to each pole piece winding, a magnetic flux is produced which interlinks each of the winding loops and extends across the air gap into the steel rotor disc, joining up with the flux created from adjacent windings (Fig. 12.36).

When the rotors revolve, a different section of the disc passes through the established flux so that in effect the flux in any part of the disc is continuously varying. As a result, the flux in any one segmental portion of the disc, as it sweeps across the faces of the pole pieces, increases and then decreases in strength as it moves towards and then away from the established flux field. The change in flux linkage with each segmental portion of the disc which passes an adjacent pole face induces an electromotive force (voltage) into the disc. Because the disc is an electrical conductor, these induced voltages will cause corresponding induced currents to flow in the rotor disc. These currents are termed *eddy currents* because of the way in which they whirl around within the metal.

Collectively the eddy currents produce an additional interlinking flux which opposes the motion of the rotor disc. This is really Lenz's Law which states that the direction of an induced voltage is such as to tend to set up a current flow, which in turn causes a force opposing the change which is producing the voltage. In other words, the eddy currents oppose the motion which produced them. Thus the magnetic field set up by these solenoids create eddy currents in the rotor discs as they revolve, and then eddy currents produce a magnetic drag force tending to slow down the rotors and consequently the propeller shaft (Fig. 12.36).

The induced eddy currents are created inside the steel discs in a perpendicular direction to the flux, and therefore heat (Rt) is produced in the metal.

The retarding drag force or resisting torque varies with both the rotational speed of the rotor and propeller shaft and the strength of the electromagnetic field, which is itself controlled by the amount of current supplied.

12.4.6 Hydraulic type retarder (Voith) (Fig. 12.37)

The design of a hydraulic retarder is similar to that of a fluid coupling. Basically, the retarder consists of two saucer-shaped discs, a revolving rotor (or impellor) and a stationary stator (or reaction member) which are cast with a number of flat radial vanes or blades for directing the flowpath of the fluid. The rotor is bolted to the flange of the internally splined drive shaft hub, which is itself mounted over the external splines formed on both the gearbox mainshaft and the flanged output shaft, thereby coupling the two drive members together. Support to the drive shaft hub and rotor is given by a roller bearing recessed in the side of the stator, which is in turn housed firmly within the retarder casing.

Theory of operation (Fig. 12.37) The two half-saucer members are placed face to face so that fluid can rotate as a vortex within the cells created by the radial vanes (Fig. 12.37).

When the transmission drives the rotor on overrun and fluid (oil) is introduced into the spaces between the rotor and stator, the fluid is subject to centrifugal force causing it to be accelerated

Fig. 12.35 (a–c)   Electric eddy current type retarder
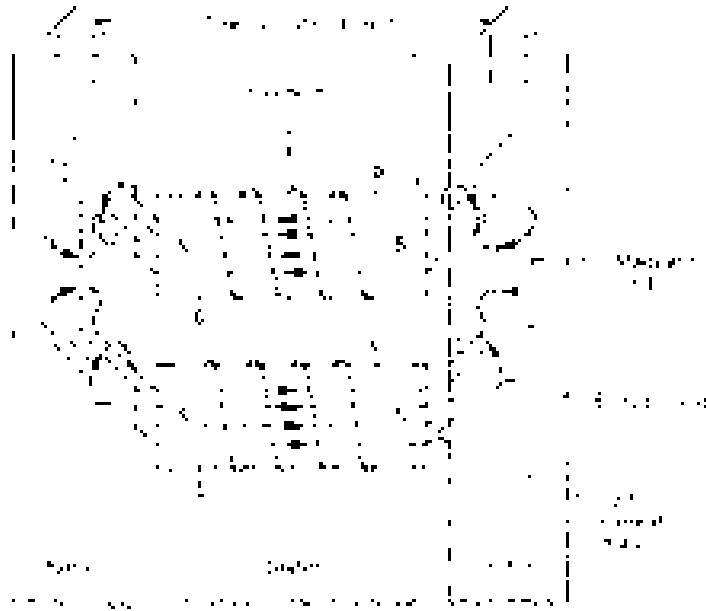
Fig. 12.36 Principle of electric eddy current retarder

radially outwards. As the fluid reaches the outmost periphery of the rotor cells, it is flung across the junction made between the rotor and stator faces. It then decelerates as it is guided towards the inner periphery of the rotor cells to where the cycle of events once again commences. The kinetic energy imparted to the fluid passing from the revolving rotor to the fixed stator produces a counter reaction against the driven rotor. This counter reaction therefore opposes the propelling energy at the road wheels developed by the momentum of the moving vehicle, causing the vehicle to reduce speed.

The kinetic energy produced by the rapidly moving fluid as it impinges onto the stator cells, and the turbulance created by the movement of the fluid between the cells is all converted into heat energy. Hence the kinetic energy of the vehicle is converted into heat which is absorbed by the fluid and then dissipated via a heat exchanger to the cooling system of the engine.

The poor absorption capacity of the hydraulic retarder increases almost with the cube of the propeller shaft speed for a given rotor diameter.

When the retarder is not in use the rotor rotates in air, generating a drag. In order to keep this drag as low as possible, a number of stator pins are mounted inside and around the stator cells. These disc-headed pins tend to interfere with the air circulating between the moving and stationary half-cells when they have been emptied of fluid,

thereby considerably reducing the relatively large windage losses which normally exist.

Output torque control (Fig. 12.37) In order to provide good retardation at low speeds, the retarder is designed so that maximum braking torque is reached at approximately a quarter of the maximum rotor speed/propeller speed. However, the torque developed is proportional to the square of the speed, and when the vehicle speed increases, the braking torque becomes too great and must, therefore, be limited. This is simply achieved by means of a relief valve, controlling the fluid pressure which then limits the maximum torque.

The preloading of the relief valve spring is increased or reduced by means of an air pressure-regulated servo assisted piston (Fig. 12.37). The control valve can be operated either by a hand lever when the unit is used as a continuous retarder, or by the foot control valve when it is used for making frequent stops.

When the retarder foot control valve is depressed, air from the auxiliary air brake reservoir is permitted to flow to the servo cylinder and piston. The servo piston is pushed downwards relaying this movement to the relief spill valve via the inner spring. This causes the relief valve spool to partially close the return flow passage to the sump and to open the passage leading to the inner

555
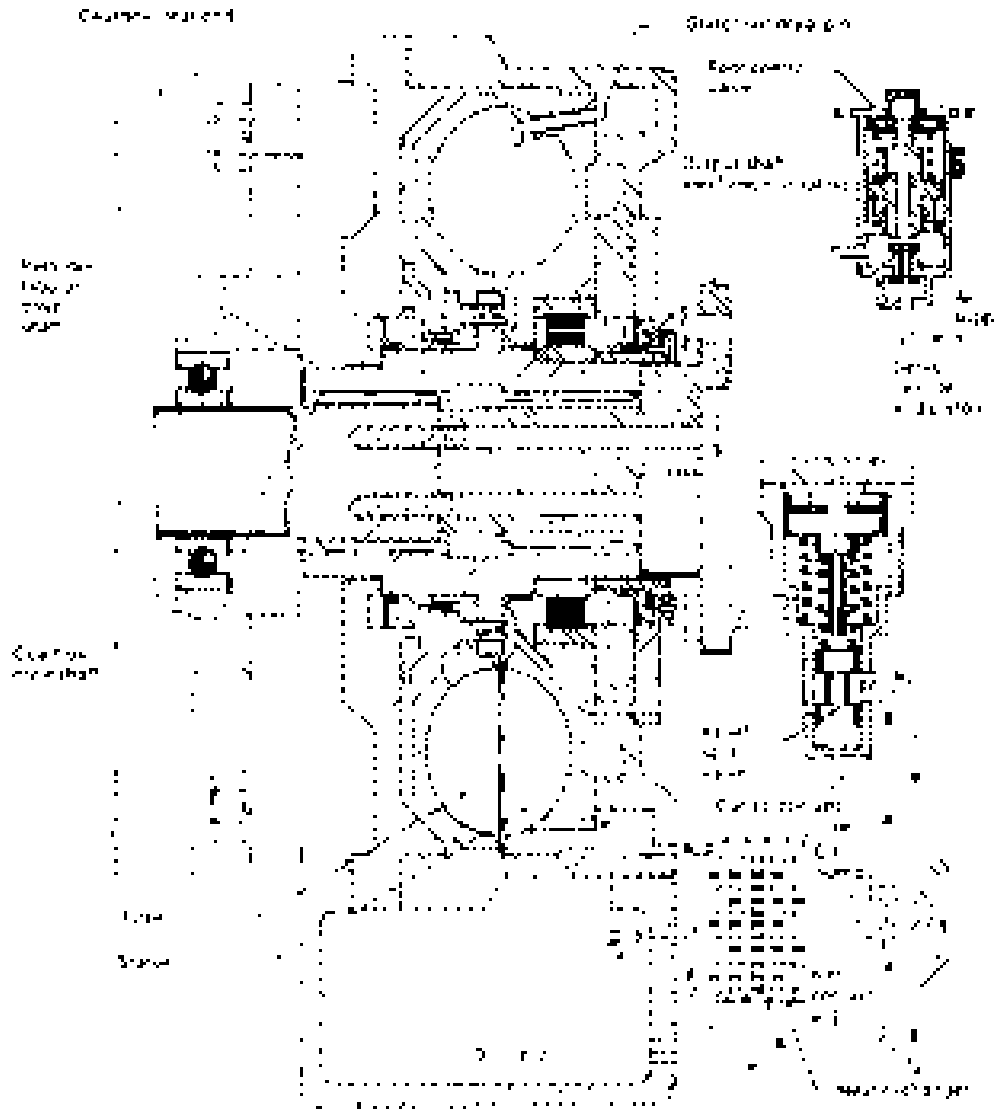
Fig. 12.37 Hydraulic type retarder

periphery of the stator. Fluid (oil) from the hydraulic pump now fills the rotor and stator cells according to the degree of retardation required, this being controlled by the foot valve movement. At any foot valve setting equilibrium is achieved between the air pressure acting on top of the servo piston and the opposing hydraulic pressure below the spool relief valve, which is itself controlled by the hydraulic pump speed and the amount of fluid escaping back to the engine's sump. The air feed pressure to the servo piston therefore permits the stepless and sensitive selection of any required retarding torque within the retarder's speed/torque characteristics.

Should the oil supply pressure become excessively high, the spool valve will lift against the control air pressure, causing the stator oil supply passage to partially close while opening the return flow passage so that fluid pressure inside the retarder casing is reduced.

12.4.7 A comparison of retarder power and torque absorbing characteristics
(Figs 12.38 and 12.39)
Retarders may be divided into those which utilize the engine in some way to produce a retarding effort and those which are mounted behind the

556
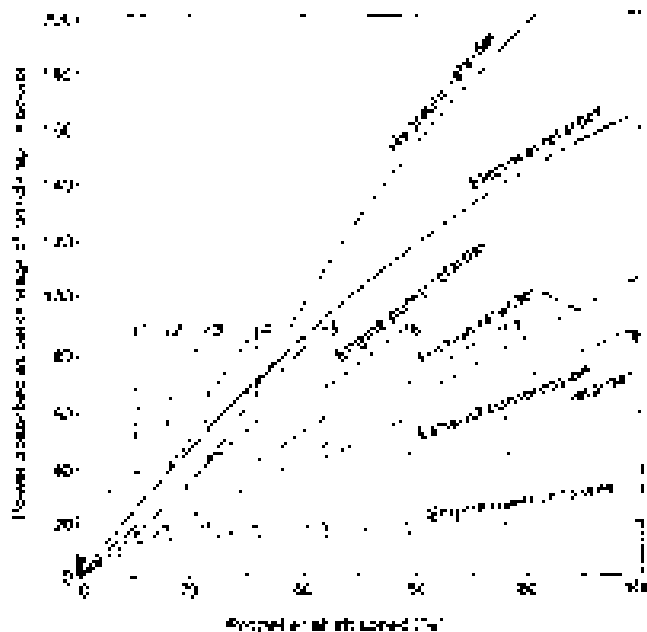
Fig. 12.38 Typical comparison of power absorption of various retarders relative to propellor shaft speed

gearbox, between the propeller shafts or in front of the final drive.

Retarders which convert the engine into a pump, such as the exhaust compression type or engine compressed air acobs type retarders, improve their performance in terms of power and torque absorption by using the gear ratios on overrun similarly to when the engine is used to propel the vehicle forwards. This is shown by the sawtooth power curve (Fig. 12.38) and the family of torque curves (Fig. 12.39) for the engine pump acobs type retarder. In the cases of the exhaust compression type retarder and engine overrun loss torque curves, the individual gear ratio torque curves are all shown merged into one for simplicity. Thus it can be seen that three methods, engine compressed air, exhaust compression and engine overrun losses, which use the engine to retard the vehicle, all depend for their effectiveness on the selection of the lowest possible gear ratio without over speeding the engine. As the gear ratio becomes more direct, the torque multiplication is reduced so that there is less turning resistance provided at the propeller shaft.

For retarders installed in the transmission after the gearbox there is only one speed range. It can be seen that retarders within this classification, such as the multi-friction plate, hydrokinetic and electrical eddy current type retarders all show an increase in power absorption in proportion to propeller shaft speed (Fig. 12.38). The slight deviation from a complete linear power rise for both hydraulic and electrical retarders is due to hydrodynamic and eddy current stabilizing conditions. It can be seen that in the lower speed range the hydraulic retarder absorbs slightly less power than the electrical retarder, but as the propeller shaft rises this is reversed and the hydraulic retarder absorbs proportionally more power, whereas the multiplate friction retarder produces a direct increase in power absorption throughout its speed range, but at a much lower rate compared to hydraulic and electrical retarders because of the difficulties in dissipating the generated heat.

When considering the torque absorption characteristics of these retarders (Fig. 12.39), the electrical retarder is capable of producing a high retarding torque when engaged almost immediately as the propeller shaft commences to rotate, reaching a peak at roughly 10% of its maximum speed range. It then declines somewhat, followed by a relatively constant output over the remainder of its speed range. However, the hydraulic retarder shows a slower resisting torque build-up which then gently exceeds that of the eddy current resisting torque curve, gradually reaching a peak followed by a very small decline as the propeller shaft speed approaches a maximum. In comparison to the

557
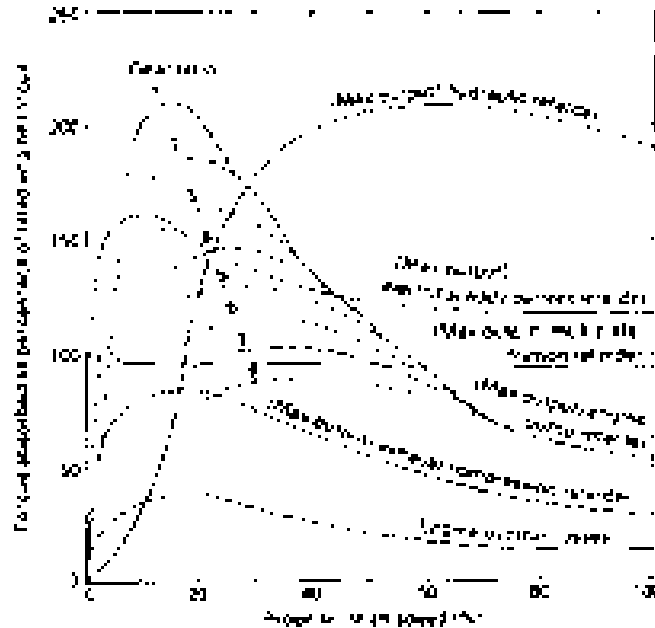
Fig. 12.39 Typical comparison of torque produced by various retarders relative to propeller shaft

other retarders, the multiplate friction retarder provides a resisting torque the instant the two sets of friction plates are pressed together. The relative slippage between plates provides the classical static high friction peak followed immediately by a much lower steady dynamic frictional torque which tends to be consistant throughout the retarder's operating speed range. What is not shown in Figs 12.38 and 12.39 is that the electrical, hydraulic and friction retarder outputs are controlled by the driver and are generally much reduced to suit the driving terrain of the vehicle.

12.5 lectronic-pneumatic brakes

12.5.1 *ntroduction to electronic-pneumatic brakes* (Fig. 12.40)

The electronic-pneumatic brake (EPB) system controls the entire braking process; this includes ABS/TCS braking when conditions demand, and the layout consists of a single electronic-pneumatic brake circuit with an additional dual pneumatic circuit. The electronic-pneumatic part of the braking system is controlled via various electronic sensors: (1) brake pedal travel; (2) brake air pressure; (3) individual wheel speed; and (4) individual lining/pad wear. Electronic-pneumatic circuit braking

does not rely on axle load sensing but relies entirely on the wheel speed and air pressure sensing.

The dual pneumatic brake system is split into three independent circuits known as the redundancy braking circuit, one for the front axle a second for the rear axle and a third circuit for trailer control. The dual circuit system is similar to that of a conventional dual line pneumatic braking system and takes over only if the electronic-pneumatic brake circuit should develop a fault. Hence the name redundancy circuit, since it is installed as a safety back-up system and may never be called upon to override the electronic-pneumatic circuit brakes. However, there will be no ABS/TCS function when the dual circuit redundancy back-up system takes over from the electronic-pneumatic circuit when braking.

The foot brake pedal movement corresponds to the driver's demand for braking and is monitored by the electronic control module (ECM) which then conveys this information to the various solenoid control valves and axle modules (AM); compressed air is subsequently delivered to each of the wheel brake actuators. Only a short application lag results from the instant reaction of the electronic-pneumatic circuit, and consequently it reduces the braking distance in comparison to a conventional pneumatic braking system.
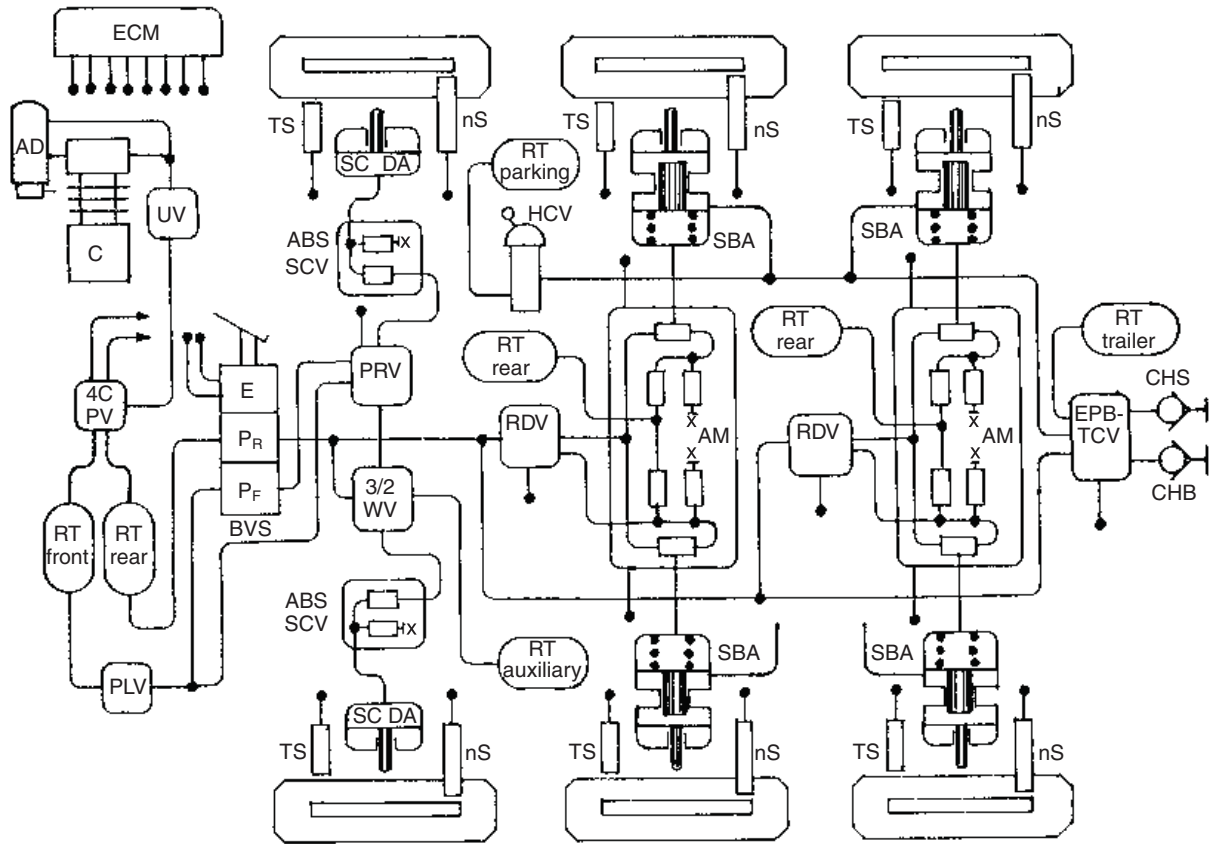
A list of key components and abbreviations used in the description of the electronic-pneumatic brake system is as follows:

1	Electronic control module	ECM	13	Axle modulator	AM
2	Air dryer	AD	14	Spring brake actuator	SBA
3	Compressor	C	15	EPB trailer control valve	EPB-TCV
4	Unloader valve	UV	16	Park hand control valve	P-HCV
5	Four circuit protection valve	4CPV	17	Coupling head for supply	CHS
6	Reservoir tank (front/rear/trailer/auxiliary/parking)	RT etc	18	Coupling head for brake	CHB
7	Brake value sensor	BVS	19	Travel sensor	TS
8	Proportional relay valve	PRV	20	Speed sensor	nS
9	3/2-way valve for auxiliary braking effect	3/2-WV-AB	21	Pneumatic control front	P_F
10	ABS solenoid control valve	ABS-SCV	22	Pneumatic control rear	P_R
11	Single circuit diaphragm actuator	SCDA	23	Electrical sensors & switches	E
12	Redundancy valve	RDV	24	Air exit (exhaust)	x

Fig. 12.40 Electronic-pneumatic brake component layout

The electronic-pneumatic part of the braking system broadly divides the braking into three operation conditions:

1 Small differences between wheel speeds under part braking conditions; here the brake lining-disc wear is optimized between the front and rear axles.
2 Medium differences between wheel speeds; here the difference in wheel speed is signalled to the controls, causing wheel slip to be maintained similar on all axles. This form of brake control is known as adhesive adapted braking.
3 Large differences between wheel speeds and possibly a wheel locking tendency; here the magnitude of the spin-lock on each wheel is registered, triggering ABS/TCS intervention.

Note antilocking braking system (ABS) prevents the wheels from locking when the vehicle rapidly decelerates whereas a traction control system (TCS)

prevents the wheels from spinning by maintaining slip within acceptable limits during vehicle acceleration.

The single circuit electronic-pneumatic brake circuit consists of the following:

1 Compressed air supply, the engine driven reciprocating compressor supplies and stores compressed air via the four circuit protection valve and numerous reservoir tanks. The compressor regulator cut-in and cut-out pressures are of the order of 10.2 bar and 12.3 bar respectively. Service foot circuits operate approximately at 10 bar whereas the parking and auxiliary circuits operate at a lower pressure of around 8.5 bar.
2 Electronic control module (ECM). This unit determines the brake force distribution corresponding to the load distribution. It is designed to receive signal currents from the following sources: foot travel sensors (TS), front axle, rear axles and trailer control air pressure sensors (PS) in addition to the individual wheel travel and speed sensors (nS). These inputs are processed and calculated to simultaneously provide the output response currents needed to activate the various electronically controlled components to match the braking requirements, such control units being the proportional relay valve (PRV), redundancy valve (RDV), front axle ABS solenoid control valves (ABS-SCV), rear axle module (AM) and the EPB trailer control valve (EPB-TCV).
3 Brake value sensor (BVS) unit which incorporates the pedal travel sensors (TS) and brake switches (BS) in addition to the dual circuit foot brake valve.
4 Redundancy valve (RDV): this valve switches into operating the rear axle dual circuit lines if a fault occurs in the electronic-pneumatic brake circuit.
5 Rear axle electronic-pneumatic axle module (AM) incorporating inlet and outlet solenoid valves used to control the application and release of the rear axle brakes.
6 Electronic-pneumatic proportional relay valve (PRV). This unit incorporates a solenoid relay valve which controls the amount of braking proportional to the needs of the front axle brakes.
7 Two front axle ABS solenoid control valves (ABS-SCV) which control the release and application of the front axle brakes.
8 Electronic-pneumatic brake-trailer control valve (EPB-TCV). This valve operates the trailer brakes via the trailer's conventional relay emergency valve during normal braking.

9 Parking hand control valve (P-HCV) which controls the release and application of the rear axle's and trailer axle's conventional spring brake part of the wheel brake actuators.
10 Pressure limiting valve (PLV). This unit reduces the air pressure supply to the front axle of the towing vehicle when the semi-trailer is de-coupled in order to reduce the braking power and maintain vehicle stability of the now much lighter vehicle.

A description explaining the operation of the electronic-pneumatic braking system now follows:

12.5.2 Front axle braking (Fig. 12.41(a–d))

ront a le foot brake released (Fig. 12.41(a)) When the brake pedal is released the foot travel sensors signal the electronic control module (ECM) to release the brake, accordingly the proportional relay valve is de-energized. As a result the proportional valve's (of the proportional relay valve unit) upper valve opens and its lower inlet valve and exit valves close and open respectively, whereas the relay valve's part of the proportional relay valve unit inlet closes and its exit opens. Hence air is released from the right hand wheel brake-diaphragm actuator via the right hand ABS solenoid control valve and the proportional relay valve exit, whereas with the left hand wheel brake-diaphragm actuator, compressed air is released via the left hand ABS solenoid control valve, 3/2-way valve and then out by the proportional relay valve exit.

ront a le foot brake applied (Fig. 12.41(b)) Air supply pressure from the front axle reservoir is directed to both the brake value sensor (BVS) and to the proportional relay valve (PRV).

When the driver pushes down the front brake pedal, the travel sensors incorporated within the brake value sensor (BVS) simultaneously measure the pedal movement and relay this information to the electronic control module (ECM). At the same time the brake switches close, thereby directing the electronic control module (ECM) to switch on the stop lights. Instantly the electronic control module (ECM) responds by sending a variable control current to the proportional valve situated in the proportional relay valve (PRV) unit. The energized solenoid allows the top valve to close whereas the lower control valve partially opens. Electronic-pneumatic control pressure now enters the relay valve's upper piston chamber, causing its piston

to close the air exit and partially open the control valve, thereby permitting pre-calculated controlled brake pressure to be delivered to the wheel-diaphragm actuators via the ABS solenoid control valves for the right hand wheel and via the 3/2-way valve for auxiliary braking effect and the ABS solenoid control valve for the left hand wheel. For effective controlled braking the individual wheel speed sensors provide the electronic control module (ECM) with instant feed-back on wheel retardation and slip; this with the brake pedal movement sensors and pressure sensors enable accurate brake pressure control to be achieved at all times. Note the electronic-pneumatic brake (EPB) circuit has priority over the pneumatic modulated front pressure regulated by the brake value sensor (BVS) unit.

ront a le foot brake applied under ABS/ CS conditions (Fig. 12.41(c)) If the brakes are applied and the feed-back from the front axle speed sensors indicates excessive lock/slip the electronic control module will put the relevant ABS solenoid control valve into ABS mode. Immediately the ABS solenoid control valve attached to the wheel axle experiencing unstable braking energizes the solenoid valve, causing its inlet and exit valves to close and open respectively. Accordingly the wheel brake-diaphragm actuator will be depressurized thus avoiding wheel lock. The continuous monitoring of the wheel acceleration and deceleration by the electronic control module calculates current signal response to the ABS solenoid control valve to open and close respectively the inlet and exit valves, thus it controls the increase and decrease in braking pressure reaching the relevant wheel brake-diaphragm actuator; consequently the tendency of wheel skid is avoided.

ront a le foot brake applied with a fault in the electronic-pneumatics (Fig. 12.41(d)) If a fault develops in the electronic-pneumatic system the proportional relay valve shuts down, that is the solenoid proportional valve is de-energized causing its inlet valve to close and for its exit valve to open. Consequently when the brakes are applied the proportional relay valve's relay piston chamber is depressurized, making the relay valve's inlet and exit to close and open respectively. As a result, with the right hand ABS solenoid control valves de-energized air will exhaust from the right hand wheel brake-diaphragm actuator via the ABS sole-

noid control valve and the proportional relay valve. However, the collapse of the electro-pneumatic control pressure in the proportional relay valve causes the closure of the 3/2-way valve passage connecting the proportional relay valve to the left hand wheel brake actuator and opens the passages joining the auxiliary relay valve to the left hand wheel brake actuator via the left hand ABS solenoid control valve. Thus if the supply pressure from the front axle brake circuit is interrupted, the redundancy (pneumatic) rear axle brake pressure regulated by the brake valve sensor's foot control valve shifts over the 3/2-way valve into auxiliary braking effect position, that is, the 3/2-way valve blocks the passage between the proportional relay valve and the ABS solenoid control valve and then supplies modulated brake pressure from the 3/2-way valve to the left hand wheel brake-diaphragm actuator. Therefore the left hand front axle brake only, is designed to support the rear axle braking when the electronic-pneumatic brake circuit fails.

ront a le braking without trailer attached (Fig. 12.41(a–d)) When the semi-trailer is disconnected from its tractor the electronic control module responds by energizing the pressure-limiting valve solenoid. This results in the solenoid valve closing the direct by-pass passage leading to the proportional relay valve and opening the valve leading to the relay valve within the pressure-limiting valve unit (see Fig. 12.41(c)). This results in the solenoid valve shutting-off the front axle reservoir tank air supply from the proportional relay valve and at the same time re-routing the air supply via the pressure-limiting valve's relay valve which then reduces the maximum braking pressure reaching the proportional relay valve and hence the front axle brakes.

Limiting the air pressure reaching the front axle of the towing vehicle when the trailer is removed is essential in retaining the balance of front to rear axle braking power of the now much shorter overall vehicle base, thereby maintaining effective and stable vehicle retardation.

12.5.3 Rear axle braking (ig. 12.42(a–d))

Rear a le — foot brake released (Fig. 12.42(a)) When the brake pedal is released the travel sensors within the brake value sensor (BVS) signal the electronic control module (ECM) which in turn informs the axle modulator to release the brake

(a) Front axle – foot brake released
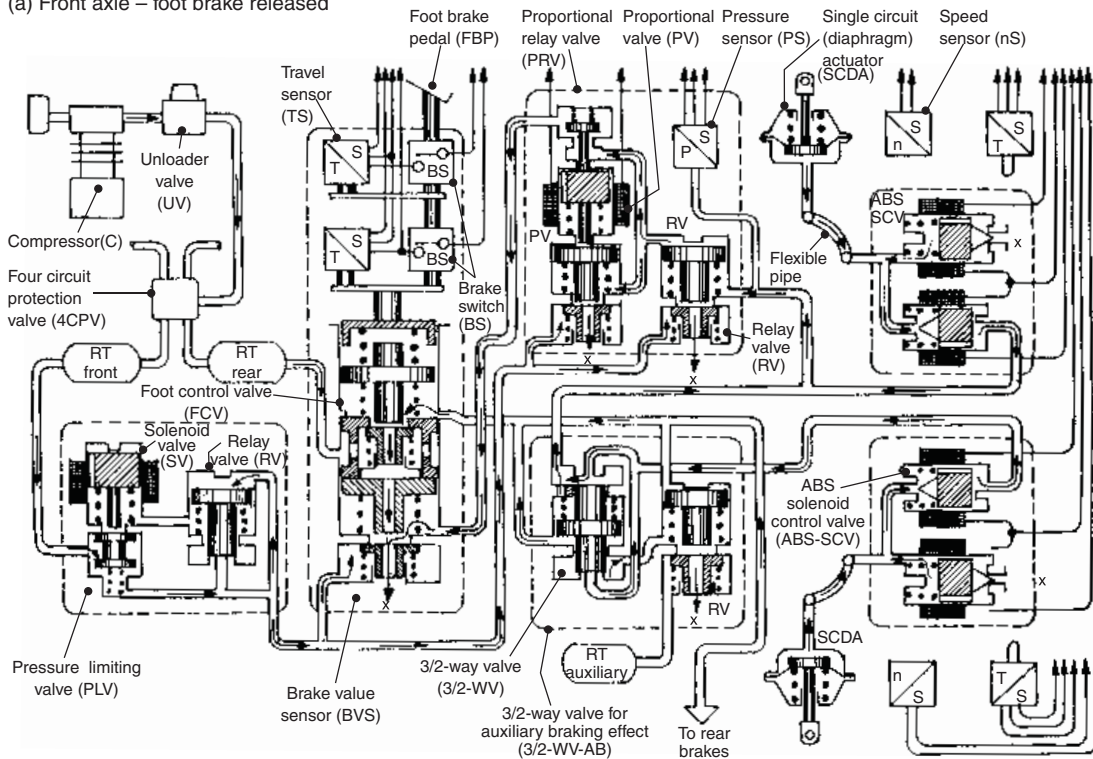
(b) Front axle – foot brake applied (normal brake operation)
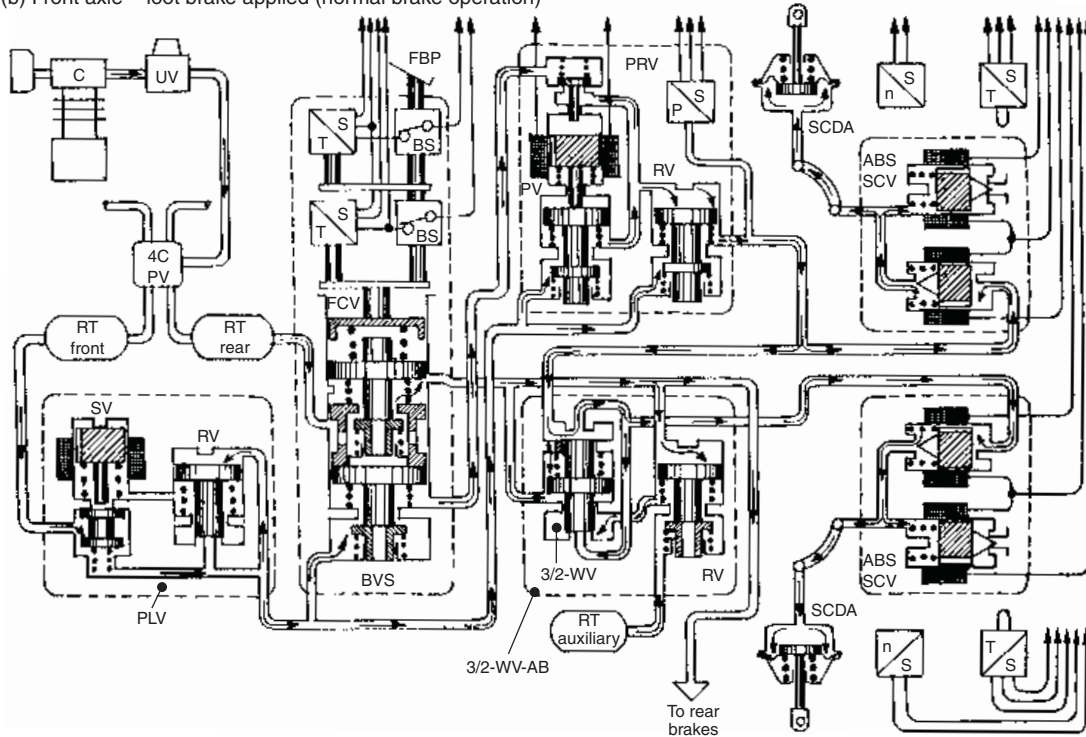
Fig. 12.41 (a and b) Electronic-pneumatic front brake system

(c) Front axle – foot brake applied under ABS/TCS conditions
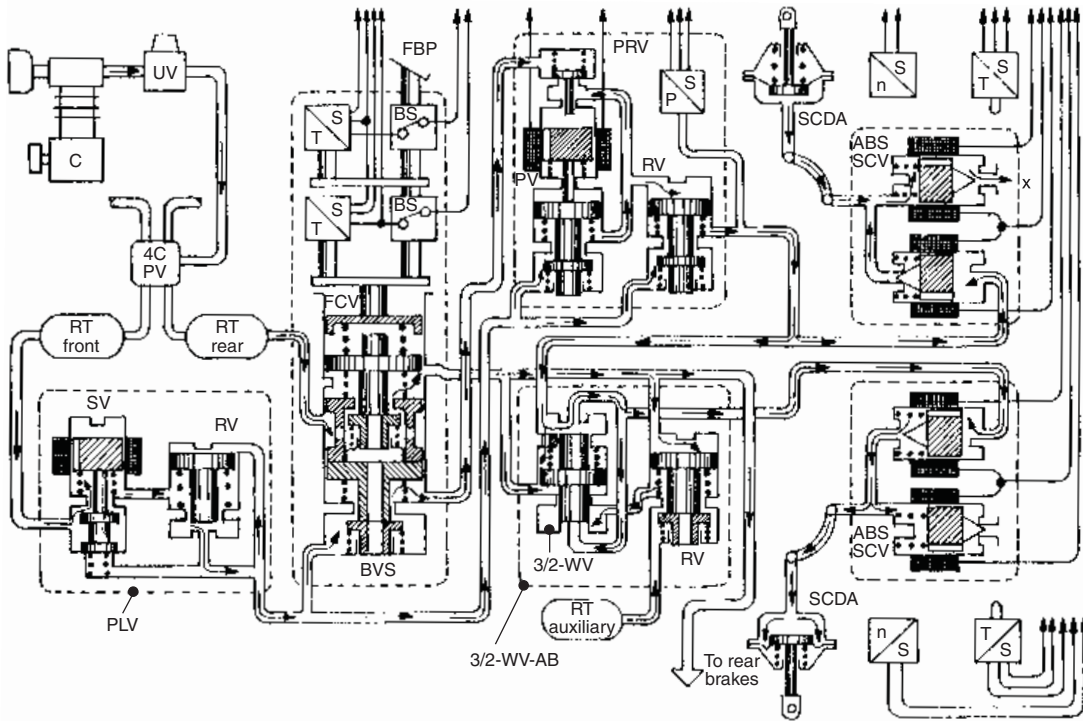
(d) Front axle – foot brake applied with a fault in the electronic-pneumatics
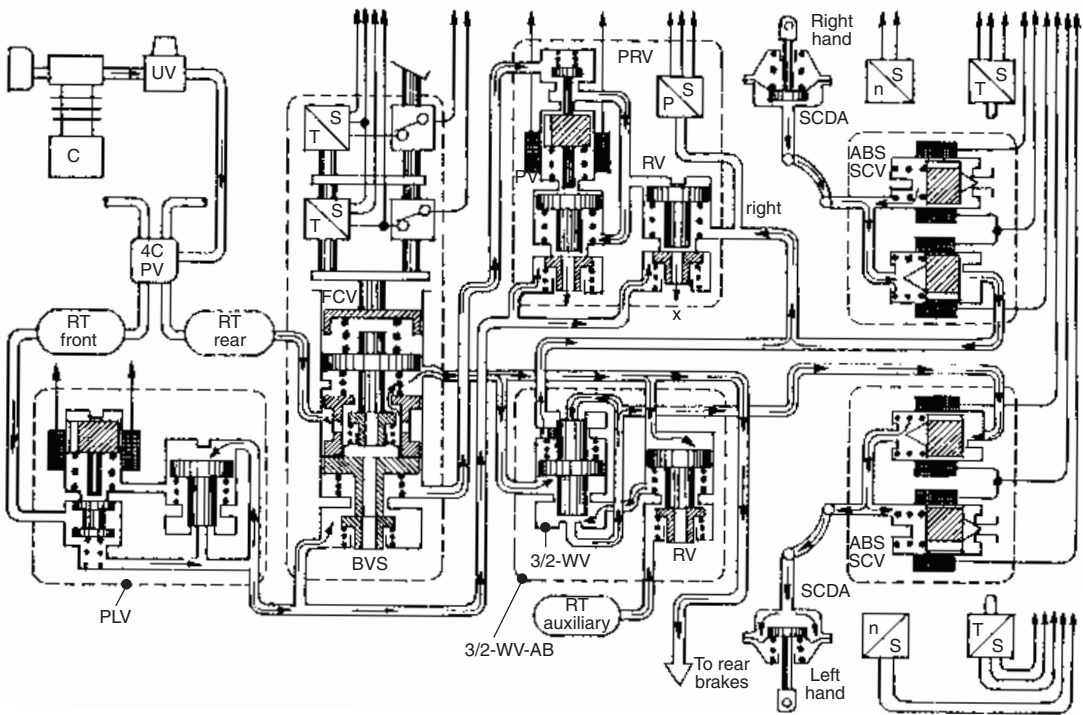
Fig. 12.41 (c and d) *Contd*

(a) Rear axle – foot brake released

2-way valve (2WV)

Dual circuit spring brake actuator (SBA)

Inlet / exhaust solenoid valve (in/ex SV)

FBP

RT rear

RT rear

Relay valve (RV)

Brake value sensor (BVS)

FCV

RT rear

2/2 solenoid valve (2/2 SV)

3/2-way valve (3/2 WV)

RT front

Redundancy valve (RDV)

To front brakes

To trailer brakes

Axle modulator for drive axle (AM)

2WV

Parking spring brake

SBA

Foot brake

(b) Rear axle – foot brake applied (normal brake operation)

FBP

2WV

in/ex SV

SBA

RT rear

RT rear

RV

BVS

FCV

3/2 WV

2/2 SV

RT rear

RT front

RDV

To front brakes

To trailer brakes

in/ex SV

SBA

AM

2WV

Fig. 12.42 (a and b) Electronic-pneumatic rear brake system

564

(c) Rear axle – foot brake applied under ABS/TCS conditions
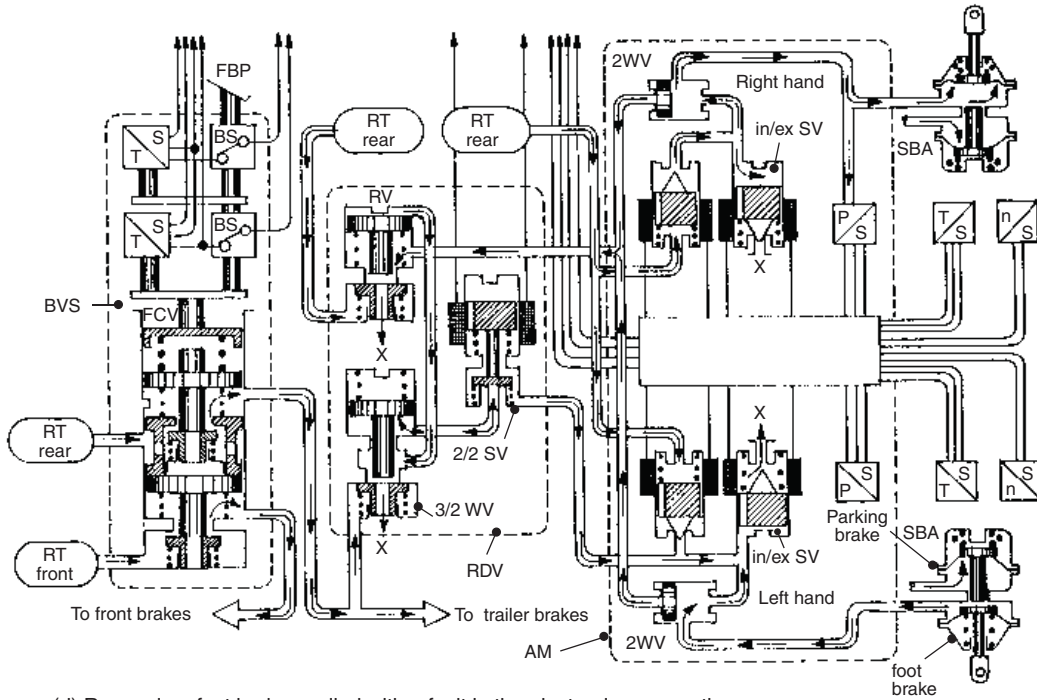
(d) Rear axle – foot brake applied with a fault in the electronic-pneumatics
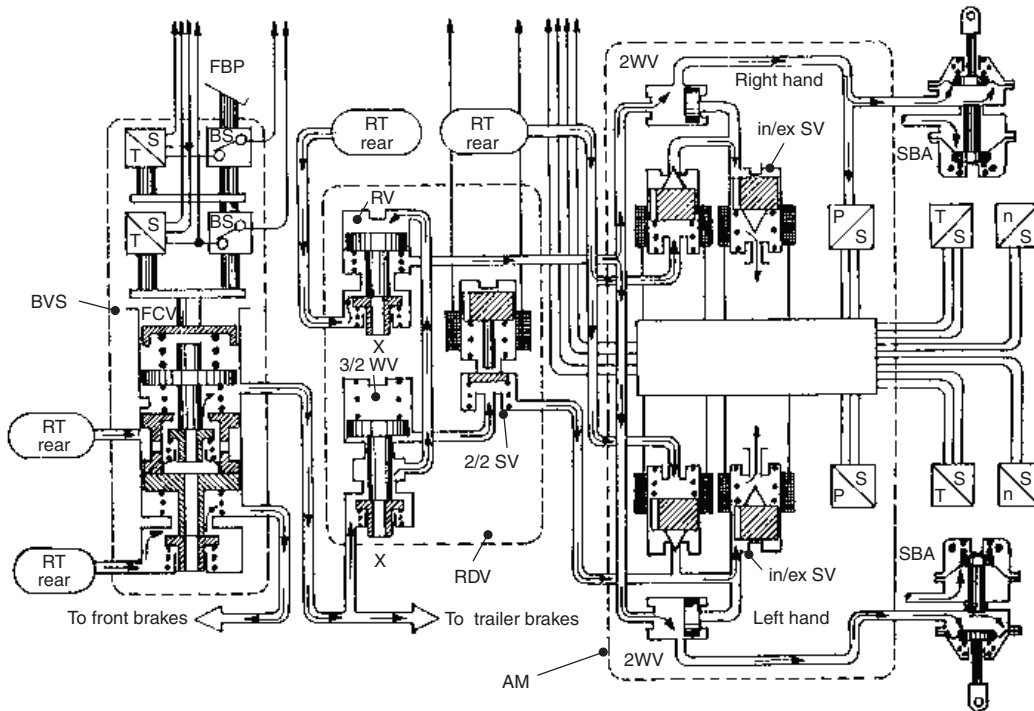
Fig. 12.42 (c and d) *Contd*

565

(a) Trailer axles – foot brake released
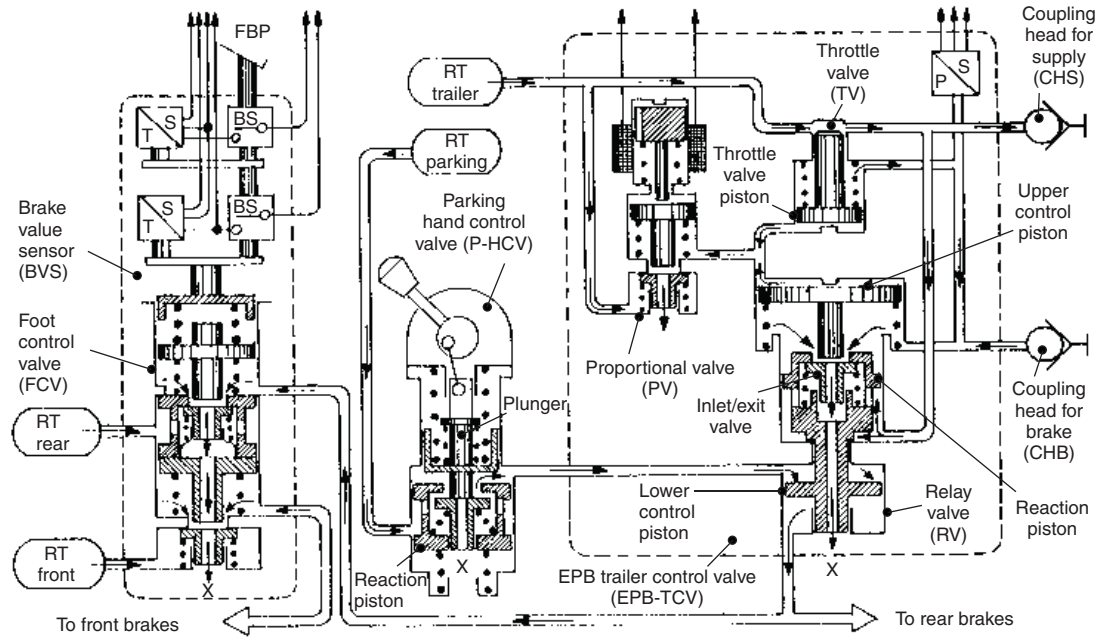
(b) Trailer axles – foot brake applied (normal brake operation)
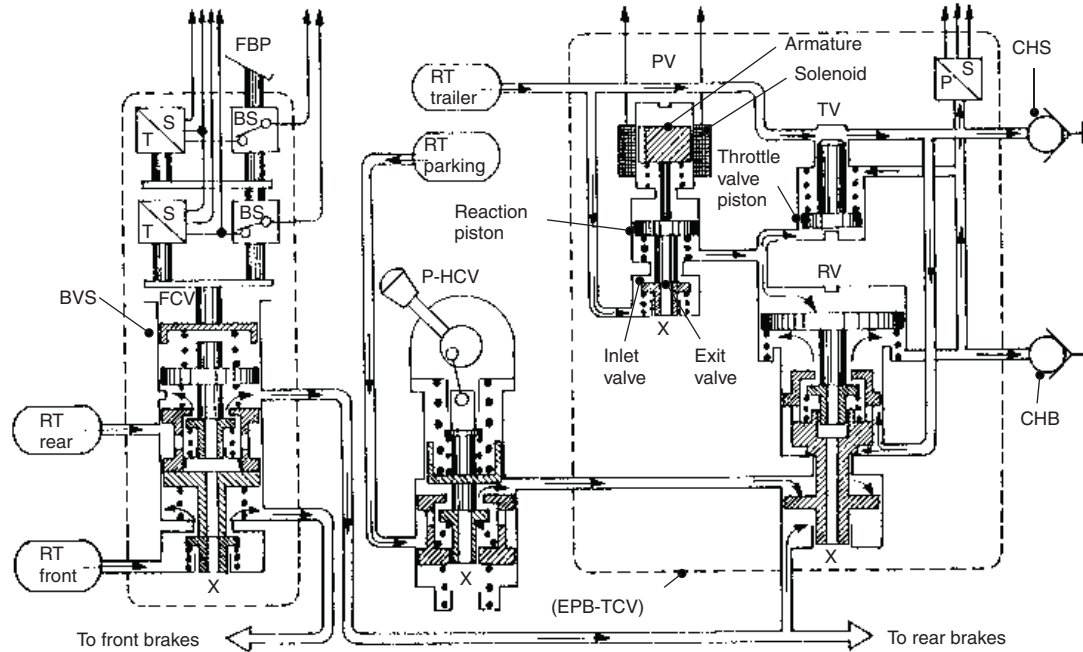
Fig. 12.43 (a and b)   Electronic-pneumatic tractor unit brakes coupled to towed trailer

(c) Trailer axles – parking brake applied
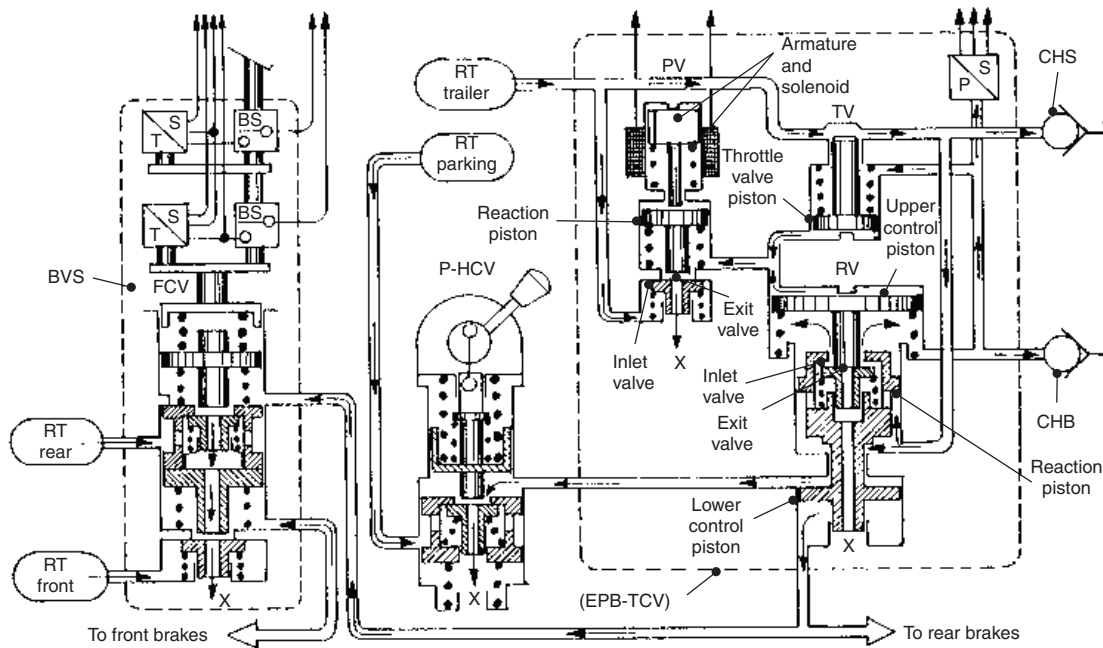
(d) Trailer axles – foot brake applied with a fault in the electronic-pneumatics
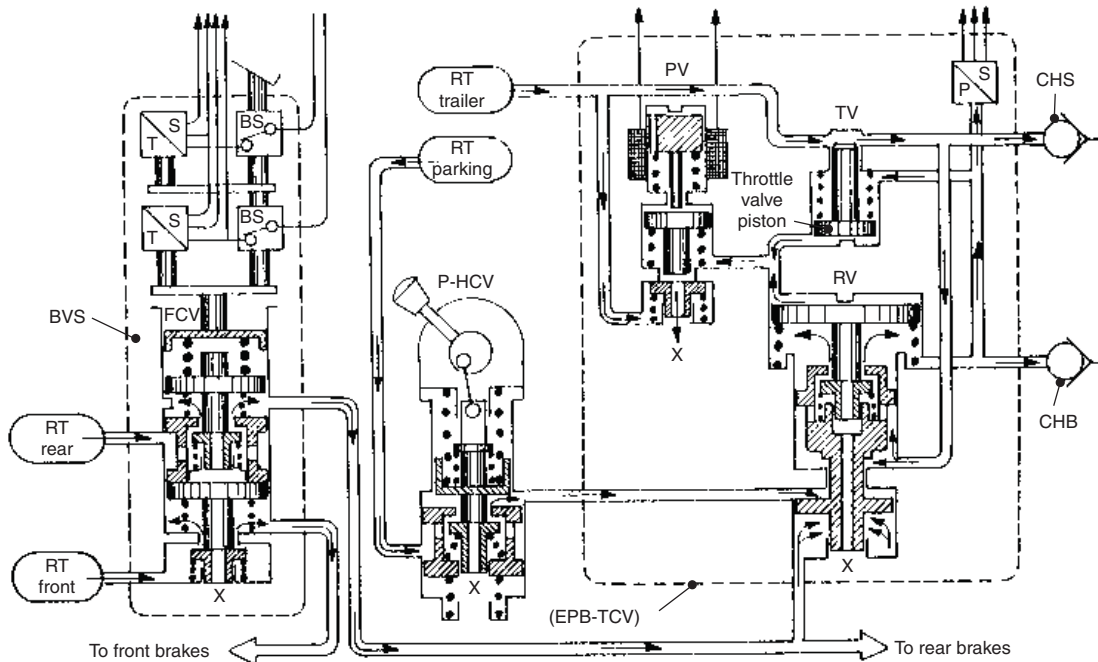
Fig. 12.43 (c and d)   *Contd*

pressure in the wheel spring brake actuator brake lines. Consequently the axle modulator (AM) de-energizes the inlet and exhaust solenoid valves, causing the inlet valve to close and the exhaust valve to open, hence brake pressure will be prevented from reaching the spring brake actuator and any existing air pressure in the spring brake actuator (SBA) will be expelled via the exhaust solenoid valve. Air pressure in the pipe lines between the brake value sensor, the redundancy valve and axle modulator unit will also be exhausted by way of the foot control valve exit and the axle modulator (AM) exhaust solenoid valve exit.

Rear a le — foot brake applied (Fig. 12.42(b)) The rear axle reservoir tank delivers maximum supply pressure to the brake value sensor (BVS), the redundancy valve (RDV) and to the rear axle modulator (AM).

When the foot brake pedal is applied the travel sensor within the brake value sensor unit monitors the pedal movement and relays this information to the electronic control module. The brake switches will also close thereby informing the electronic control module (ECM) to operate the stop lights. The electronic control module (ECM) responds by signalling the axle modulators (AMs) to energize their corresponding inlet/exhaust solenoid valves. The exhaust valve will therefore close, whereas the inlet valves now open to permit rear reservoir tank supply pressure to flow via the 2-way valve to the dual circuit spring brake actuators (SBA) thereby operating the brakes. In addition, brake pressure is conveyed to the redundancy valve (RDV) where it flows though the 2/2 solenoid valve and then actuates the 3/2-way valve. This closes the 3/2-way valve, preventing redundancy circuit (pneumatic control pressure) control pressure reaching the axle modulator solenoid valves and at the same time exhausts the air holding down the relay valve's piston, hence it causes the relay valve to block the rear axle redundancy valve reservoir tank supply pressure entering the redundancy brake circuit.

Control of rear axle braking is achieved via the speed sensors giving feed-back on each wheel retardation or acceleration to the axle modulator and with the calculated brake pressure needs derived by the electronic control module (ECM) delivers the appropriate brake pressure to the wheel spring brake actuators.

Rear a les — foot brake applied under ABS/ CS conditions (Fig 12.42(c)) If the brakes are applied

and the feed-back from one of the rear axle speed sensors registers excessive wheel lock/spin, then the electronic control module (ECM) will actuate the axle modulator (AM). Instantly the unstable breakaway wheel inlet/exhaust solenoid valves are energized, that is, the inlet solenoid valve closes and the corresponding exit valve opens, therefore blocking the air pressure passage leading to the wheel brake actuator and causing the air pressure in the actuator diaphragm chamber to collapse. The energizing and de-energizing (opening and closing) of each pair of solenoid valves is repeated continuously to adjust the magnitude of the wheel braking as demanded by the driver without the individual wheel locking when braking or for one of the wheels to spin due to poor road grip when the vehicle is accelerating.

During ABS/TCS braking conditions the redundancy (pneumatic circuit) brake system must not become active; to achieve this, the 2/2 solenoid valve is energized and closes so that air pressure is maintained underneath the 3/2-way valve piston. Accordingly the space above the relay valve remains open to the atmosphere and the 3/2-way valve inlet remains shut, hence locking out the redundancy brake circuit. Braking will revert to normal electronic-pneumatic control when the difference in wheel speeds is relatively small.

Rear a le — foot brake applied with a fault in the electronic-pneumatics (Fig. 12.42(d)) Should the electronic-pneumatic brake circuit fail, the axle modulator (AM) solenoid valve de-energizes so that the solenoid inlet valves close, whereas the exit valves open. Consequently compressed air under the 3/2-way valve piston escapes from the left hand solenoid exit valve thereby permitting the 3/2-way valve inlet valve to open. Foot control valve modulated brake pressure now enters the relay valve's upper piston chamber where it controls the delivery of redundancy circuit pneumatic pressure to both wheel brake spring actuators via the 2-way valves which are now positioned to block compressed air reaching the axle modulator solenoid valves. Note with the redundancy circuit operating there will be no active ABS at the front and rear axles.

12.5.4 Trailer axle braking (Fig. 12.43(a–d))

railer a le — foot brake released (Fig. 12.43(a)) When the brake pedal is released the foot travel

sensor signals the electronic control module (ECM) to release the brakes by de-energizing the proportional valve (PV). As a result the proportional valve (PV) inlet closes and its exit opens so that air pressure in the throttle valve and relay valve piston chamber is exhausted. Accordingly the relay valve inlet closes and its exit opens, thus permitting the brake pressure leading to the coupling head to collapse and for the brakes to be released.

Trailer axles — foot brake applied (Fig. 12.43(b)) Maximum supply pressure from the rear axle and trailer reservoirs is routed to both the brake value sensor (BVS) and to the proportional relay valve (PRV) respectively.

When the driver operates the foot brake pedal, the travel sensors located inside the brake value sensor (BVS) unit measure the pedal downward movement and feed this information to the central electronic control module (ECM). At the same time the brake switch closes thereby instructing the electronic control module to switch on the stop light.

The electronic control module (ECM) responds by directing a calculated variable control current to the proportional valve (PV) which forms part of the EPB-trailer control valve unit. Energizing the proportional valve's solenoid, closes the exit valve and opens the inlet valve in proportion to the amount of braking requested. Controlled air pressure now enters the relay valve piston upper chamber; this closes the exit valve and opens the inlet valve in proportion to the degree of braking demanded. Modulated brake pressure will now pass to the coupling head brake circuit where it is then relayed to the trailer wheel brake actuators via the trailer-mounted relay emergency valve.

Trailer axle — parking brake applied (Fig. 12.43(c)) With the 'park' hand control valve in the 'off' position the hand control valve central plunger closes the exit and pushes open the inlet valve. Compressed air from the parking reservoir tank is therefore able to flow to the relay valve part of the EPB trailer control valve via the open inlet valve inside the hand control valve.

When the 'park' control valve lever moves towards 'park' position the central plunger rises, causing the exit to open and the inlet valve to close. Air pressure therefore exhausts from above the lower control piston within the relay valve. Supply pressure acting beneath the reaction piston will now be able to lift the reaction piston and inner valve assembly until the upper control piston plunger closes the exit. Further upward movement then opens the inlet valve, thus permitting supply air pressure to flow through the partially open inlet valve to the 'coupling head for brake', and hence to the trailer attached relay emergency valve where it modulates the supply pressure reaching the wheel brake actuators.

Trailer axle — foot brake applied with a fault in the electronic-pneumatics (Fig. 12.43(d)) If there is a fault in the electronic-pneumatic system the proportional valve (PV) is de-energized, causing its inlet valve to close and its exit to open; pressurized air is therefore able to exhaust from the relay valve's upper control piston chamber via the proportional valve exit. Note the relay valve consists of an upper control piston and a lower assembly with upper and lower piston regions and which incorporates an internal double seat inlet and exit valve. As a result the upper control piston moves to its uppermost position, the inlet valve initially closes and the trapped supply pressure and the redundancy brake pressure (foot control valve pneumatic pressure) acting underneath the inner assembly's upper and lower piston region, pushes up the assembly against the hand control valve park pressure sufficiently to close the exit valve and to open the inlet valve. Brake pressure will hence be delivered to the trailer wheel brake actuators via the 'coupling head for brakes'.

Brake line to trailer defective (Fig. 12.43(b)) If the brake line to the trailer fractures the output pressure from the relay valve drops, causing the pressure above the throttle valve piston to collapse; this forces the throttle valve piston to rise and partially close the throttle valve, thereby causing a rapid reduction in the supply pressure flowing to the coupling head supply line (diagram not shown with throttle valve in defective pipe line position). As a result the relay emergency valve mounted on the trailer switches into braking mode and hence overrides the electronic-pneumatic circuit brake control to bring the vehicle to rest.

13 Vehicle refrigeration

Refrigeration transport is much in demand to move frozen or chilled food from storage centres to shops and supermarkets. Thermally insulated body containers used for frozen and chilled food deliveries for both small rigid trucks and large articulated vehicles are shown in Figs 13.1 and 13.2 respectively. Refrigeration systems designed for motor vehicle trucks are basically made up of two parts supported on an aluminium alloy or steel frame. The condenser unit which is mounted outside the thermally insulated cold storage compartment, comprises a diesel engine (and optional electric motor) compressor, condenser coil, fan thermostat and accessories. The evaporator unit protruding inside the cold storage body contains the evaporator coil, evaporator fan expansion valve remote feeler bulb and any other accessories. For certain applications a standby electric motor is incorporated to drive the compressor when the truck is being parked at the loading or delivery site for a long period of time such as overnight, the electricity supply being provided by the local premises' mains power. Typical self-contained refrigeration unit arrangements incorporating an engine, compressor, evaporator, condenser, fans and any other accessories for small to medium and large frozen storage compartments are shown in Figs 13.3 and 13.4 respectively. Temperature control is fully automatic on a start–stop cycle. With small and medium size refrigeration systems the engine runs at full governed speed until the thermostat temperature setting is reached. It then automatically reduces speed and disconnects the magnetic or centrifugal clutch which stops the compressor. A slight increase of temperature will return the engine to full speed and again driving the compressor.

The cold storage compartment temperature for frozen food is usually set between $-22°C$ and $-25°C$ whereas the chilled compartment temperature is set between $+3°C$ and $+5°C$.

13.1 Refrigeration terms (Fig. 13.5)

To understand the operating principles of a refrigeration system it is essential to appreciate the following terms:

Refrigerant This is the working fluid that circulates though a refrigeration system and produces both cooling and heating as it changes state. The desirable properties of a refrigerant fluid are such that it flows through the evaporator in its vapour state, absorbs heat from its surroundings, then transfers this heat via the flow of the refrigerant
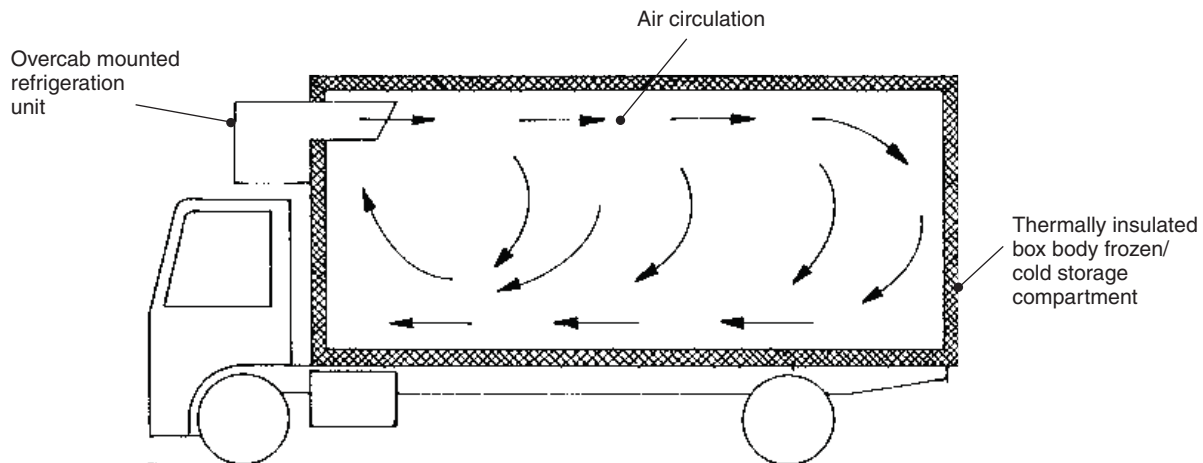
Fig. 13.1 Overcab mounted self-contained refrigeration system for small and medium rigid trucks

570
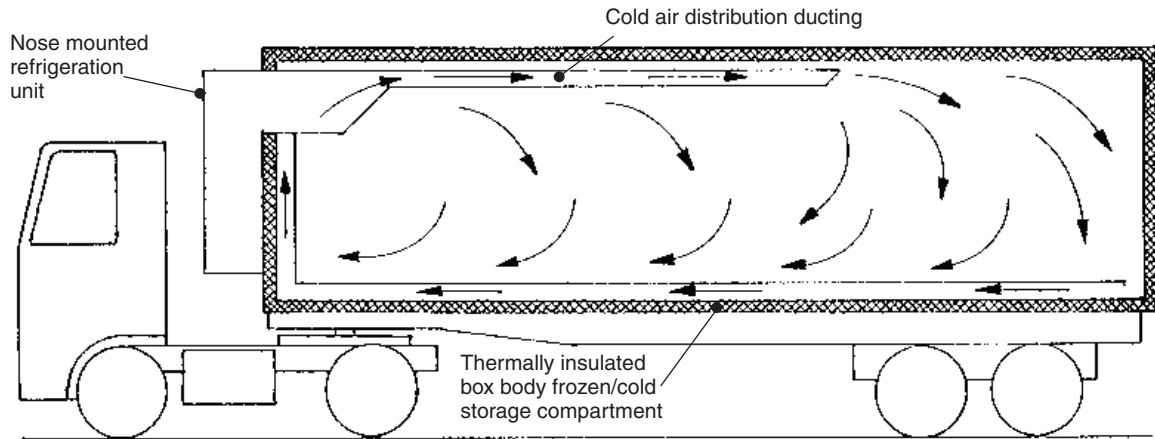
Fig. 13.2 Nose mounted self-contained refrigeration system for large articulated truck

to the condenser; the refrigerant then condenses to a liquid state and in the process dissipates heat taken in by the evaporator to the surrounding atmosphere. Refrigerants are normally in a vapour state at atmospheric pressure and at room temperature because they boil at temperatures below zero on the celsius scale; however, under pressure the refrigerant will convert to a liquid state.

Subcooled liquid (Fig. 13.5)   This is a liquid at any temperature below its saturated (boiling) temperature.

Saturated temperature (Fig. 13.5)   This is the temperature at which a liquid converts into vapour or a vapour converts into liquid, that is, the boiling point temperature.

Saturated liquid (Fig. 13.5)   This is a liquid heated to its boiling point, that is, it is at the beginning of vaporization.

Saturated vapour (Fig. 13.5)   This is the vapour which is formed above the surface of a liquid when heated to its boiling point.
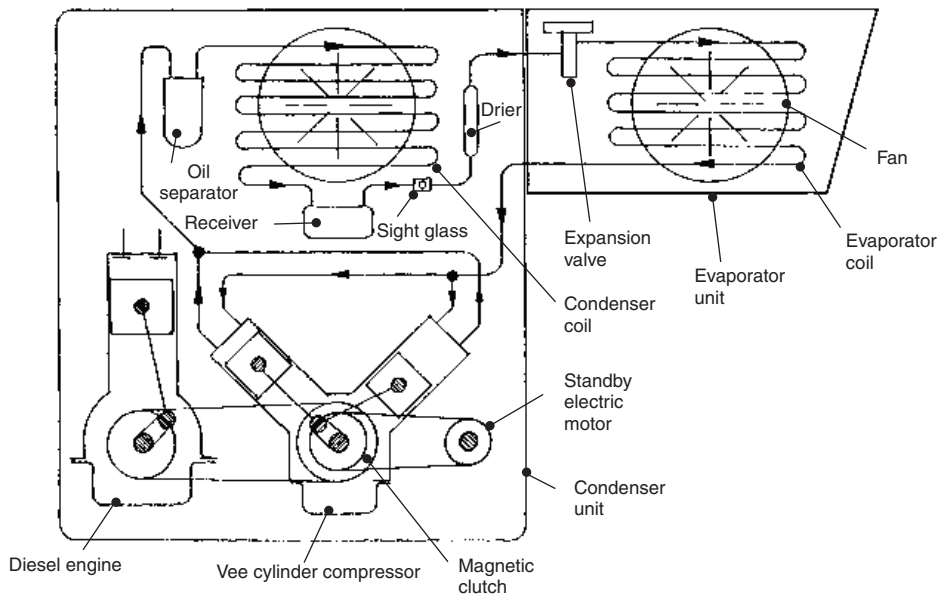
Fig. 13.3 Light to medium duty diesel engine and standby electric motor belt driven compressor refrigeration unit

571
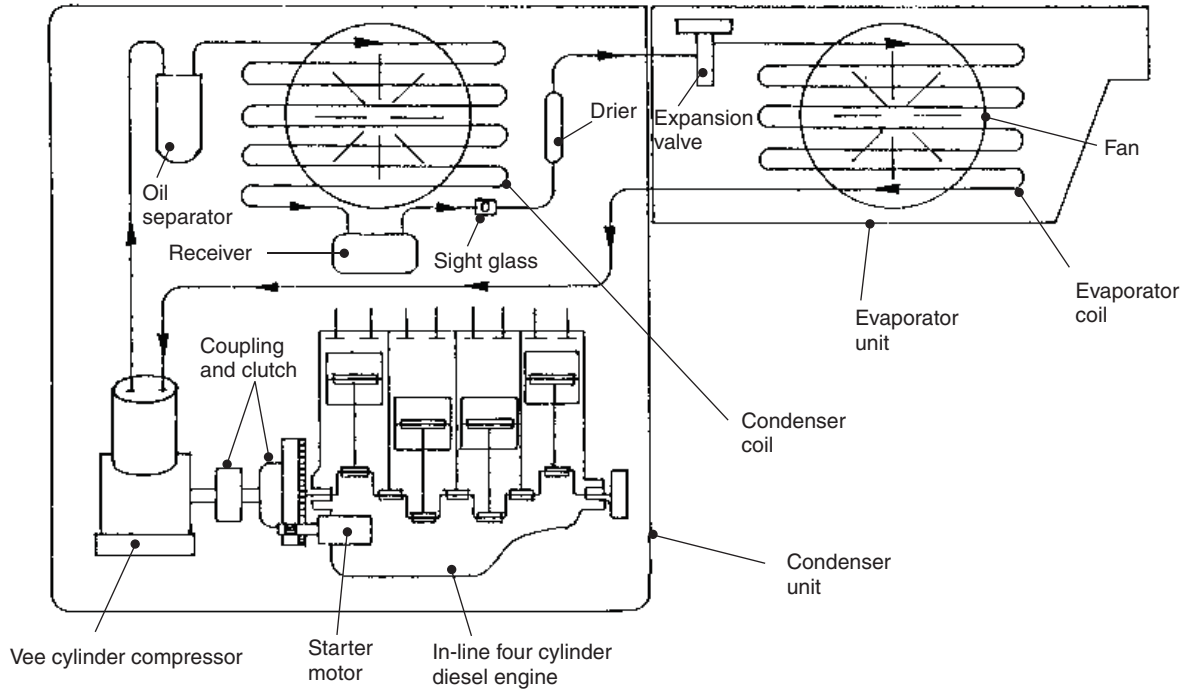
Fig. 13.4 Heavy duty diesel engine shaft driven compressor refrigeration unit
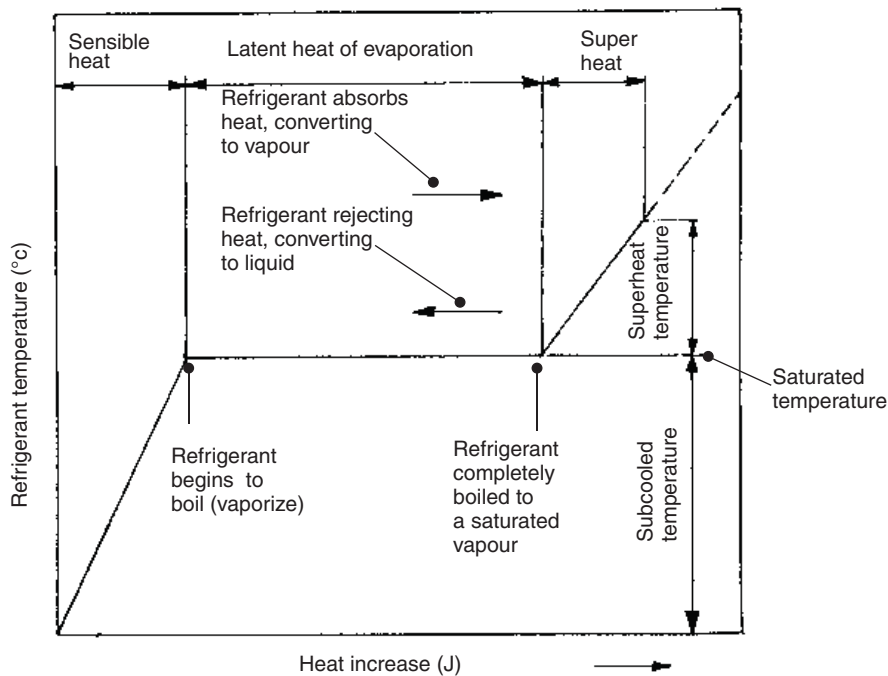
Fig. 13.5 Illustrative relationship between the refrigerant's temperature and heat content during a change of state

Latent heat of evaporation (Fig. 13.5) This is the heat needed to completely convert a liquid to a vapour and takes place without any temperature rise.

Superheated vapour (Fig. 13.5) This is a vapour heated to a temperature above the saturated temperature (boiling point); superheating can only occur once the liquid has been completely vaporized.

13.2 Principles of a vapour–compression cycle refrigeration system (Fig. 13.6)

1 High pressure subcooled liquid refrigerant at a typical temperature and pressure of 30 °C and 10 bar respectively flows from the receiver to the expansion valve via the sight glass and drier. The refrigerant then rapidly expands and reduces its pressure as it passes out from the valve restriction and in the process converts the liquid into a vapour flow.

2 The refrigerant now passes into the evaporator as a mixture of liquid and vapour, its temperature being lowered to something like −10 °C with a corresponding pressure of 2 bar (under these conditions the refrigerant will boil in the evaporator). The heat (latent heat of evaporation) necessary to cause this change of state will come from the surrounding frozen compartment in which the evaporator is exposed.
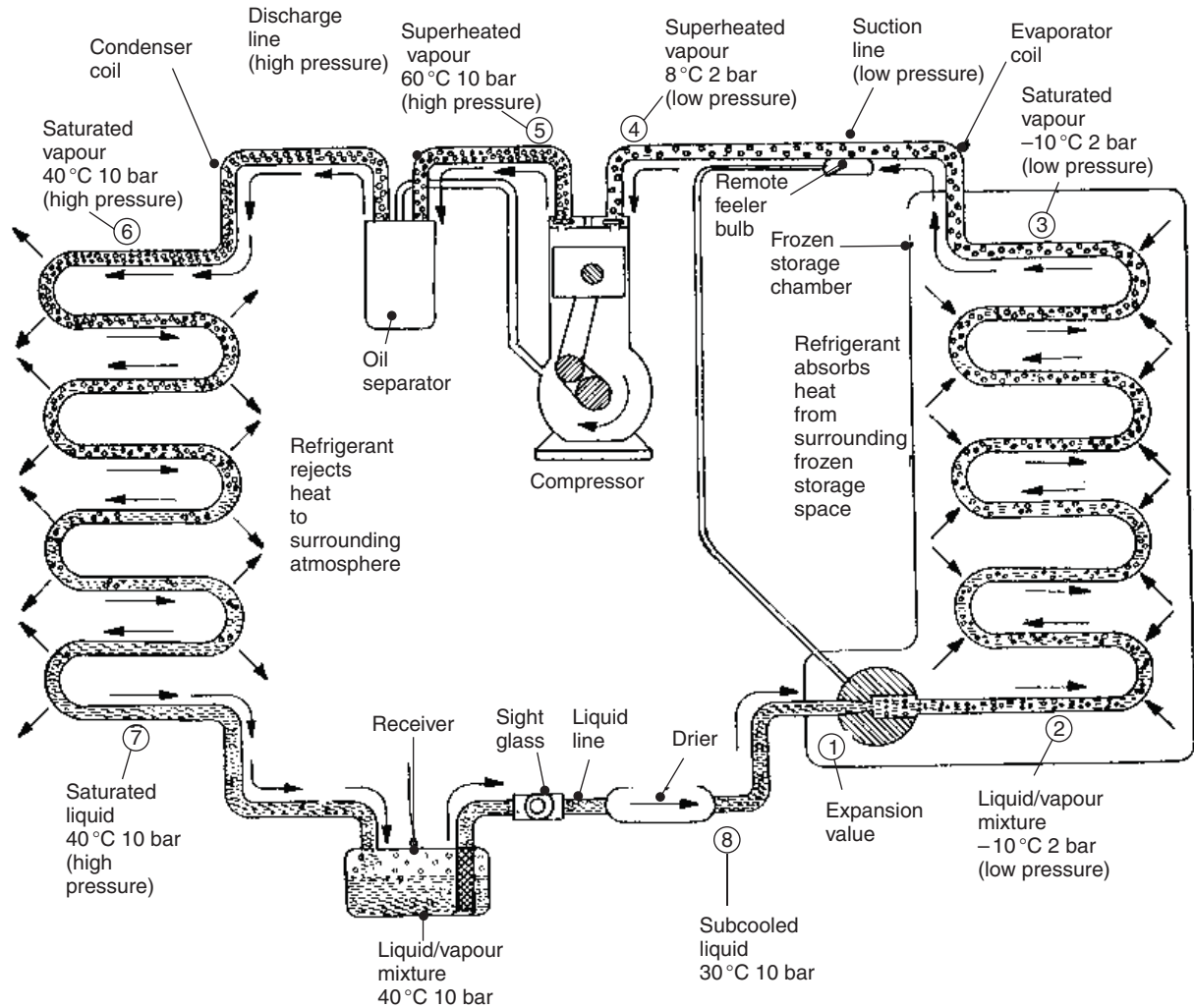
Fig. 13.6 Refrigeration vapour–compression cycle

573

3 As the refrigerant moves through the evaporator coil it absorbs heat and thus cools the space surrounding the coil. Heat will be extracted from the cold storage compartment until its pre-set working temperature is reached, at this point the compressor switches off. With further heat loss through the storage container insulation leakage, doors opening and closing and additional food products being stored, the compressor will automatically be activated to restore the desired degree of cooling. The refrigerant entering the evaporator tube completes the evaporation process as it travels through the evaporator coil so that the exit refrigerant from the evaporator will be in a saturated vapour state but still at the same temperature and pressure as at entry, that is, $-10\,°C$ and 2 bar respectively.

4 The refrigerant is now drawn towards the compressor via the suction line and this causes the heat from the surrounding air to superheat the refrigerant thus raising its temperature to something like $8\,°C$; however, there is no change in the refrigerant's pressure.

5 Once in the compressor the superheated vapour is rapidly compressed, consequently the superheated vapour discharge from the compressor is at a higher temperature and pressure in the order of $60\,°C$ and 10 bar respectively.

6 Due to its high temperature at the exit from the compressor the refrigerant quickly loses heat to the surrounding air as it moves via the discharge line towards the condenser. Thus at the entry to the condenser the refrigerant will be in a saturated vapour state with its temperature now lowered to about $40\,°C$; however, there is no further change in pressure which is still therefore 10 bar.

7 On its way through the condenser the refrigerant saturated vapour condenses to a saturated liquid due to the stored latent heat in the refrigerant transferring to the surrounding atmosphere via the condenser coil metal walls. Note the heat dissipated to the surrounding atmosphere by the condenser coil is equal to the heat taken in by the evaporator coil from the cold storage compartment and the compressor.

8 After passing through the condenser where heat is given up to the surrounding atmosphere the saturated liquid refrigerant now flows into the receiver. Here the increased space permits a small amount of evaporation to occur, the refrigerant then completes the circuit to the expansion valve though the liquid line where again heat is lost to the atmosphere, and this brings the refrigerant's temperature down to something like $30\,°C$ but without changing pressure which still remains at 10 bar.

13.3 Refrigeration system components

A description and function of the various components incorporated in a refrigeration system will be explained as follows:

13.3.1 Reciprocating compressor cycle of operation (Fig. 13.7(a–d))

Circulation of the refrigerant between the evaporator and the condenser is achieved by the pumping action of the compressor. The compressor draws in low pressure superheated refrigerant vapour from the evaporator and discharges it as high pressure superheated vapour to the condenser. After flowing through the condenser coil the high pressure refrigerant is now in a saturated liquid state; it then flows to the expansion valve losing heat on the way and thus causing the liquid to become subcooled. Finally the refrigerant expands on its way through the expansion valve causing it to convert into a liquid-vapour mix before re-entering the evaporator coil.

The reciprocating compressor completes a suction and discharge cycle every revolution; the outward moving piston from TDC to BDC forms the suction-stroke whereas the inward moving piston from BDC to TDC becomes the discharge stroke.

Suction stroke (Fig. 13.7(a and b)) As the crank shaft rotates past the TDC position the piston commences its suction stroke with the discharge reed valve closed and the suction reed valve open (Fig. 13.7(a and b)). The downward sweeping piston now reduces the cylinder pressure from P_1 to P_2 as its volume expands from V_1 to V_2, the vapour refrigerant in the suction line is now induced to enter the cylinder. The cylinder continues to expand and to be filled with vapour refrigerant at a constant pressure P_1 to the cylinder's largest volume of V_3, that is the piston's outermost position BDC, see Fig. 13.8.

ischarge stroke (Fig. 13.9(c and d)) As the crankshaft turns beyond BDC the piston begins its upward discharge stroke, the suction valve closes and the discharge valve opens (see Fig. 13.7(c and d)). The upward moving piston now compresses the refrigerant vapour thereby increasing the cylinder pressure from P_1 to P_2 through a volume reduction from V_3 to V_4 at which point the cylinder pressure
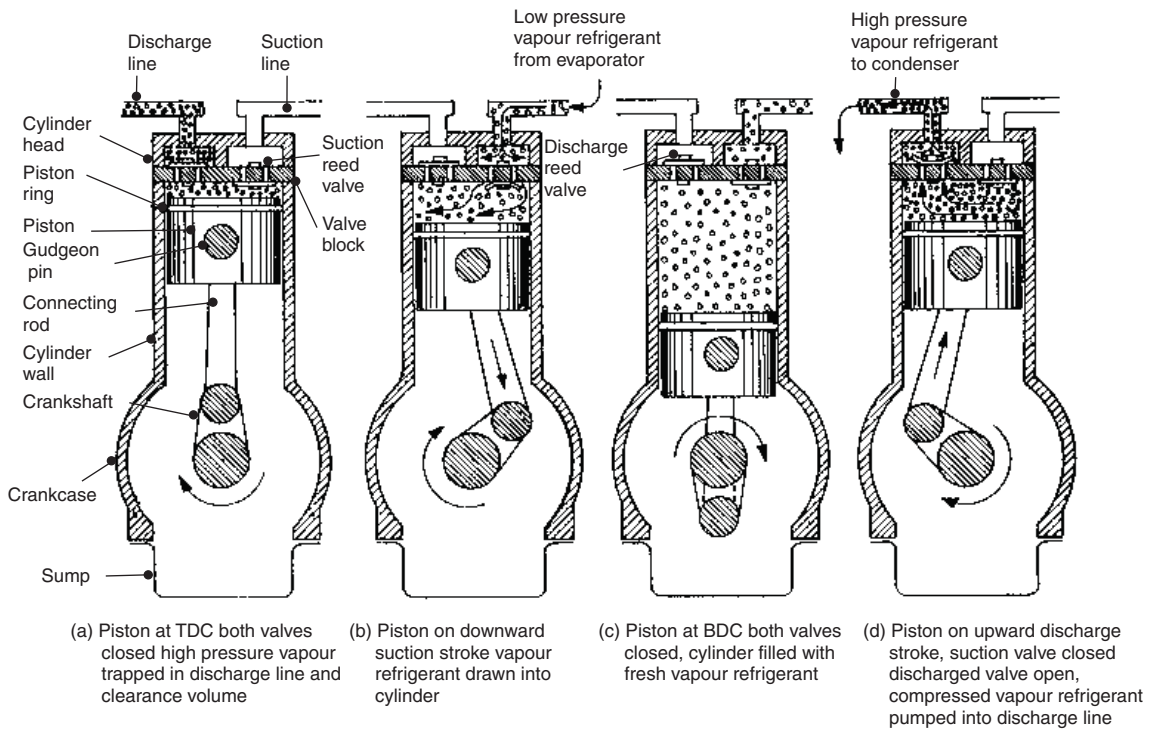
Fig. 13.7 (a–d)   Reciprocating compressor cycle of operation

(a) Piston at TDC both valves closed high pressure vapour trapped in discharge line and clearance volume

(b) Piston on downward suction stroke vapour refrigerant drawn into cylinder

(c) Piston at BDC both valves closed, cylinder filled with fresh vapour refrigerant

(d) Piston on upward discharge stroke, suction valve closed discharged valve open, compressed vapour refrigerant pumped into discharge line

Fig. 13.8   Reciprocating compressor pressure-volume cycle
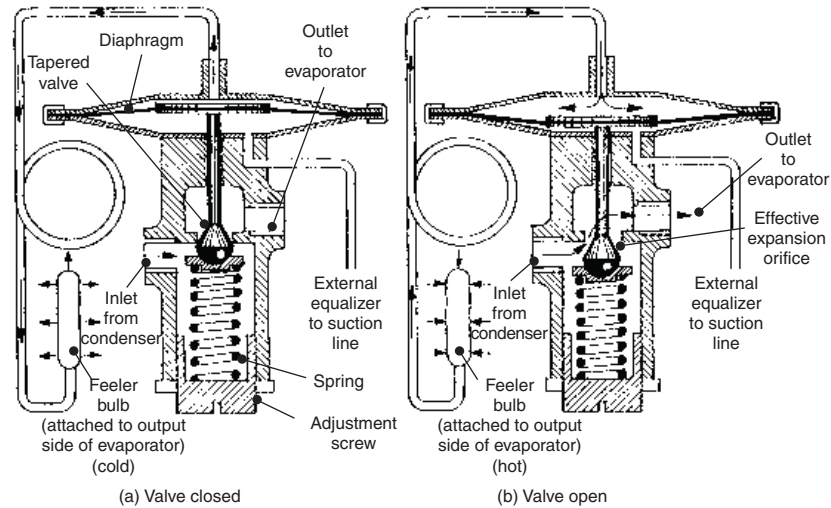
Fig. 13.9 (a and b) Thermostatic expansion valve

equals the discharge line pressure; the final cylinder volume reduction therefore from V_4 back to V_1 will be displaced into the high pressure discharge line at the constant discharge pressure of P_2 (see Fig. 13.8).

13.3.2 Evaporator (Fig. 13.6)

The evaporator's function is to transfer heat from the food being stored in the cold compartment into the circulating refrigerant vapour via the fins and metal walls of the evaporator coil tubing by convection and conduction respectively. The refrigerant entering the evaporator is nearly all liquid but as it moves through the tube coil, it quickly reaches its saturation temperature and is converted steadily into vapour. The heat necessary for this change of state comes via the latent heat of evaporation from the surrounding cold chamber atmosphere.

The evaporator consists of copper, steel or stainless steel tubing which for convenience is shaped in an almost zigzag fashion so that there are many parallel lengths bent round at their ends thus enabling the refrigerant to flow from side to side. To increase the heat transfer capacity copper fins are attached to the tubing so that relatively large quantities of heat surrounding the evaporator coil can be absorbed through the metal walls of the tubing, see Fig. 13.15(a and b).

13.3.3 Condenser (Fig. 13.6)

The condenser takes in saturated refrigerant vapour after it has passed though the evaporator and compressor, progressively cooling then takes place as it travels though the condenser coil,

accordingly the refrigerant condenses and reverts to a liquid state. Heat will be rejected from the refrigerant during this phase change via conduction though the metal walls of the tubing and convection to the surrounding atmosphere.

A condenser consists of a single tube shaped so that there are many parallel lengths with semicircular ends which therefore form a continuous winding or coil. Evenly spaced cooling fins are normally fixed to the tubing, this greatly increases the surface area of the tubing exposed to the convection currents of the surrounding atmosphere, see Fig. 13.15(a and b).

Fans either belt driven or directly driven by an electric motor are used to increase the amount of air circulation around the condenser coil, this therefore improves the heat transfer taking place between the metal tube walls and fins to the surrounding atmosphere. This process is known as forced air convection.

13.3.4 Thermostatic expansion valve
(Fig. 13.9(a and b))

An expansion valve is basically a small orifice which throttles the flow of liquid refrigerant being pumped from the condenser to the evaporator; the immediate exit from the orifice restriction will then be in the form of a rapidly expanding refrigerant, that is, the refrigerant coming out from the orifice is now a low pressure continuous liquid-vapour stream. The purpose of the thermostatic valve is to control the rate at which the refrigerant passes from the liquid line into the evaporator and

to keep the pressure difference between the high and low pressure sides of the refrigeration system.

The thermostatic expansion valve consists of a diaphragm operated valve (see Fig. 13.9(a and b)). One side of the diaphragm is attached to a spring loaded tapered/ball valve, whereas the other side of the diaphragm is exposed to a refrigerant which also occupies the internal space of the remote feeler bulb which is itself attached to the suction line tube walls on the output side of the evaporator. If the suction line saturated/superheated temperature decreases, the pressure in the attached remote feeler bulb and in the outer diaphragm chamber also decreases. Accordingly the valve control spring thrust will partially close the taper/ball valve (see Fig. 13.9(a)). Consequently the reduced flow of refrigerant will easily now be superheated as it leaves the output from the evaporator. In contrast if the superheated temperature rises, the remote feeler bulb and outer diaphragm chamber pressure also increases, this therefore will push the valve further open so that a larger amount of refrigerant flows into the evaporator, see Fig. 13.9(b). The extra quantity of refrigerant in the evaporator means that less superheating takes place at the output from the evaporator. This cycle of events is a continuous process in which the constant superheated temperature control in the suction line maintains the desired refrigerant supply to the evaporator.

A simple type of thermostatic expansion valve assumes the input and output of an evaporator are both working at the same pressure; however, due to internal friction losses the output pressure will be slightly less than the input. Consequently the lower output pressure means a lower output saturated temperature so that the refrigerant will tend to vaporize completely before it reaches the end of the coil tubing. As a result this portion of tubing converted completely into vapour and which is in a state of superheat does not contribute to the heat extraction from the surrounding cold chamber so that the effective length of the evaporator coil is reduced. To overcome early vaporization and superheating, the diaphragm chamber on the valve-stem side is subjected to the output side of the evaporator down stream of the remote feeler bulb. This extra thrust opposing the remote feeler bulb pressure acting on the outer diaphragm chamber now requires a higher remote feeler bulb pressure to open the expansion valve.

13.3.5 Suction pressure valve (throttling valve)
(Fig. 13.10(a and b))
This valve is incorporated in the compressor output suction line to limit the maximum suction
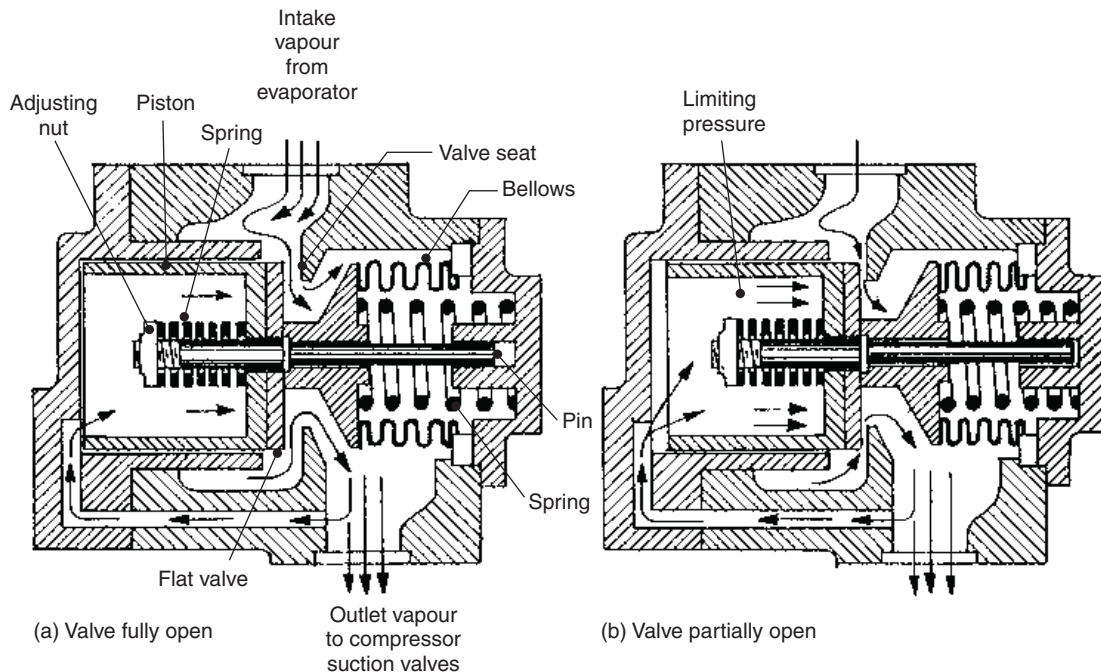
Fig. 13.10   Suction pressure regulating valve (throttling valve)

pressure generated by the compressor thereby safe-guarding the compressor and drive engine/motor from overload. If the maximum suction pressure is exceeded when the refrigeration system is switched on and started up (pull down) excessive amounts of vapour or vapour/liquid or liquid refrigerant may enter the compressor cylinder, which could produce very high cylinder pressures; this would therefore cause severe strain and damage to the engine/electric motor components, conversely if the suction line pressure limit is set very low the evaporator may not operate efficiently.

The suction pressure valve consists of a com-bined piston and bellows controlled valve subjected to suction vapour pressure.

When the compressor is being driven by the engine/motor the output refrigerant vapour from the evaporator passes to the intake port of the suction pressure valve unit; this exposes the bellows to the refrigerant vapour pressure and temperature. Thus as the refrigerant pressure rises the bellows will contract against the force of the bellows spring; this restricts the flow of refrigerant to the compres-sor (see Fig. 13.10(a)). However, as the bellows temperature rises its internal pressure also increases and will therefore tend to oppose the contraction of

the bellows. At the same time the piston will be subjected to the outlet vapour pressure from the suction pressure valve before entering the compres-sor cylinders, see Fig. 13.10(b). If this becomes excessive the piston and valve will move towards the closure position thus restricting the entry of refrigerant vapour or vapour/liquid to the com-pressor cylinders. Hence it can be seen that the suction pressure valve protects the compressor and drive against abnormally high suction line pressure.

13.3.6 Reverse cycle valve (Fig. 13.11(a and b))
The purpose of this valve is to direct the refrigerant flow so that the refrigerant system is in either a cooling or heating cycle mode.

Refrigerant cycle mode (Fig. 13.11(a)) With the pilot solenoid valve de-energized the suction pas-sage to the slave cylinder of the reverse cycle valve is cut off whereas the discharge pressure supply from the compressor is directed to the slave pis-ton. Accordingly the pressure build-up pushes the piston and both valve stems inwards; the left hand compressor discharge valve now closes the
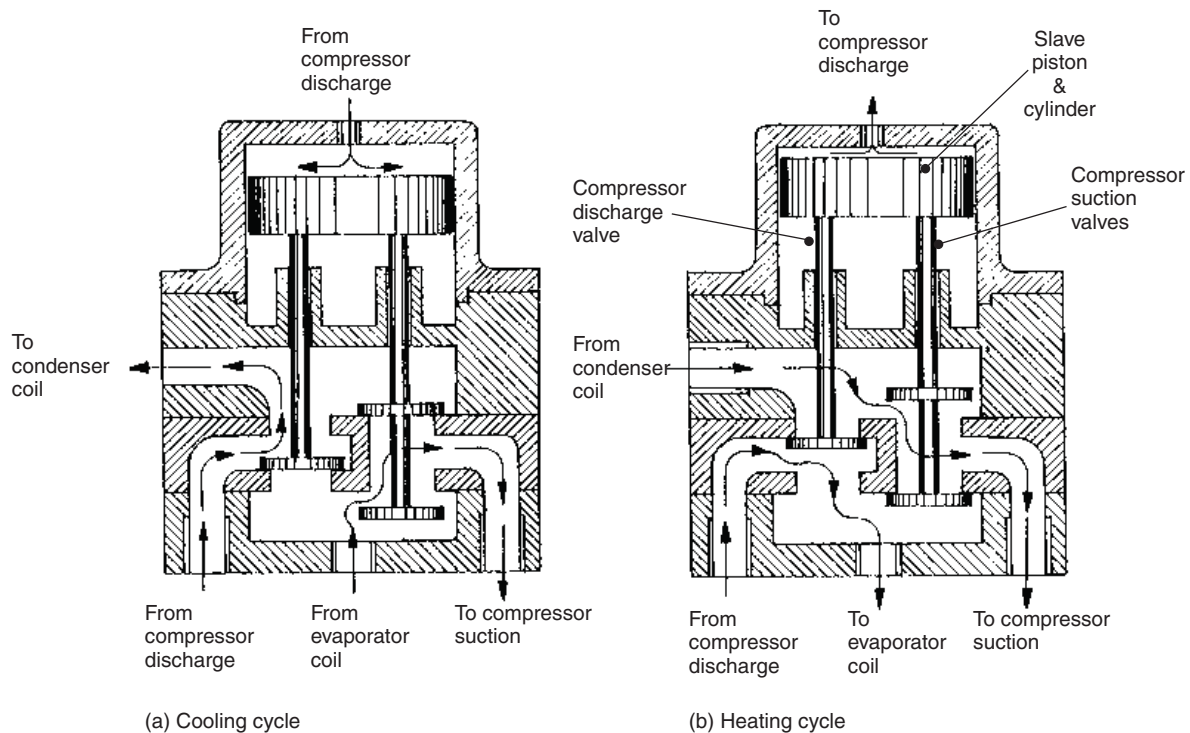
Fig. 13.11 (a and b)   Reverse cycle valve

compressor discharge passage to the evaporator and opens the compressor discharge passage to the condenser whereas the right hand double compressor discharge valve closes the condenser to compressor suction passage and opens the evaporator to the compressor suction pressure.

Heat/defrost cycle mode (Fig. 13.11(b)) Energizing the pilot solenoid valve cuts off the compressor discharge pressure to the slave cylinder of the reverse cycle valve and opens it to the compressor suction line. As a result the trapped refrigerant vapour in the slave cylinder escapes to the compressor suction line thus permitting the slave piston and both valves to move to their outermost position. The left hand compressor discharge valve now closes the compressor discharge to the condenser passage and opens the compressor discharge to the evaporator passage whereas the right hand compressor suction double valve closes the evaporator to the compressor suction passage and opens the condenser to compressor suction pressure.

13.3.7 *Drier* (Fig. 13.12)
Refrigerant circulating the refrigerator system must be dry, that is, the fluid, be it a vapour or a liquid, should not contain water. Water in the form of moisture can promote the formation of acid which can attack the tubing walls and joints and cause the refrigerant to leak out. It may initiate the formation of sludge and restrict the circulation of the refrigerant. Moisture may also cause ice to form in the thermostatic expansion valve which again could reduce the flow of refrigerant. To overcome problems with water contamination driers are normally incorporated in the liquid line; these liquid line driers not only remove water contained in the refrigerant, they also remove sludge and other impurities. Liquid line driers are plumbed in on the output side of the receiver, this is because the moisture is concentrated in a relatively small space when the refrigerant is in a liquid state.

A liquid line drier usually takes the form of a cylindrical cartridge with threaded end connections so that the drier can be replaced easily when necessary. Filter material is usually packed in at both ends; in the example shown Fig. 13.12 there are layers, a coarse filter, felt pad and a fine filter. In between the filter media is a desiccant material, these are generally of the adsorption desiccant kind such as silica gel (silicon dioxide) or activated alumna (aluminium oxide). The desiccant substance has microscopic holes for the liquid refrigerant to pass through; however, water is attracted to the desiccant and therefore is prevented from moving on whereas the dry (free of water) clean refrigerant will readily flow through to the expansion valve.

13.3.8 *Oil separator* (Fig. 13.13)
Oil separators are used to collect any oil entering the refrigeration system through the compressor and to return it to the compressor crankcase and sump. The refrigerant may mix with the compressor's lubrication oil in the following way:

1 During the cycle of suction and discharge refrigerant vapour periodically enters and is displaced from the cylinders; however, if the refrigerant flow becomes excessive liquid will pass through the expansion valve and may eventually enter the suction line via the evaporator. The fluid may then drain down the cylinder walls to the crankcase and sump. Refrigerant mixing with oil dilutes it so that it loses its lubricating properties: the wear and tear of the various rubbing components in the compressor will therefore increase.
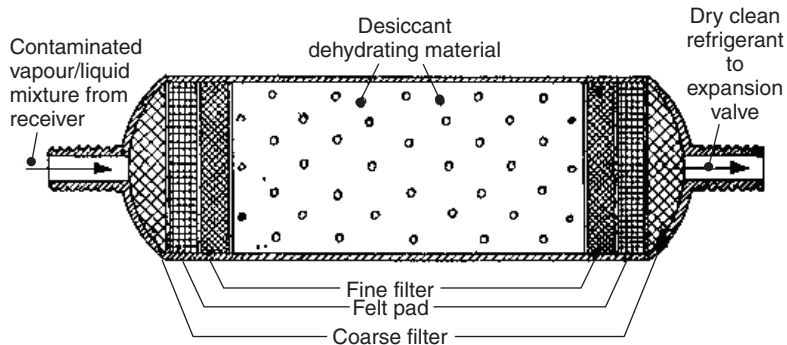
Fig. 13.12 Adsorption type liquid line drier
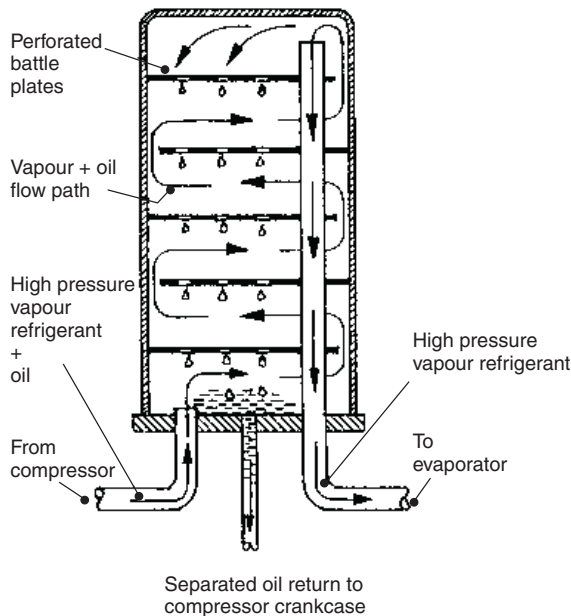
Fig. 13.13 Oil separator

2 When the refrigerator is switched off the now static refrigerant in the evaporator may condense and enter the suction line and hence the compressor cylinders where it drains over a period of time into the crankcase and sump.

3 Refrigerant mixing with the lubricant in the crankcase tends to produce oil frothing which finds its way past the pistons and piston rings into the cylinders; above each piston the oil will then be pumped out into the discharge line with the refrigerant where it then circulates. Oil does not cause a problem in the condenser as the temperature is fairly high so that the refrigerant remains suspended; however, in the evaporator the temperature is low so that the liquid oil separates from the refrigerant vapour, therefore tending to form a coating on the inside bore of the evaporator coil. Unfortunately oil is a very poor conductor of heat so that the efficiency of the heat transfer process in the evaporator is very much impaired.

After these observations it is clear that the refrigerant must be prevented from mixing with the oil but this is not always possible and therefore an oil separator is usually incorporated on the output side of the compressor in the discharge line which separates the liquid oil from the hot refrigerant vapour. An oil separator in canister form consists of a cylindrical chamber with a series of evenly spaced perforated baffle plates or wire mesh partitions attached to the container walls; each baffle plate has a small segment removed to permit the flow of refrigerant vapour (Fig. 13.13), the input from the compressor discharge being at the lowest point whereas the output is via the extended tube inside the container. A small bore pipe connects the base of the oil separator to the compressor crankcase to provide a return passage for the liquid oil accumulated. Thus when the refrigerator is operating, refrigerant will circulate and therefore passes though the oil separator. As the refrigerant/oil mix zigzags its way up the canister the heavier liquid oil tends to be attracted and attached to the baffle plates; the accumulating oil then spreads over the plates until it eventually drips down to the base of the canister, and then finally drains back to the compressor crankcase.

13.3.9 Receiver (Fig. 13.6)
The receiver is a container which collects the condensed liquid refrigerant and any remaining vapour from the condenser; this small amount of vapour will then have enough space to complete the condensation process before moving to the expansion valve.

13.3.10 Sight glass (Fig. 13.14)
This device is situated in the liquid line on the output side of the receiver; it is essentially a viewing port which enables the liquid refrigerant to be seen. Refrigerant movement or the lack of movement due to some kind of restriction, or bubbling caused by insufficient refrigerant, can be observed.

13.4 Vapour–compression cycle refrigeration system with reverse cycle defrosting
(Fig. 13.15(a and b))
A practical refrigeration system suitable for road transportation as used for rigid and articulated vehicles must have a means of both cooling and
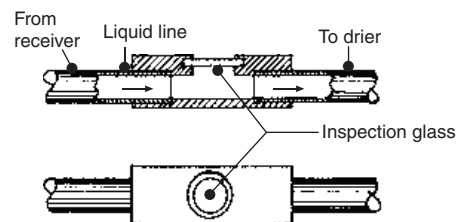
Fig. 13.14 Sight glass
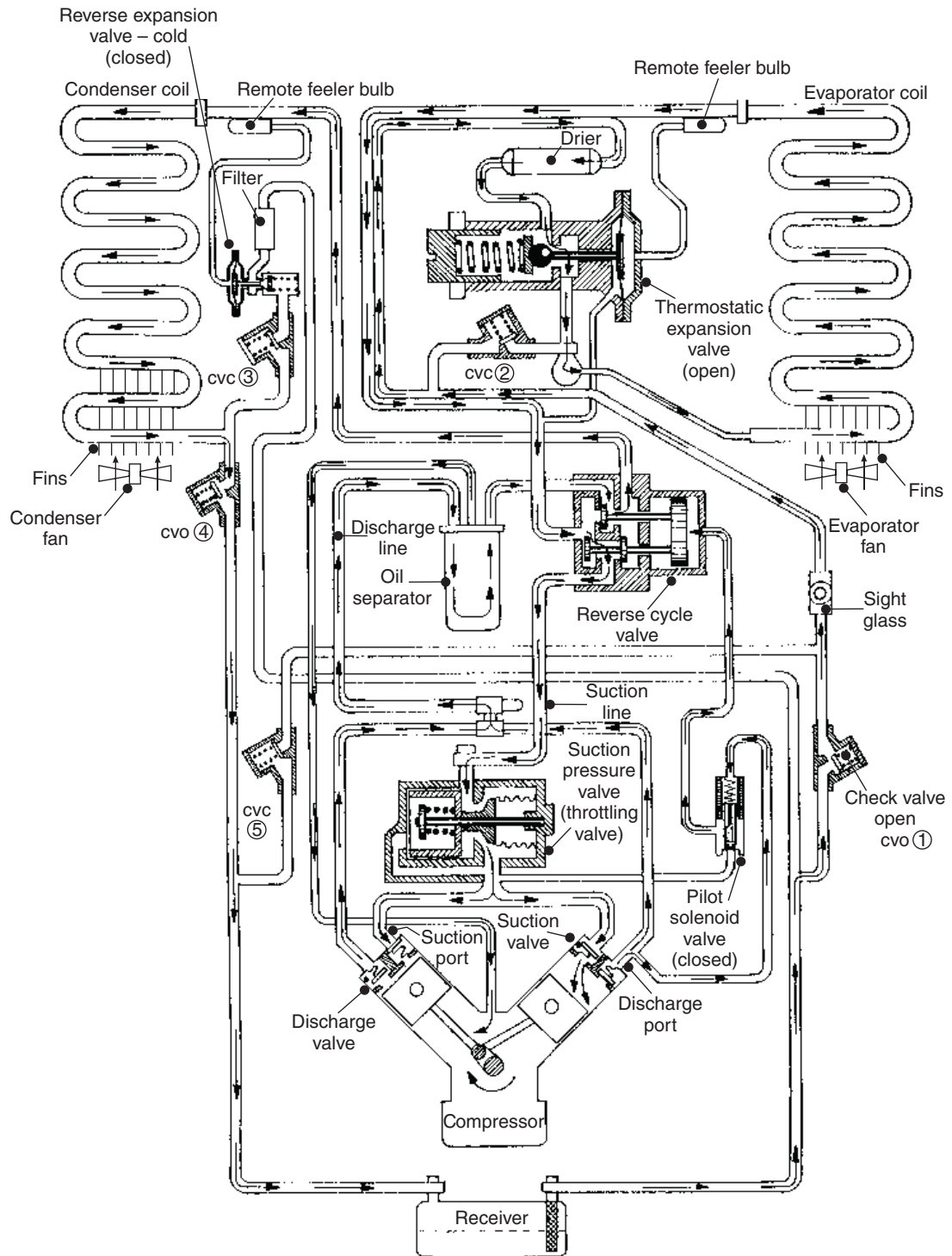
(a) Refrigeration cycle

Fig. 13.15 (a and b) Refrigeration system with reverse cycle defrosting

581
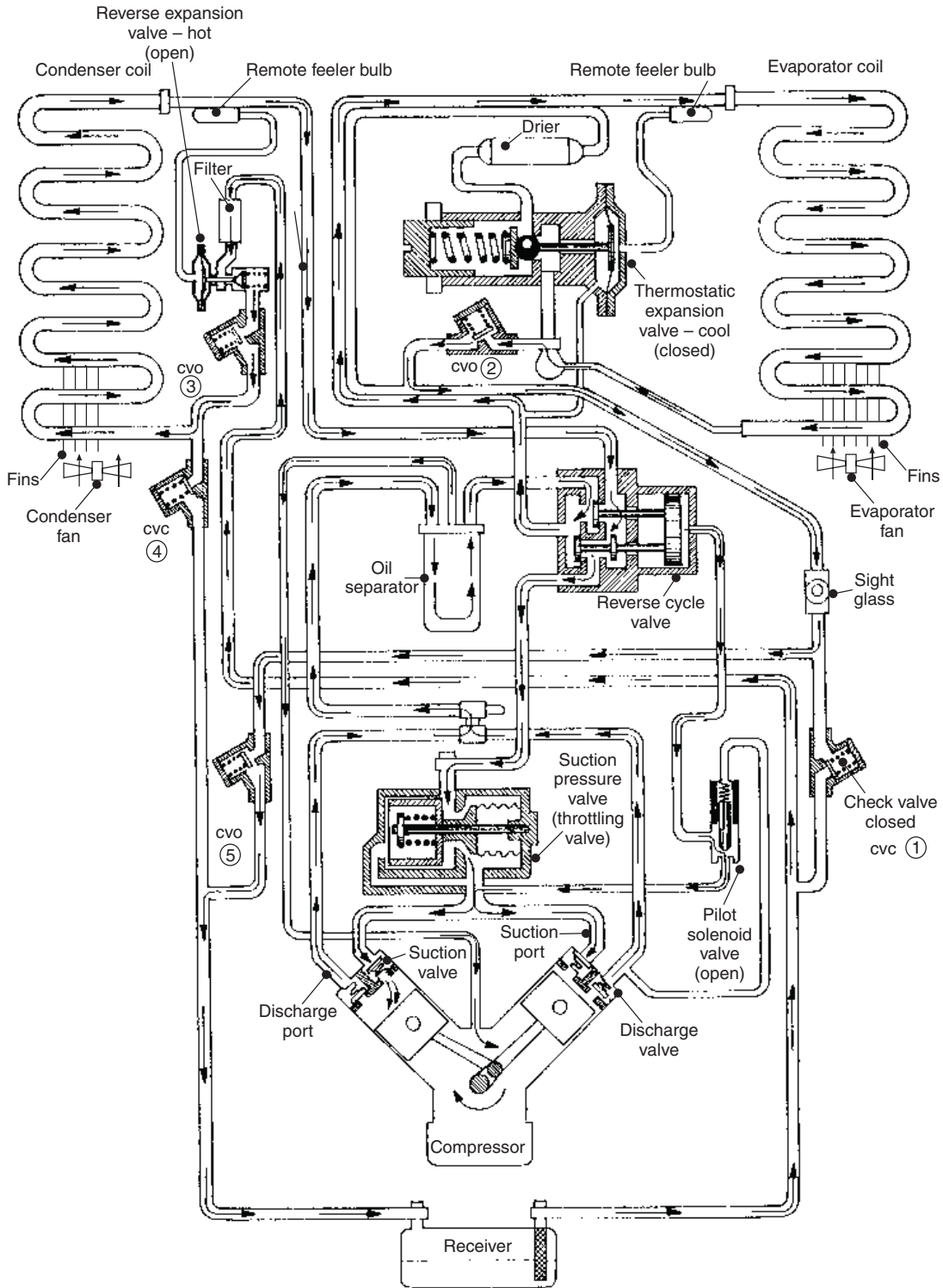
(b) Heating and defrost cycle

Fig. 13.15 *Contd*

582

defrosting the cold compartment. The operation of such a system involving additional valves enables the system to be switched between cooling and heat/defrosting, which will now be described.

13.4.1 Refrigeration cooling cycle (Fig. 13.15(a))

With the pilot solenoid valve de-energized and the compressor switched on and running the refrigerant commences to circulate through the system between the evaporator and condenser.

Discharge line pressure from the right hand compressor cylinder is transferred via the pilot valve to the reverse cycle valve; this pushes the slave piston and valves inwards to the left hand side into the 'cooling' position, see Fig. 13.15(a). Low pressure refrigerant from the receiver flows via the open check valve (1), sight glass and drier to the thermostatic expansion valve where rapid expansion in the valve converts the refrigerant to a liquid/vapour mixture. Low pressure refrigerant then passes through the evaporator coil where it absorbs heat from the cold storage compartment: the refrigerant then comes out from the evaporator as low pressure saturated vapour. Refrigerant now flows to the compressor suction port via the reverse cycle valve and suction pressure valve as superheated vapour. The compressor now converts the refrigerant to a high pressure superheated vapour before pumping it to the condenser inlet via the oil separator and reverse cycle valve; at this point the refrigerant will have lost heat to the surroundings so that it is now in a high pressure saturated vapour state. It now passes through the condenser where it gives out its heat to the surrounding atmosphere; during this process the high pressure refrigerant is transformed into a saturated liquid. Finally the main liquid refrigerant flows into the receiver via the open check valve (4) where there is enough space for the remaining vapour to condense. This cycle of events will be continuously repeated as the refrigerant is alternated between reducing pressure in the expansion valve before passing through the evaporator to take heat from the cold chamber, to increasing pressure in the compressor before passing through the condenser to give off its acquired heat to the surroundings. Note check valves (1) and (4) are open whereas check valves (2), (3) and (5) are closed for the cool cycle.

13.4.2 Heating and defrosting cycle
(Fig. 13.15(b))

With constant use excessive ice may build up around the evaporator coil; this restricts the air movement so that the refrigerant in the evaporator is unable to absorb the heat from the surrounding atmosphere in the cold storage compartment, therefore a time will come when the evaporator must be defrosted.

Heating/defrosting is achieved by temporarily reversing the refrigerant flow circulation so that the evaporator becomes a heat dissipating coil and the condenser converts to a heat absorbing coil.

To switch to the heat/defrosting cycle the pilot solenoid valve is energized; this causes the solenoid valve to block the discharge pressure and connect the suction pressure to the servo cylinder reverse cycle valve, see Fig. 13.15(b). Subcooled high pressure liquid refrigerant is permitted to flow from the receiver directly to the now partially opened reverse thermostatic expansion valve (due to the now hot remote feeler bulb's increased pressure). The refrigerant expands in the reverse expansion valve and accordingly converts to a liquid/vapour; this then passes through the condenser via the open check valve (3) in the reverse direction to the normal refrigeration cycle and in the process absorbs heat from the surroundings so that it comes out as a low pressure saturated vapour. The refrigerant then flows to the compressor suction port via the reverse cycle valve and suction pressure valve but due to the high surrounding atmospheric temperature it is now superheated vapour. The compressor then transforms the low pressure superheated vapour into a high pressure superheated vapour and discharges it to the evaporator via the oil separator and reverse cycle valve. Hence the saturated vapour stream dissipates its heat through the tubing walls to the ice which is encasing the tubing coil until it has all melted. The refrigerant at the exit from the evaporator will now be in a saturated liquid state and is returned to the receiver via the open check valve (2), sight glass, and open check valve (5) for the heating/defrosting cycle to be repeated. Note during the refrigeration cycle the condenser's reverse expansion valve and remote feeler bulb sense the reduction in temperature at the exit from the condenser, thus the corresponding reduction in internal bulb pressure is relayed to the reverse expansion valve which therefore closes during the defrosting cycle. Defrosting is fully automatic. A differential air pressure switch which senses any air circulation restriction around the evaporator coil automatically triggers defrosting of the evaporator coil before ice formation can reduce its efficiency. A manual defrost switch is also provided.

14 Vehicle body aerodynamics

The constant need for better fuel economy, greater vehicle performance, reduction in wind noise level and improved road holding and stability for a vehicle on the move, has prompted vehicle manufacturers to investigate the nature of air resistance or drag for different body shapes under various operating conditions. Aerodynamics is the study of a solid body moving through the atmosphere and the interaction which takes place between the body surfaces and the surrounding air with varying relative speeds and wind direction. Aerodynamic drag is usually insignificant at low vehicle speed but the magnitude of air resistance becomes considerable with rising speed. This can be seen in Fig. 14.1 which compares the aerodynamic drag forces of a poorly streamlined, and a very highly streamlined medium sized car against its constant rolling resistance over a typical speed range. A vehicle with a high drag resistance tends only marginally to hinder its acceleration but it does inhibit its maximum speed and increases the fuel consumption with increasing speed.

Body styling has to accommodate passengers and luggage space, the functional power train, steering, suspension and wheels etc. thus vehicle design will conflict with minimizing the body surface drag so that the body shape finally accepted is nearly always a compromise.

An appreciation of the fundamentals of aerodynamics and the methods used to counteract high air resistance for both cars and commercial vehicles will now be explained.

14.1 Viscous air flow fundamentals

14.1.1 Boundary layer (Fig. 14.2)

Air has viscosity, that is, there is internal friction between adjacent layers of air, whenever there is relative air movement, consequently when there is sliding between adjacent layers of air, energy is dissipated. When air flows over a solid surface a thin boundary layer is formed between the main airstream and the surface. Any relative movement between the main airstream flow and the surface of
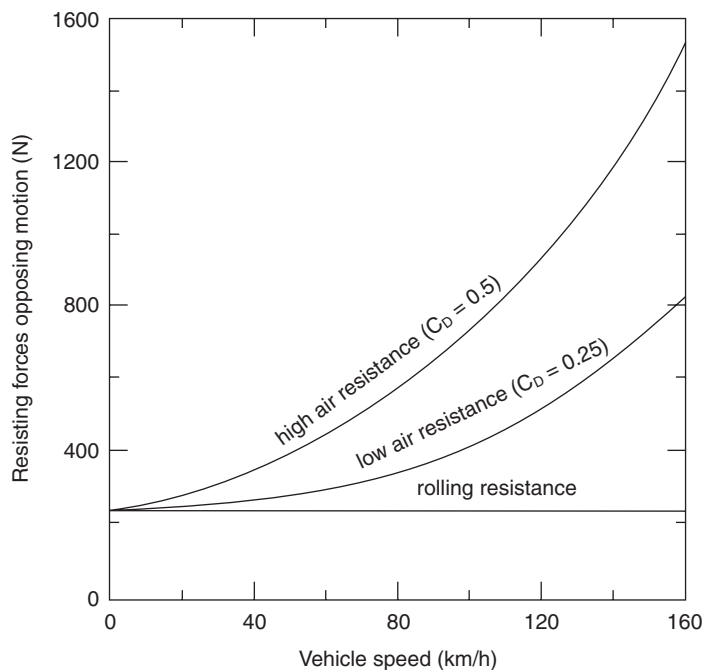
Fig. 14.1 Comparison of low and high aerodynamic drag forces with rolling resistance
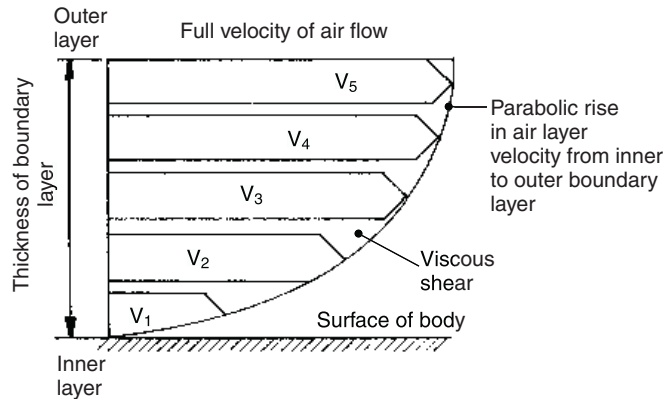
Fig. 14.2 Boundary layer velocity gradient
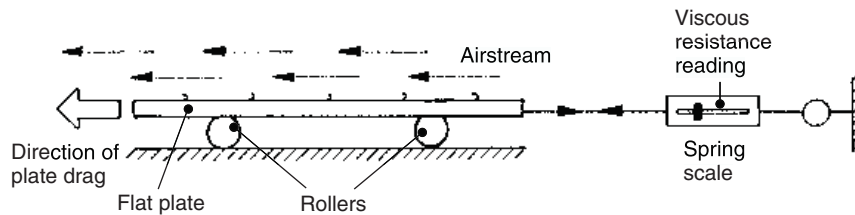
Fig. 14.3 Apparatus to demonstrate viscous drag

the body then takes place within this boundary layer via the process of shearing of adjacent layers of air. When air flows over any surface, air particles in intimate contact with the surface loosely attach themselves so that relative air velocity at the surface becomes zero, see Fig. 14.2. The relative speed of the air layers adjacent to the arrested air surface film will be very slow; however, the next adjacent layer will slide over an already moving layer so that its relative speed will be somewhat higher. Hence the relative air velocity further out from the surface rises progressively between air layers until it attains the unrestricted main airstream speed.

14.1.2 Skin friction (surface friction drag) (Fig. 14.3)
This is the restraining force preventing a thin flat plate placed edgewise to an oncoming airstream being dragged along with it (see Fig. 14.3), in other words, the skin friction is the viscous resistance generated within the boundary layer when air flows over a solid surface. Skin friction is dependent on the surface area over which the air flows, the degree of surface roughness or smoothness and the air speed.

14.1.3 Surface finish (Fig. 14.4(a and b))
Air particles in contact with a surface tend to be attracted to it, thus viscous drag will retard the

layer of air moving near the surface. However, there will be a gradual increase in air speed from the inner to the outer boundary layer. The thickness of the boundary layer is influenced by the surface finish. A smooth surface, see Fig. 14.4(b), allows the free air flow velocity to be reached nearer the surface whereas a rough surface, see Fig. 14.4(a), widens the boundary so that the full air velocity will be reached further out from the surface. Hence the thicker boundary layer associated with a rough surface will cause more adjacent layers of air to be sheared, accordingly there will be more resistance to air movement compared with a smooth surface.

14.1.4 Venturi (Fig. 14.5)
When air flows through a diverging and converging section of a venturi the air pressure and its speed changes, see Fig. 14.5. Initially at entry the unrestricted air will be under atmospheric conditions where the molecules are relatively close together, consequently its pressure will be at its highest and its speed at its minimum.

As the air moves into the converging section the air molecules accelerate to maintain the volume flow. At the narrowest region in the venturi the random air molecules will be drawn

(a) Rough surface (b) Smooth surface
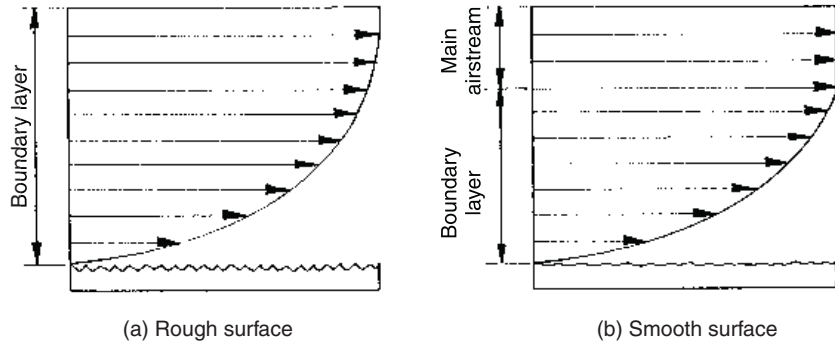
Fig. 14.4 (a and b) Influence of surface roughness on boundary layer velocity profile
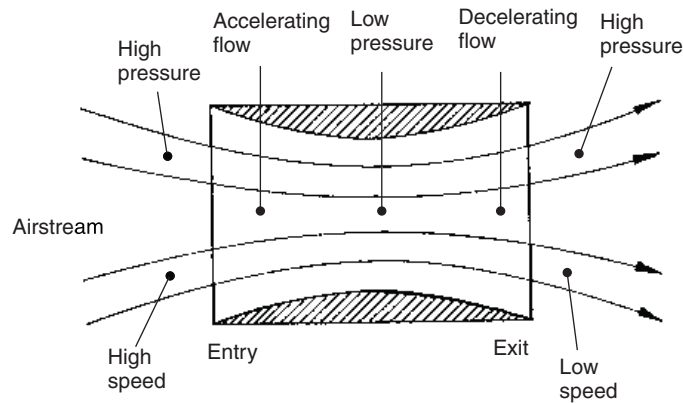
Fig. 14.5 Venturi
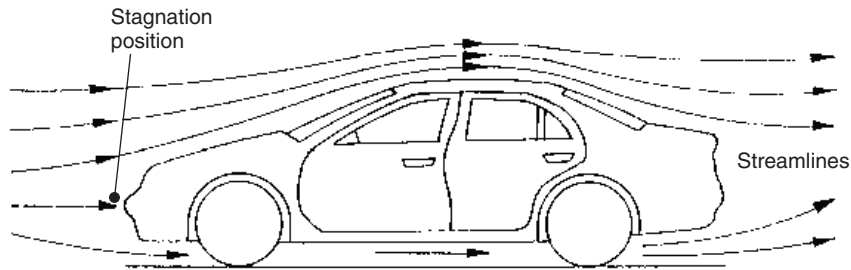
Fig. 14.6 Streamline air flow around car

apart thus creating a pressure drop and a faster movement of the air. Further downstream the air moves into the diverging or expanding section of the venturi where the air flow decelerates, the molecules therefore are able to move closer together and by the time the air reaches the exit its pressure will have risen again and its movement slows down.

14.1.5 Air streamlines (Fig. 14.6)
A moving car displaces the air ahead so that the air is forced to flow around and towards the rear. The pattern of air movement around the car can be visualized by airstreamlines which are imaginary lines across which there is no flow, see Fig. 14.6. These streamlines broadly follow the contour of the body but any sudden change in the car's shape

High pressure low speed Converging accelerating flow Low pressure high speed Diverging decelerating flow High pressure low speed

Low pressure (subatmospheric pressure) high speed
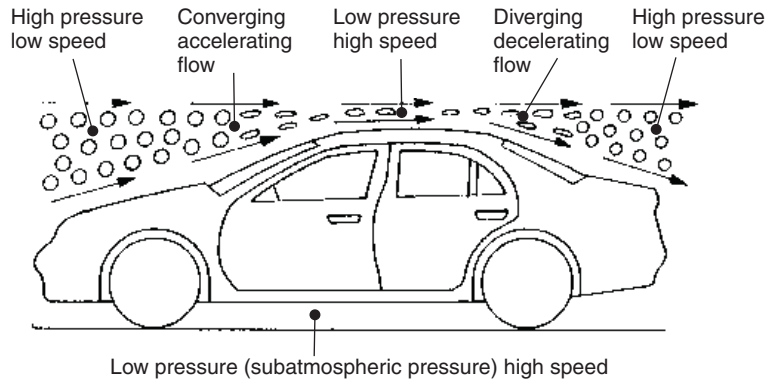
Fig. 14.7 Relative air speed and pressure conditions over the upper profile of a moving car

compels the streamlines to deviate, leaving zones of stagnant air pockets. The further these streamlines are from the body the more they will tend to straighten out.

14.1.6 Relative air speed and pressure conditions over the upper profile of a moving car (Figs 14.7 and 14.8)

The space between the upper profile of the horizontal outer streamlines relative to the road surface generated when the body is in motion can be considered to constitute a venturi effect, see Fig. 14.7. Note in effect it is the car that moves whereas air remains stationary; however, when wind-tunnel tests are carried out the reverse happens, air is drawn through the tunnel with the car positioned inside on a turntable so that the air passes over and around the parked vehicle. The air gap between the horizontal airstreamlines and front end bonnet (hood) and windscreen profile and the back end screen and boot (trunk) profile produces a diverging and converging air wedge, respectively. Thus the air scooped into the front wedge can be considered initially to be at atmospheric pressure and moving at car speed. As the air moves into the diverging wedge it has to accelerate to maintain the rate of volumetric displacement. Over the roof the venturi is at its narrowest, the air movement will be at its highest and the air molecules will be stretched further apart, consequently there will be a reduction in air pressure in this region. Finally the relative air movement at the rear of the boot will have slowed to car speed, conversely its pressure will have again risen to the surrounding atmospheric pressure conditions, thus allowing the random network of distorted molecules to move closer together to a more stable condition. As the air moves beyond the roof into the diverging wedge

region it decelerates to cope with the enlarged flow space.

As can be seen in Fig. 14.8 the pressure conditions over and underneath the car's body can be plotted from the data; these graphs show typical pressure distribution trends only. The pressure over the rear half of the bonnet to the mid-front windscreen region where the airstream speed is slower is positive (positive pressure coefficient C_p), likewise the pressure over the mid-position of the rear windscreen and the rear end of the boot where the airstream speed has been reduced is also positive but of a lower magnitude. Conversely the pressure over the front region of the bonnet and particularly over the windscreen/roof leading edge and the horizontal roof area where the airstream speed is at its highest has a negative pressure (negative pressure coefficient C_p). When considering the air movement underneath the car body, the restricted airstream flow tends to speed up the air movement thereby producing a slight negative pressure distribution along the whole underside of the car. The actual pattern of pressure distribution above and below the body will be greatly influenced by the car's profile style, the vehicle's speed and the direction and intensity of the wind.

14.1.7 Lamina boundary layer (Fig. 14.9(a))

When the air flow velocity is low sublayers within the boundary layer are able to slide one over the other at different speeds with very little friction; this kind of uniform flow is known as lamina.

14.1.8 Turbulent boundary layer (Fig. 14.9(b))

At higher air flow velocity the sublayers within the boundary layer also increase their relative sliding speed until a corresponding increase in interlayer friction compels individual sublayers to randomly
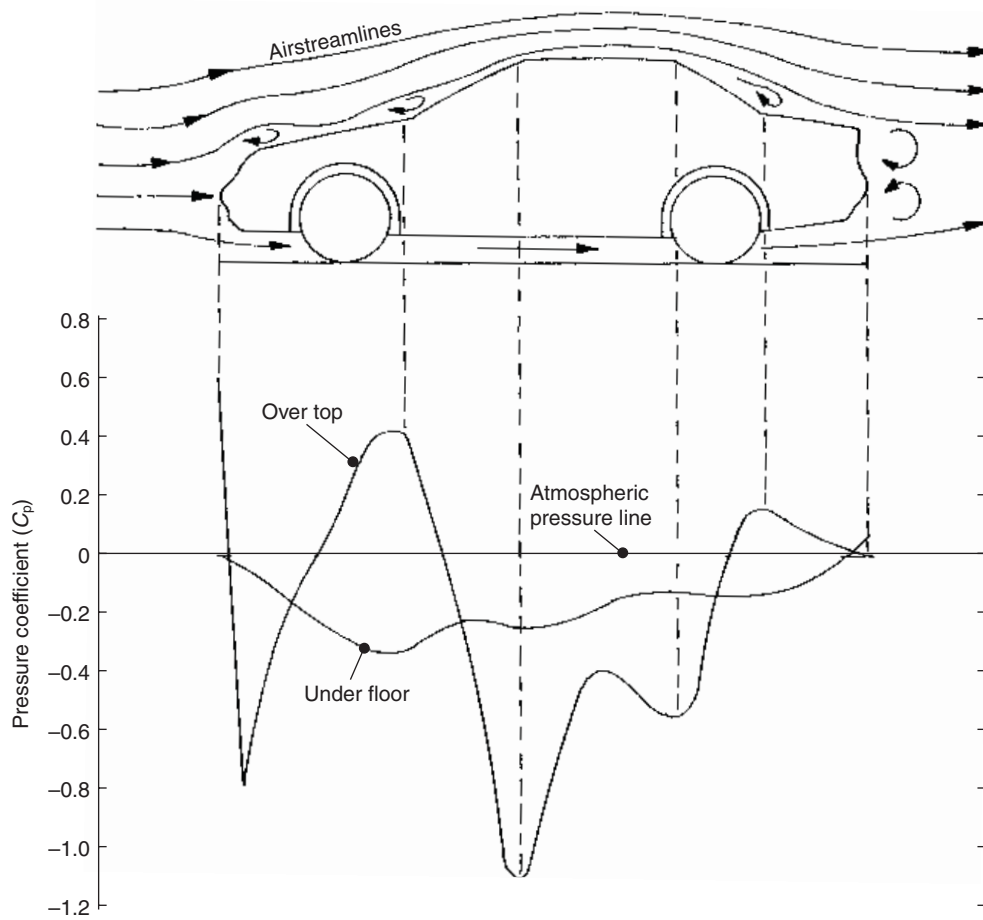
Fig. 14.8   Pressure distribution above and below the body structure

(a) Lamina air flow (low velocity)   (b) Turbulent air flow (high velocity)
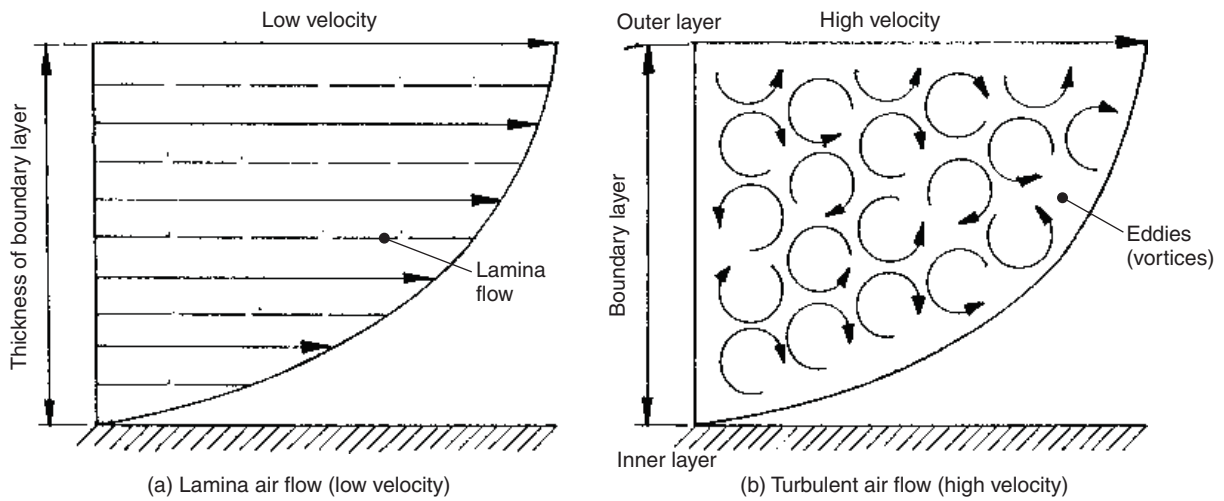
Fig. 14.9 (a and b)   Lamina and turbulent air flow

L = Lamina
T = Turbulent
TP = Transition point

(a) Low speed

(b) High speed
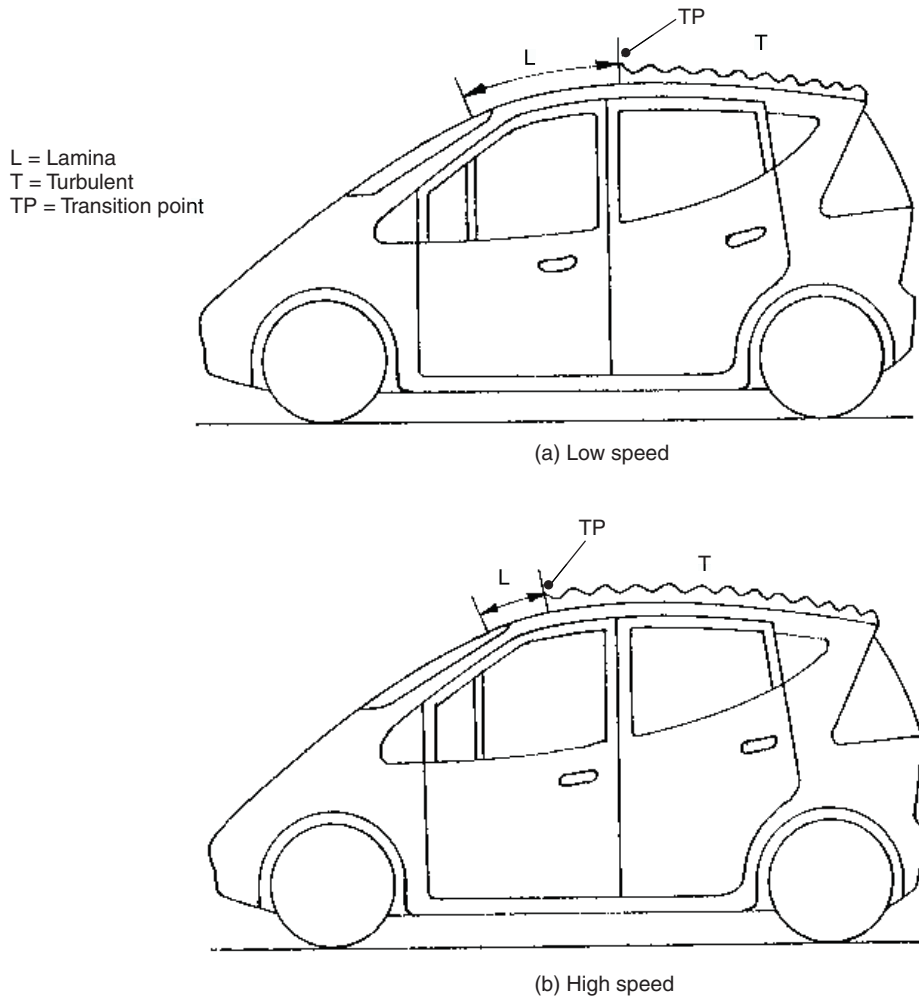
Fig. 14.10 (a and b) Lamina/turbulent boundary layer transition point

break away from the general direction of motion; they then whirl about in the form of eddies, but still move along with the air flow.

14.1.9 *Lamina/turbulent boundary layer transition point* (Fig. 14.10(a and b))

A boundary layer over the forward surface of a body, such as the roof, will generally be lamina, but further to the rear a point will be reached called the transition point when the boundary layer changes from a lamina to a turbulent one, see Fig. 14.10(a). As the speed of the vehicle rises the transition point tends to move further to the front, see Fig. 14.10(b), therefore less of the boundary layer will be lamina and more will become turbulent; accordingly this will correspond to a higher level of skin friction.

14.1.10 *Flow separation and reattachment* (Fig. 14.11(a and b))

The stream of air flowing over a car's body tends to follow closely to the contour of the body unless there is a sudden change in shape, see Fig. 14.11(a). The front bonnet (hood) is usually slightly curved and slopes up towards the front windscreen, from here there is an upward windscreen tilt (rake), followed by a curved but horizontal roof; the rear windscreen then tilts downwards where it either merges with the boot (trunk) or continues to slope gently downwards until it reaches the rear end of the car.

The air velocity and pressure therefore reaches its highest and lowest values, respectively, at the top of the front windscreen; however, towards the rear of the roof and when the screen tilts downwards

589
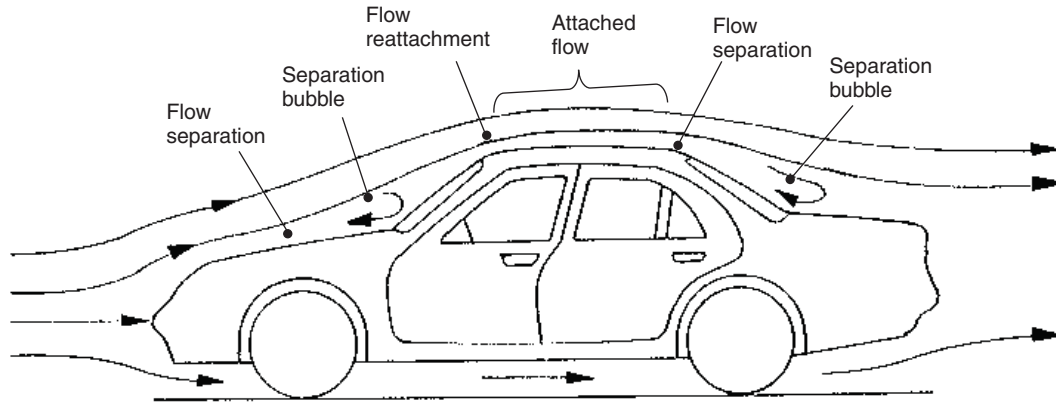
(a) Notch front and rear windscreens
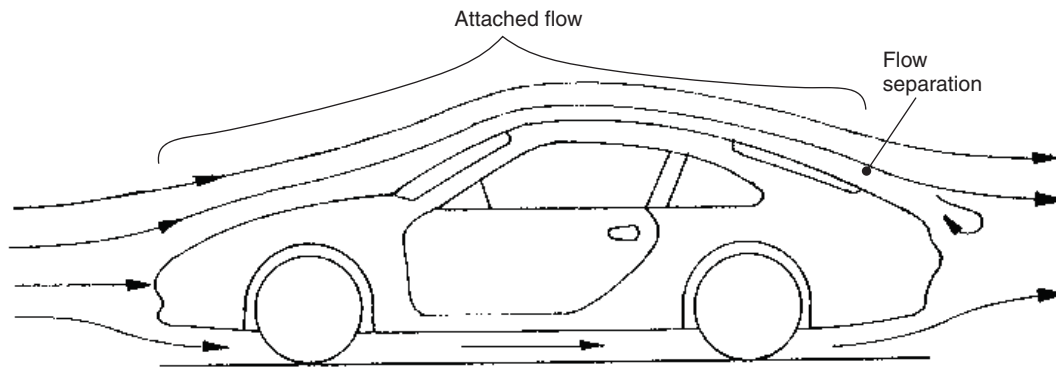
(b) Very streamlined shape

Fig. 14.11 (a and b) Flow separation and reattachment

there will be a reduction in air speed and a rise in pressure. If the rise in air pressure towards the rear of the car is very gradual then mixing of the airstream with the turbulent boundary layers will be relatively steady so that the outer layers will be drawn along with the main airstream, see Fig. 14.11(b). Conversely if the downward slope of the rear screen/boot is considerable, see Fig. 14.11(a), the pressure rise will be large so that the mixing rate of mainstream air with the boundary layers cannot keep the inner layers moving, consequently the slowed down boundary layers thicken. Under these conditions the mainstream air flow breaks away from the contour surface of the body, this being known as flow separation. An example of flow separation followed by reattachment can be visualized with air flowing over the bonnet and

front windscreen; if the rake angle between the bonnet and windscreen is large, the streamline flow will separate from the bonnet and then reattach itself near the top of the windscreen or front end of the roof, see Fig. 14.11(a). The space between the separation and reattachment will then be occupied by circulating air which is referred to as a separation bubble, and if this rotary motion is vigorous a transverse vortex will be established.

14.2 Aerodynamic drag

14.2.1 Pressure (form) drag (Figs 14.12(a–e) and 14.13)
When viscous air flows over and past a solid form, vortices are created at the rear causing the flow

590

(a) Flat plate

Direction of air flow

Vortices

−Ve

(b) Circular section

(c) Circular/lobe section

(d) Aerofoil section

(e) Fineness ratio (b/a)
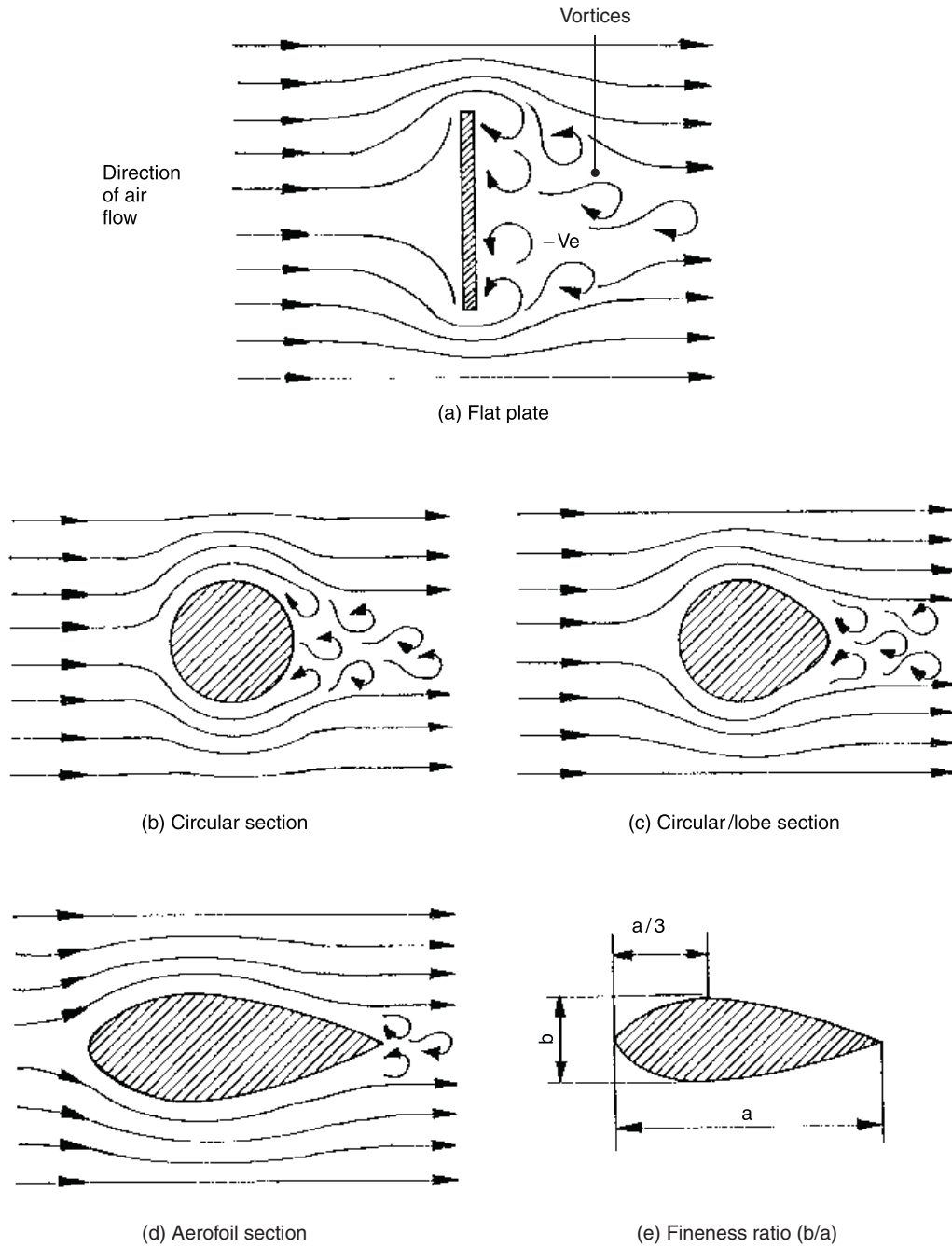
Fig. 14.12 (a–e)   Air flow over various shaped sections

to deviate from the smooth streamline flow, see Fig. 14.12(a). Under these conditions the air flow pressure in front of the solid object will be higher than atmospheric pressure while the pressure behind will be lower than that of the atmosphere, consequently the solid body will be dragged (sucked) in the direction of air movement. Note that this effect is created in addition to the skin friction drag. An extreme example of pressure drag (sometimes known as form drag) can be seen in Fig. 14.13 where a flat plate placed at right angles to the air movement will experience a drag force in the

591

direction of flow represented by the pulley weight which opposes the movement of the plate.

Pressure drag can be reduced by streamlining any solid form exposed to the air flow, for instance a round tube (Fig. 14.12(b)) encourages the air to flow smoothly around the front half and part of the rear before flow separation occurs thereby reducing the resistance by about half that of the flat plate. The resistance of a tube can be further reduced to about 15% of the flat plate by extending the rear of the circulating tube in the form of a curved tapering lobe, see Fig. 14.12(c). Even bigger reductions in resistance can be achieved by proportioning the tube section (see Fig. 14.12(d)) with a fineness ratio a/b of between 2 and 4 with the maximum thickness b set about one-third back from the nose, see Fig. 14.12(e). This gives a flow resistance of roughly one-tenth of a round tube or 5% of a flat plate.

14.2.2 Air resistance opposing the motion of a vehicle (Fig. 14.13)

The formula for calculating the opposing resistance of a body passing though air can be derived as follows:

Let us assume that a flat plate body (Fig. 14.13) is held against a flow of air and that the air particles are inelastic and simply drop away from the perpendicular plate surface. The density of air is the mass per unit volume and a cubic metre of air at sea level has an approximate mass of 1.225 kg, therefore the density of air is 1.225 kg/m^3.

Then let

$$\text{Mass} = m \text{ kg}$$
$$\text{Volume} = Q \text{ m}^3$$
$$\text{Density} = \rho \text{ kg/m}$$

Hence

$$\rho = \frac{m}{Q} \text{ kg/m}^{-3}$$

or

$$m = \rho Q \left(\frac{\text{kg}}{\text{m}^3} \text{m}^3 \right) \text{kg}$$

Let

$$\text{Density of air flow} = \rho \text{ kg/m}^3$$
$$\text{Frontal area of plate} = A \text{ m}^2$$
$$\text{Velocity of air striking surface} = v \text{ m/s}$$
$$\text{Volume of air striking plate per second} = Q = vA \text{ m}^3$$
$$\text{Mass movement of air per second} = \rho Q = \rho \times vA$$
$$\text{since} \quad Q = vA$$
$$\text{Momentum of this air (mv)} = \rho vA \times v$$
$$\text{therefore momentum lost by air per second} = \rho A v^2$$

From Newton's second law the rate at which the movement of air is changed will give the force exerted on the plate.
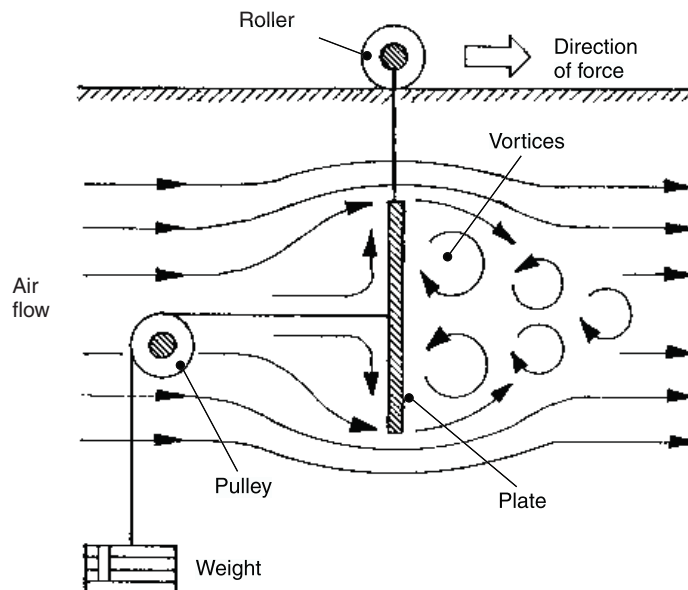
Fig. 14.13 Pressure drag apparatus

Hence

$$\text{force on plate} = \rho\, A v^2 \quad \text{Newton's}$$

However, the experimental air thrust against a flat plate is roughly 0.6 of the calculated $\rho A v^2$ force. This considerable 40% error is basically due to the assumption that the air striking the plate is brought to rest and falls away, where in fact most of the air escapes round the edges of the plate and the flow then becomes turbulent. In fact the theoretical air flow force does not agree with the actual experimental force (F) impinging on the plate, but it has been found to be proportional to $\rho\, A v^2$

hence

$$F \;\propto\; A\, v^2$$

therefore air resistance $F = C_D\, A\, v^2$ where C_D is the coefficient of proportionality.

The constant C_D is known as the coefficient of drag, it has no unity and its value will depend upon the shape of the body exposed to the airstream.

14.2.3 After flow wake (Fig. 14.14)

This is the turbulent volume of air produced at the rear end of a forward moving car and which tends to move with it, see Fig. 14.14. The wake has a cross-sectional area equal approximately to that of the rear vertical boot panel plus the rearward projected area formed between the level at which the air flow separates from the downward sloping rear window panel and the top edge of the boot.

14.2.4 Drag coefficient

The aerodynamic drag coefficient is a measure of the effectiveness of a streamline aerodynamic body shape in reducing the air resistance to the forward motion of a vehicle. A low drag coefficient implies that the streamline shape of the vehicle's body is such as to enable it to move easily through the surrounding viscous air with the minimum of resis-

tance; conversely a high drag coefficient is caused by poor streamlining of the body profile so that there is a high air resistance when the vehicle is in motion. Typical drag coefficients for various classes of vehicles can be seen as follows:

Vehicle type drag coefficient C_D	
Saloon car	0.22–0.4
Sports car	0.28–0.4
Light van	0.35–0.5
Buses and coaches	0.4–0.8
Articulated trucks	0.55–0.8
Ridged truck and draw bar trailer	0.7–0.9

14.2.5 Drag coefficients and various body shapes
(Fig. 14.15(a–f))
A comparison of the air flow resistance for different shapes in terms of drag coefficients is presented as follows:

(a) *Circular plate* (Fig. 14.15(a))   Air flow is head on, and there is an immediate end on pressure difference. Flow separation takes place at the rim; this provides a large vortex wake and a correspondingly high drag coefficient of 1.15.

(b) *Cube* (Fig. 14.15(b))   Air flow is head on but a boundary layer around the sides delays the flow separation; nevertheless there is still a large vortex wake and a high drag coefficient of 1.05.

(c) *Sixty degree cone* (Fig. 14.15(c))   With the piecing cone shape air flows towards the cone apex and then spreads outwards parallel to the shape of the cone surface. Flow separation however still takes place at the periphery thereby producing a wide vortex wake. This profile halves the drag coefficient to about 0.5 compared with the circular plate and the cube block.

Fig. 14.14   After flow wake

(a) Circular disc ($C_D = 1.15$) (b) Cube ($C_D = 1.05$) (c) 60° cone ($C_D = 0.5$)

(d) Sphere ($C_D = 0.47$) (e) Hemisphere ($C_D = 0.42$) (f) Tear drop ($C_D = 0.05$)
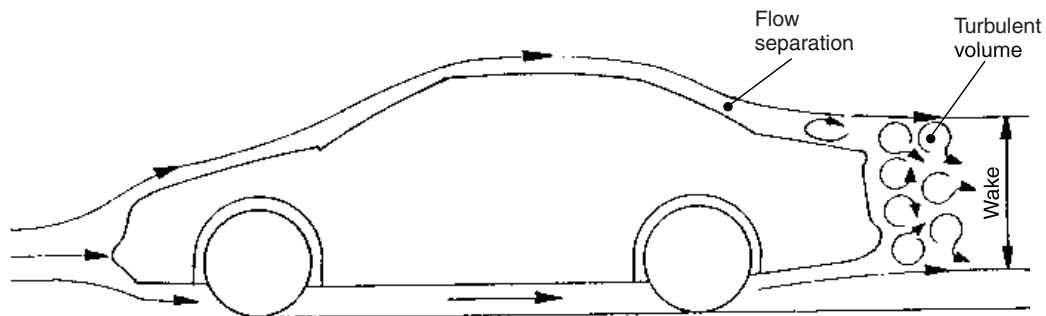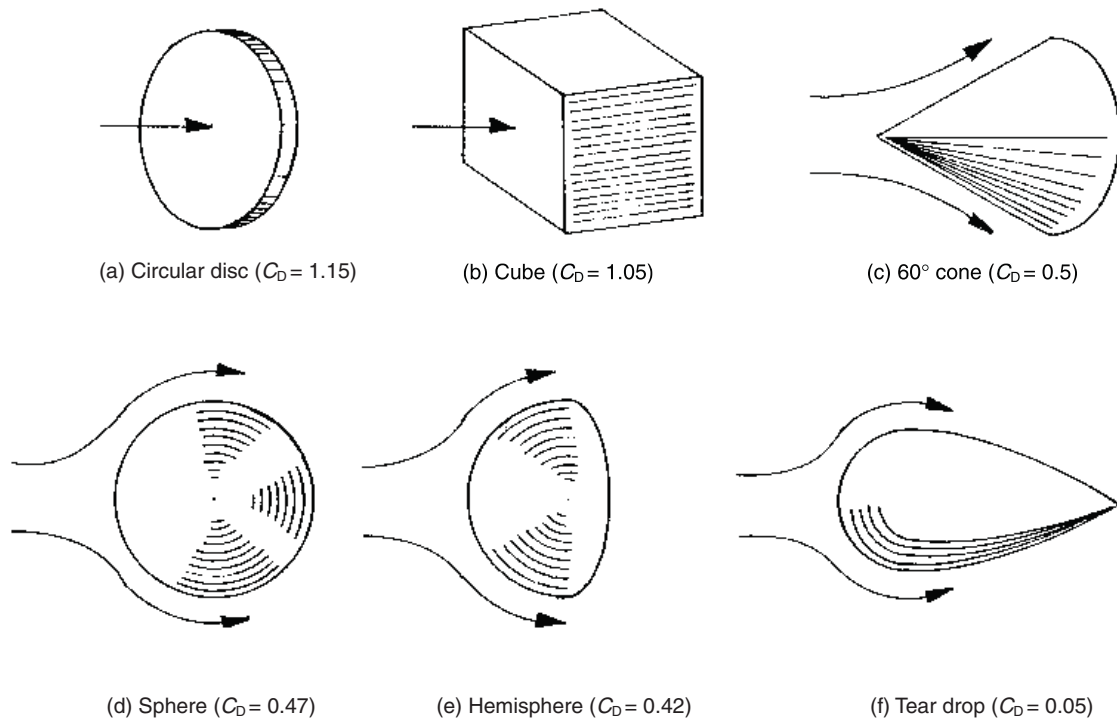
Fig. 14.15 (a–e) Drag coefficient for various shaped solids

(d) *Sphere* (Fig. 14.15(d)) Air flow towards the sphere, it is then diverted so that it flows outwards from the centre around the diverging surface and over a small portion of the converging rear half before flow separation occurs. There is therefore a slight reduction in the vortex wake and similarly a marginal decrease in the drag coefficient to 0.47 compared with the 60° cone.

(e) *Hemisphere* (Fig. 14.15(e)) Air flow towards and outwards from the centre of the hemisphere. The curvature of the hemisphere gradually aligns with the main direction of flow after which flow separation takes place on the periphery. For some unknown reason (possibly due to the very gradual alignment of surface curvature with the direction of air movement near the rim) the hemisphere provides a lower drag coefficient than the cone and the sphere shapes this, being of the order of 0.42.

(f) *Tear drop* (Fig. 14.15(f)) If the proportion of length to diameter is well chosen, for example 0.25, the streamline shape can maintain a boundary layer before flow separation occurs almost to the end of its tail. Thus the resistance to body movement will be mainly due to viscous air flow and little to do with vortex wake suc-

tion. With these contours the drag coefficient can be as low as 0.05.

14.2.6 Base drag (Fig. 14.16(a and b))
The shape of the car body largely influences the pressure drag. If the streamline contour of the body is such that the boundary layers cling to a converging rear end, then the vortex area is considerably reduced with a corresponding reduction in rear end suction and the resistance to motion. If the body was shaped in the form of a tear drop, the contour of the body would permit a boundary layer to continue a considerable way towards the tail before flow separation occurs, see Fig. 14.16(a), consequently the area heavily subjected to vortex swirl and negative pressure will be at a minimum. However, it is impractical to design a tear drop body with an extended tapering rear end, but if the tail is cut off (bobtailed) at the point where the air flow separates from the contour of the body (see Fig. 14.16(b)), the same vortex (negative pressure) exists as if the tail was permitted to converge. The cut off cross-section area where flow separation would occur is known as the 'base area' and the negative vortex pressure produced is referred to as the 'base drag'. Thus there is a trend

(a) Tear drop shaped body

(b) Bobtailed tear drop
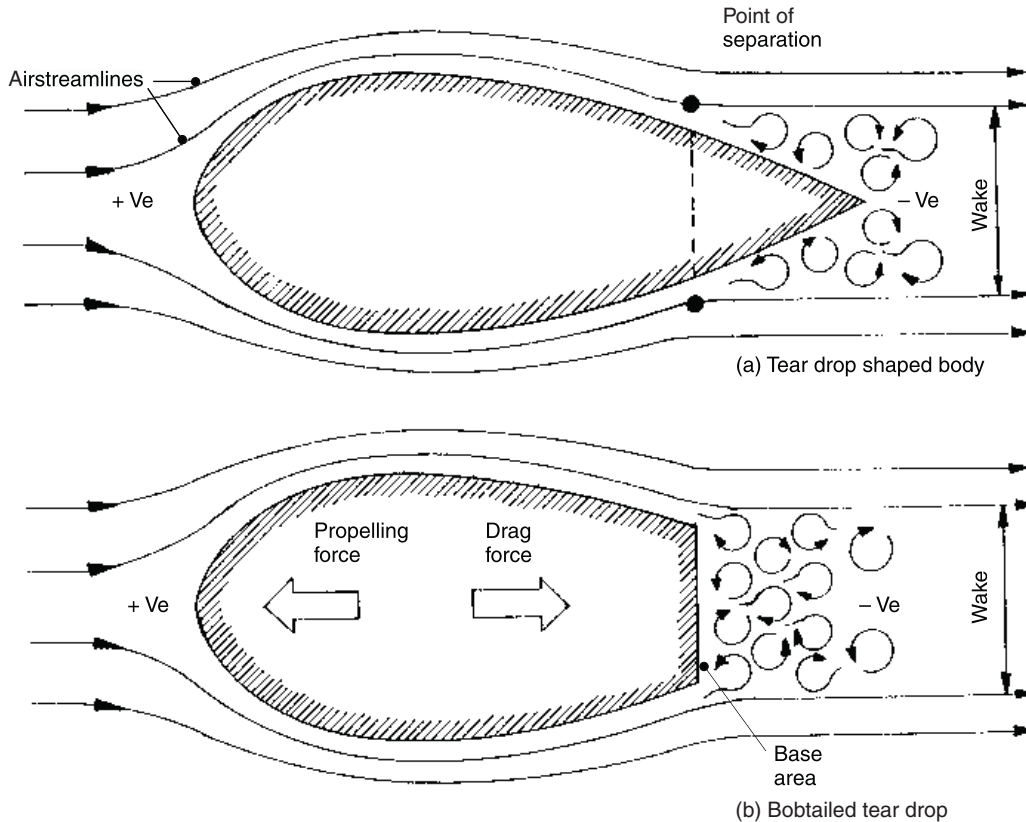
Fig. 14.16 (a and b)   Base drag

for car manufacturers to design bodies that taper slightly towards the rear so that flow separation occurs just beyond the rear axle.

14.2.7 Vortices (Fig. 14.17)

Vortices are created around various regions of a vehicle when it is in motion. Vortices can be described as a swirling air mass with an annular cylindrical shape, see Fig. 14.17. The rotary speed at the periphery is at its minimal, but this increases inversely with the radius so that its speed near the centre is at a maximum. However, there is a central core where there is very little movement, consequently viscous shear takes place between adjacent layers of the static core and the fast moving air swirl; thus the pressure within the vortex will be below atmospheric pressure, this being much lower near the core than in the peripheral region.

14.2.8 Trailing vortex drag (Fig. 14.18(a and b))

Consider a car with a similar shape to a section of an aerofoil, see Fig. 14.18(a), when air flows

from the front to the rear of the car, the air moves between the underside and ground, and over the raised upper body profile surfaces. Thus if the upper and lower airstreams are to meet at the rear at a common speed the air moving over the top must move further and therefore faster than the more direct underfloor airstream. The air pressure will therefore be higher in the slower underfloor airstream than that for the faster airstream moving over the top surface of the car. Now air moves from high to low pressure regions so that the high pressure airstream underneath the car will tend to move diagonally outwards and upwards towards the low pressure airstream flowing over the top of the body surface (see Fig. 14.18(b)). Both the lower and the upper airstreams eventually interact along the side-to-top profile edges on opposite sides of the body to form an inward rotary air motion that continues to whirl for some distance beyond the rear end of the forward moving car, see Fig. 14.18(a and b). The magnitude and intensity of these vortices will to a great extent depend upon the rear styling of the

595

Fig. 14.17 The vortex

(a) Pictorial view

(b) Plan view
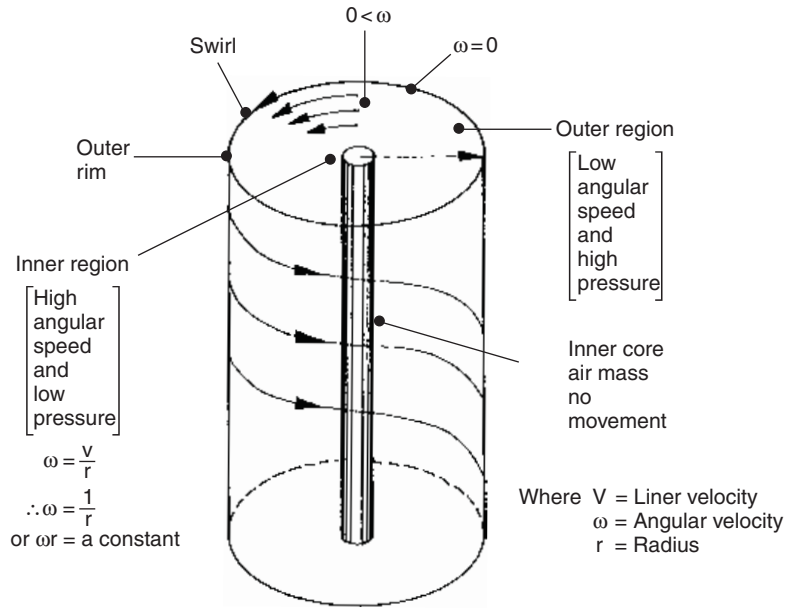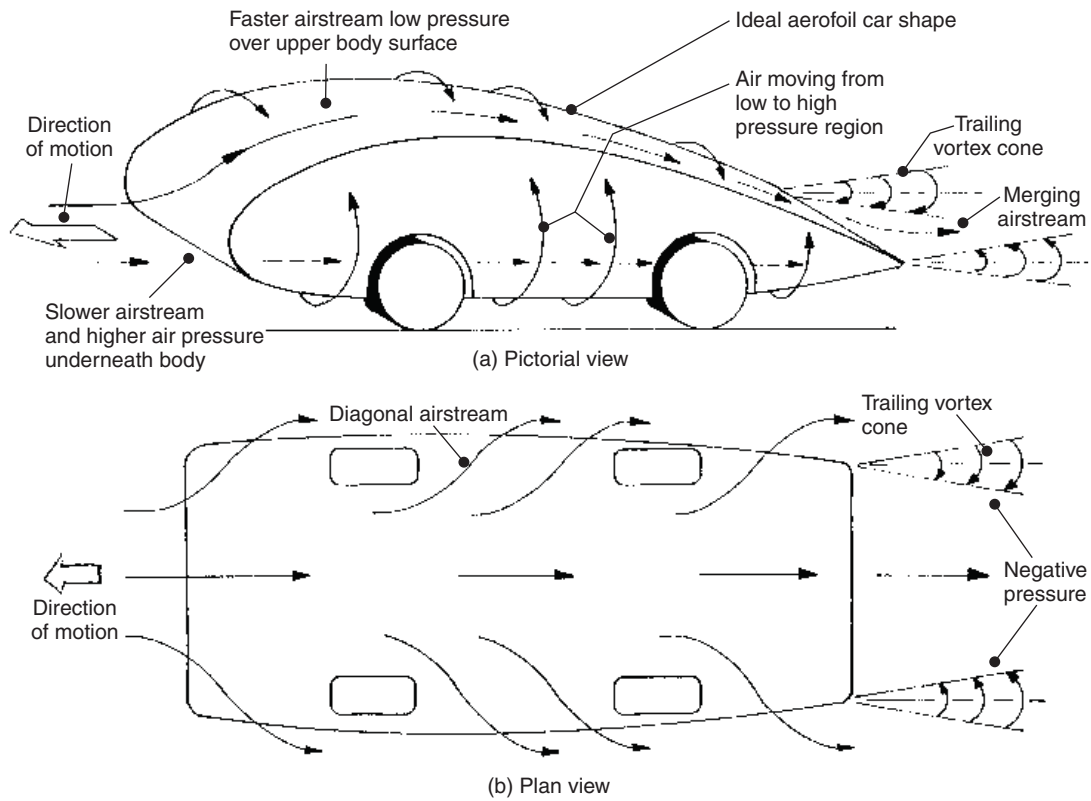
Fig. 14.18 (a and b) Establishment of trailing vortices

car. The negative (below atmospheric) pressure created in the wake of the trailing vortices at the rear of the car attempts to draw it back in the opposite direction to the forward propelling force; this resistance is therefore referred to as the 'trailing vortex drag'.

14.2.9 Attached transverse vortices
(Fig. 14.19(a and b))
Separation bobbles which form between the bonnet (hood) and front windscreen, the rear screen and boot (trunk) lid and the boot and rear light panel tend to generate attached transverse vortices (see Fig. 14.19(a and b)). The front attached vortices work their way around the 'A' post and then extend along the side windows to the rear of the car and beyond. Any overspill from the attached vortices in the rear window and rear light panel regions merges and strengthens the side panel vortices (see Fig. 14.19(b)); in turn the products of these secondary transverse vortices combine and enlarge the main trailing vortices.

(a) Front and side vortices

(b) Rear and side vortices
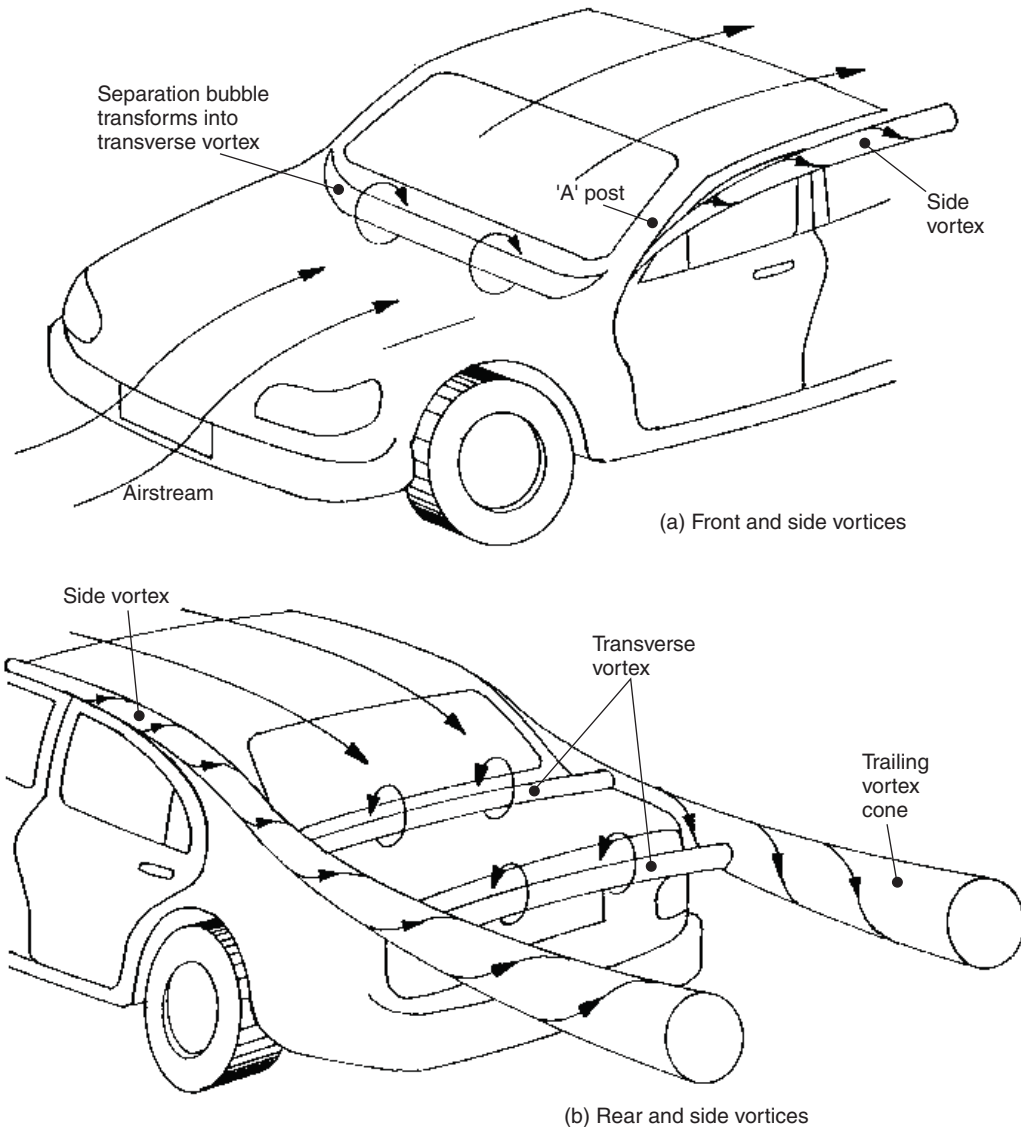
Fig. 14.19 (a and b) Notch back transverse and trailing vortices

14.3 Aerodynamic lift

14.3.1 Lift coefficients

The aerodynamic lift coefficient C_L is a measure of the difference in pressure created above and below a vehicle's body as it moves through the surrounding viscous air. A resultant upthrust or downthrust may be produced which mainly depend upon the body shape; however, an uplift known as positive lift is undesirable as it reduces the tyre to ground grip whereas a downforce referred to as negative lift enhances the tyre's road holding.

14.3.2 Vehicle lift (Fig. 14.20)

When a car travels along the road the airstream moving over the upper surface of the body from front to rear has to move further than the underside airstream which almost moves in a straight line (see Fig. 14.20). Thus the direct slower moving underside and the indirect faster moving top side airstream produces a higher pressure underneath the car than over it, consequently the resultant vertical pressures generated between the upper and under surfaces produce a net upthrust or lift. The magnitude of the lift depends mainly upon the styling profile of both over and under body surfaces, the distance of the underfloor above the ground, and the vehicle speed. Generally, the nearer the underfloor is to the ground the greater the positive lift (upward force); also the positive lift tends to increase with the square of the vehicle speed. Correspondingly a reduction in wheel load due to the lift upthrust counteracts the downward load; this

therefore produces a reduction in the tyre to ground grip. If the uplift between the front and rear of the car is different, then the slip-angles generated by the front and rear tyres will not be equal; accordingly this will result in an under- or over-steer tendency instead of more neutral-steer characteristics. Thus uncontrolled lift will reduce the vehicle's road holding and may cause steering instability.

14.3.3 Underbody floor height versus aerodynamic lift and drag

(Figs 14.21(a and b) and 14.22)

With a large underfloor to ground clearance the car body is subjected to a slight negative lift force (downward thrust). As the underfloor surface moves closer to the ground the underfloor air space becomes a venturi, causing the air to move much faster underneath the body than over it, see Figs 14.21(a) and 14.22. Correspondingly with these changing conditions the air flow pressure on top of the body will be higher than for the underbody reduced venturi effect pressure, hence there will be a net down force (negative lift) tending to increase the contact pressure acting between the wheels and ground. Conversely a further reduction in underfloor to ground clearance makes it very restrictive for the underbody air flow (see Figs 14.21(b) and 14.22), so that much of the airstream is now compelled to flow over the body instead of underneath it, which results in an increase in air speed and a reduction in pressure over the top to cope with the reduction in the underfloor air
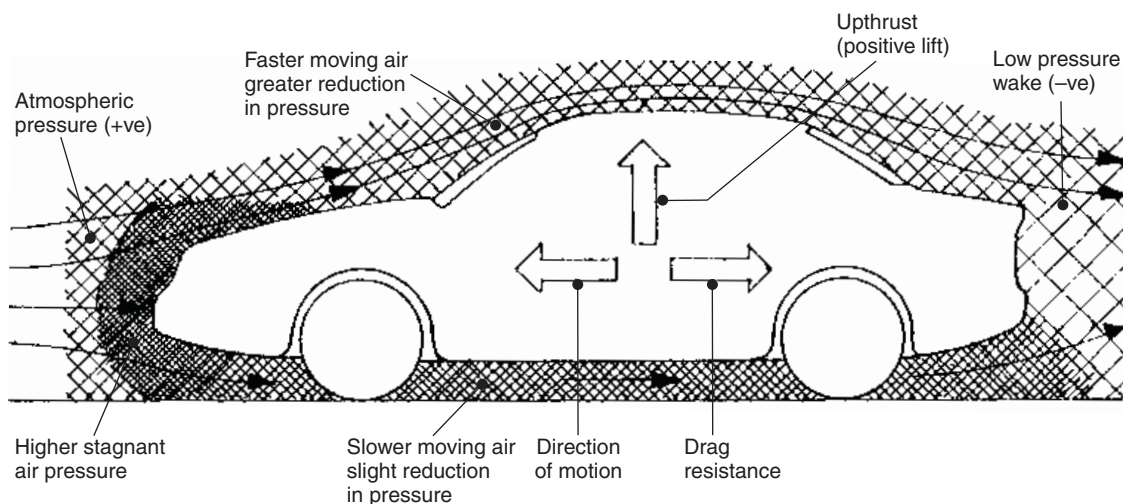
Fig. 14.20 Aerodynamic lift

(a) Large ground clearance (negative lift downthrust)

Small reduction in air speed small increase in pressure (compared with Fig. 14.21(b))

Small increase in air speed small reduction in pressure (compared with Fig. 14.21(b))

Negative lift (−ve)

h

Venturi effect

(b) Small ground clearance (positive lift upthrust)

Fast air flow low pressure

Slow air flow high pressure

h

Positive lift (+ve)

Fig. 14.21 (a and b) Effects of underfloor to ground clearance on the surrounding air speed, pressure and aerodynamic lift

movement. Thus the over and under pressure conditions have been reversed which subsequently now produces a net upward suction, that is, a tendency toward a positive lift.

14.3.4 Aerofoil lift and drag
(Figs 14.23(a–d), 14.24(a and b) and 14.25)
Almost any object moving through an airstream will be subjected to some form of lift and drag. Consider a flat plate inclined to the direction of air flow, the pressure of air above the surface of the plate is reduced while that underneath it is increased. As a result there will be a net pressure on the plate striving to force it both upwards and backwards, see Fig. 14.23(a). It will be seen that the vertical and horizontal components of the resultant reaction represents both lift and drag respectively, see Fig. 14.23(b). The greater the angle of inclination, the smaller will be the upward lift component, while the backward drag component

will increase, see Fig. 14.23(c and d). Conversely as the angle of inclination decreases, the lift increases and the drag decreases; however, as the angle of inclination is reduced so does the resultant reaction force. If an aerofoil profile is used instead of the flat plate, (see Fig. 14.24(a and b)), the airstream over the top surface now has to move further and faster than the underneath air movement. This produces a greater pressure difference between the upper and lower surfaces and consequently greatly enhances the aerodynamic lift and promotes a smooth air flow over the upper profiled surface. A typical relationship between the C_L, C_D and angle of attack (inclination angle) is shown for an aerofoil section in Fig. 14.25.

14.3.5 Front end nose shape (Fig. 14.26(a–c))
Optimizing a protruding streamlined nose profile shape influences marginally the drag coefficient and to a greater extent the front end lift coefficient.
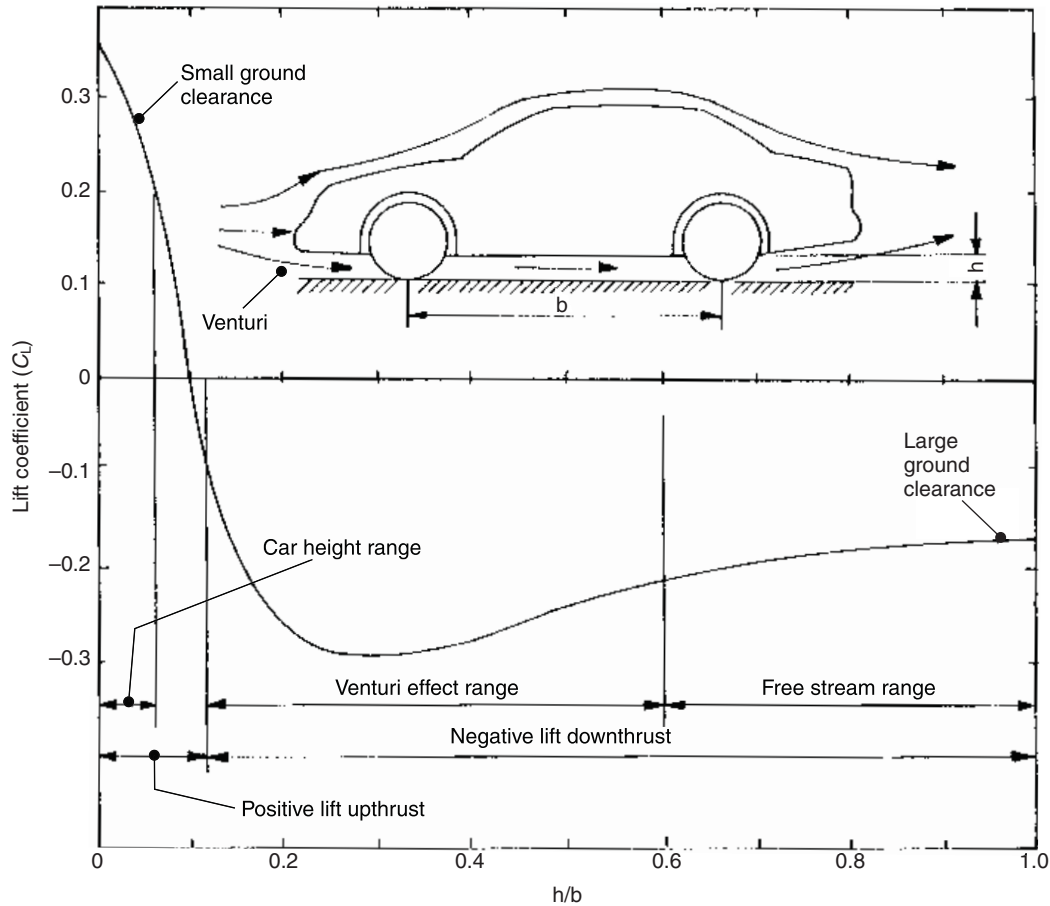
Fig. 14.22 Aerodynamic lift versus ground, floor height

With a downturned nose (see Fig. 14.26(a)) the streamlined nose profile directs the largest proportion of the air mass movement over the body, and only a relatively small amount of air flows underneath the body. If now a central nose profile is adopted (see Fig. 14.26(b)) the air mass movement is shared more evenly between the upper and lower body surfaces; however, the air viscous interference with the underfloor and ground still causes the larger proportion of air to flow above than below the car's body. Conversely a upturned nose (see Fig. 14.26(c)) induces still more air to flow beneath the body with the downward curving entry gap shape producing a venturi effect. Consequently the air movement will accelerate before reaching its highest speed further back at its narrowest body to ground clearance. Raising the mass airflow in the space between the body and ground increases the viscous interaction of the

air with the under body surfaces and therefore forces the air flow to move diagonally out and upward from the sides of the car. It therefore strengthens the side and trailing vortices and as a result promotes an increase in front end aerodynamic lift force.

The three basic nose profiles discussed showed, under windtunnel tests, that the upturned nose had the highest drag coefficient C_D of 0.24 whereas there was very little difference between the central and downturned nose profiles which gave drag coefficients C_D of 0.223 and 0.224 respectively. However the front end lift coefficient C_L for the three shapes showed a marked difference, here the upturned nose profile gave a positive lift coefficient C_L of +0.2, the central nose profile provided an almost neutral lift coefficient C_L of +0.02, whereas the downturned nose profile generated a negative lift coefficient C_L of −0.1.

(a) Reaction force on an inclined plate

(b) Lift and drag components on an inclined plate

(c) Small angle of inclination

(d) Large angle of inclination
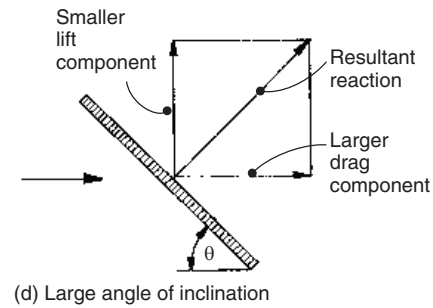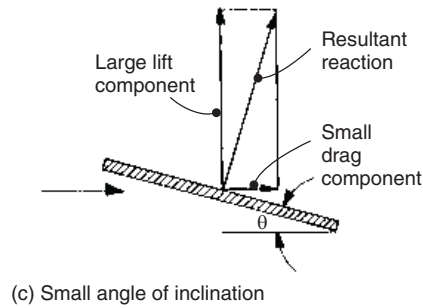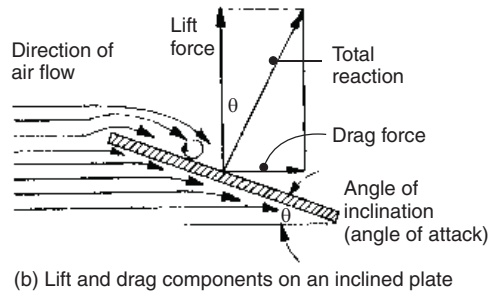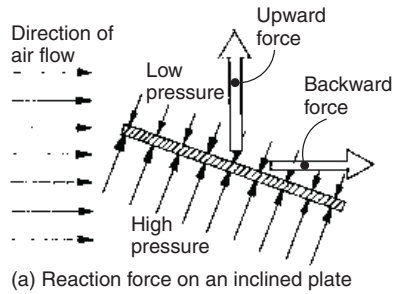
Fig. 14.23 (a–d) Lift and drag on a plate inclined at a small angle to the direction of air flow

14.4 Car body drag reduction

14.4.1 Profile edge rounding or chamfering
(Fig. 14.27(a and b))
There is a general tendency for aerodynamic lift and drag coefficients to decrease with increased edge radius or chamfer: experiments carried out showed for a particular car shape (see Fig. 14.27(a)) how the drag coefficient was reduced from 0.43 to 0.40 with an edge radius/chamfer increasing from zero to 40 mm (see Fig. 14.27(b)), and there was a slightly greater reduction with chamfering than

rounding the edges; however, beyond 40 mm radius there was no further advantage in increasing the edge radius or chamfer.

14.4.2 Bonnet slope and windscreen rake
(Fig. 14.28(a–c))
Increasing the bonnet (hood) slope angle α from zero to roughly 10° reduces the drag coefficient, but beyond 10° the drag reduction is insignificant, see Fig. 14.28(b). Likewise, increasing the rake angle γ reduces the drag coefficient (see Fig. 14.28(c)) particularly when the rake angle becomes large;

(a) Inclined plate

(b) Inclined aerofoil
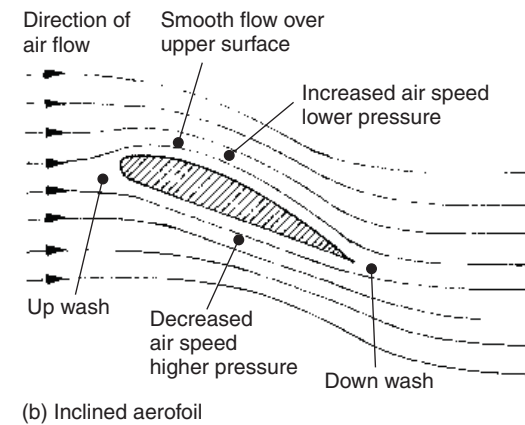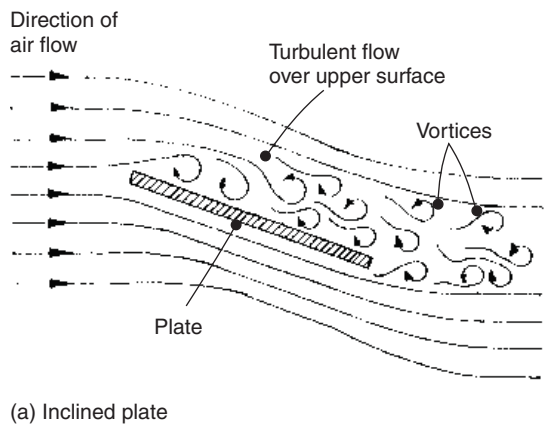
Fig. 14.24 (a and b) Air flow over a flat plate and aerofoil inclined at a small angle

601
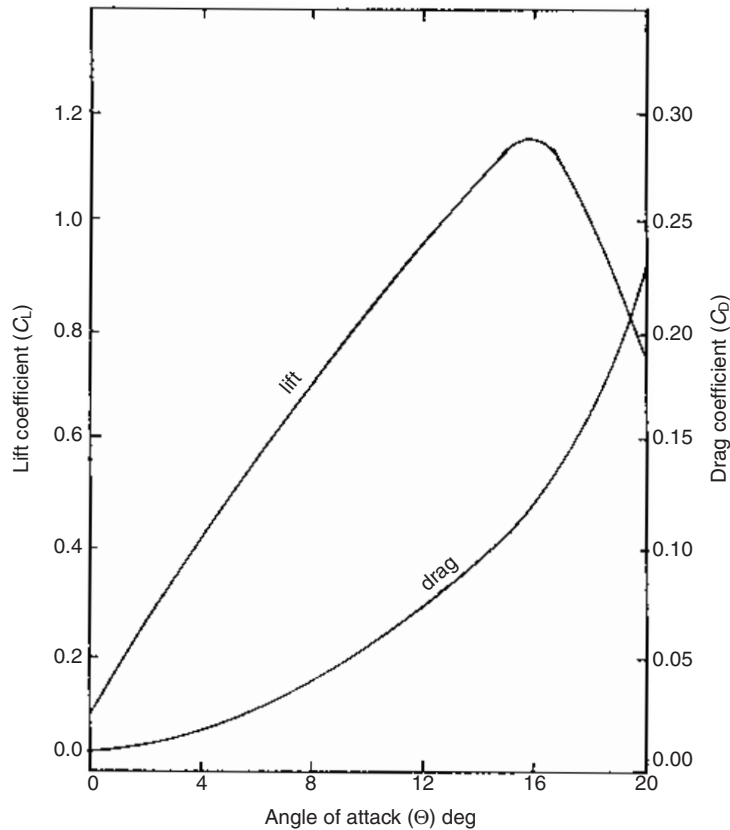
Fig. 14.25 Lift and drag coefficient versus angle of inclination (attack)

however, very large rake angles may conflict with the body styling.

14.4.3 Roof and side panel cambering
(Figs 14.29(a and b) and 14.30(a and b))
Cambering the roof (see Fig. 14.29(a and b)) and the side panels (see Fig. 14.30(a and b)) reduces the drag coefficient. However, if the roof camber curvature becomes excessive the drag coefficient commences to rise again (see Fig. 14.29(b)), whereas the reduction in drag coefficient with small amounts of side-panel cambering is marked (see Fig. 14.30(b)), but with excessive camber the reduction in the drag coefficient becomes only marginal. Both roof and side panel camber should not be increased at the expense of enlarging the frontal area of the car as this would in itself be counter-productive and would increase the drag coefficient.

14.4.4 Rear side panel taper
(Fig. 14.31(a and b))
Tapering inwards the rear side panel reduces the drag coefficient. This can be seen in Fig. 14.31

(a and b) which shows a marked reduction in the drag coefficient with both a 50 mm and then a 125 mm rear end contraction on either side of the car; however, there was no further reduction in the drag coefficient when the rear end contraction was increased to 200 mm.

14.4.5 Underbody rear end upward taper
(Fig. 14.32(a and b))
Tilting upwards the underfloor rear end produces a diffuser effect which shows a promising way to reduce the drag coefficient, see Fig. 14.32(a and b). However, it is important to select the optimum ratio of length of taper to overall car length and the angle β of upward inclination for best results.

14.4.6 Rear end tail extension
(Fig. 14.33(a and b))
Windtunnel investigation with different shaped tail models have shown that the minimal drag coefficients were produced with extended tails, see Fig. 14.33(a and b), but this shape would be impractical for design reasons. Conversely if the rear end tail is

602

Small mass
air flow below

Large mass
air flow overhead

Small trailing
vortices

Small diagonal
side flow

(a) Downturned nose profile

Medium mass
air flow below

Medium mass
air flow overhead

Medium trailing
vortices

Medium side
air flow

(b) Central nose profile

Large mass
air flow below

Small mass
air flow overhead

Large trailing
vortices

Large side
air flow

Viscous air
interaction

(c) Upturned nose profile

Fig. 14.26 (a–c) A greatly exaggerated air mass distribution around a car body for various nose profiles

cropped at various lengths and curved downwards there is an increase in the drag coefficient with each reduction in tail length beyond the rear wheels.

14.4.7 Underbody roughness
(Fig. 14.34(a and b))
The underbody surface finish influences the drag coefficient just as the overbody curvature, tapering,

edge rounding and general shape dictates the drag resistance. Moulding in individual compartments in the underfloor pan to house the various components and if possible enclosing parts of the underside with plastic panels helps considerably to reduce the drag resistance. The underside of a body has built into it many cavities and protrusions to cater with the following structural requirements and operating
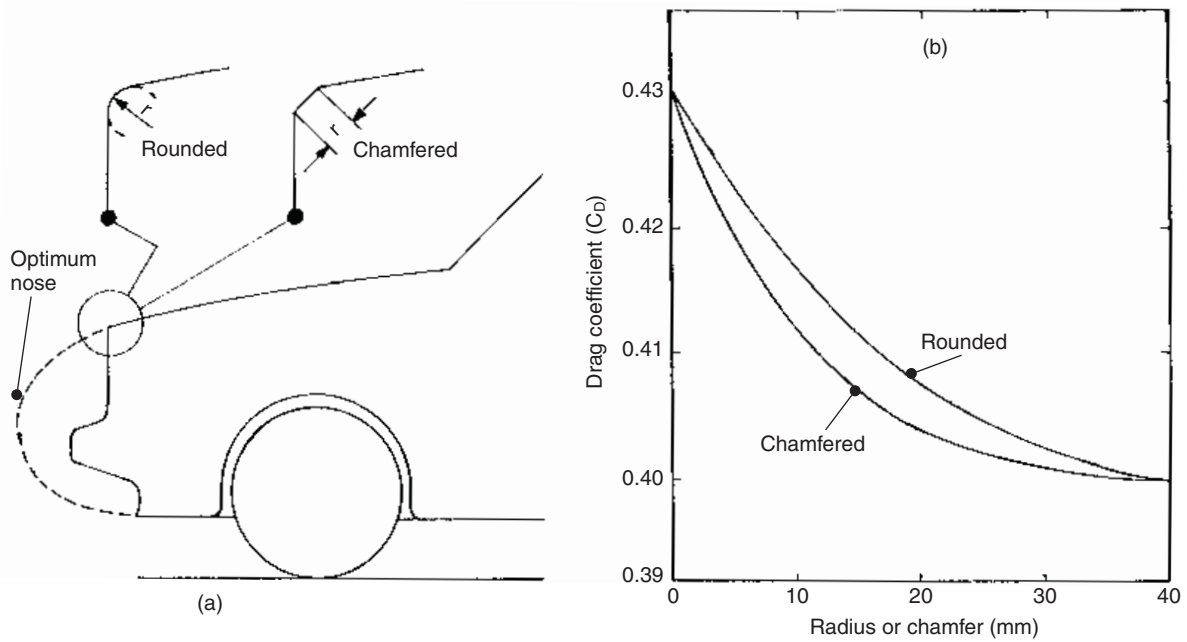
Fig. 14.27 (a and b)   Influence of forebody bonnet (hood) edge shape on drag coefficient
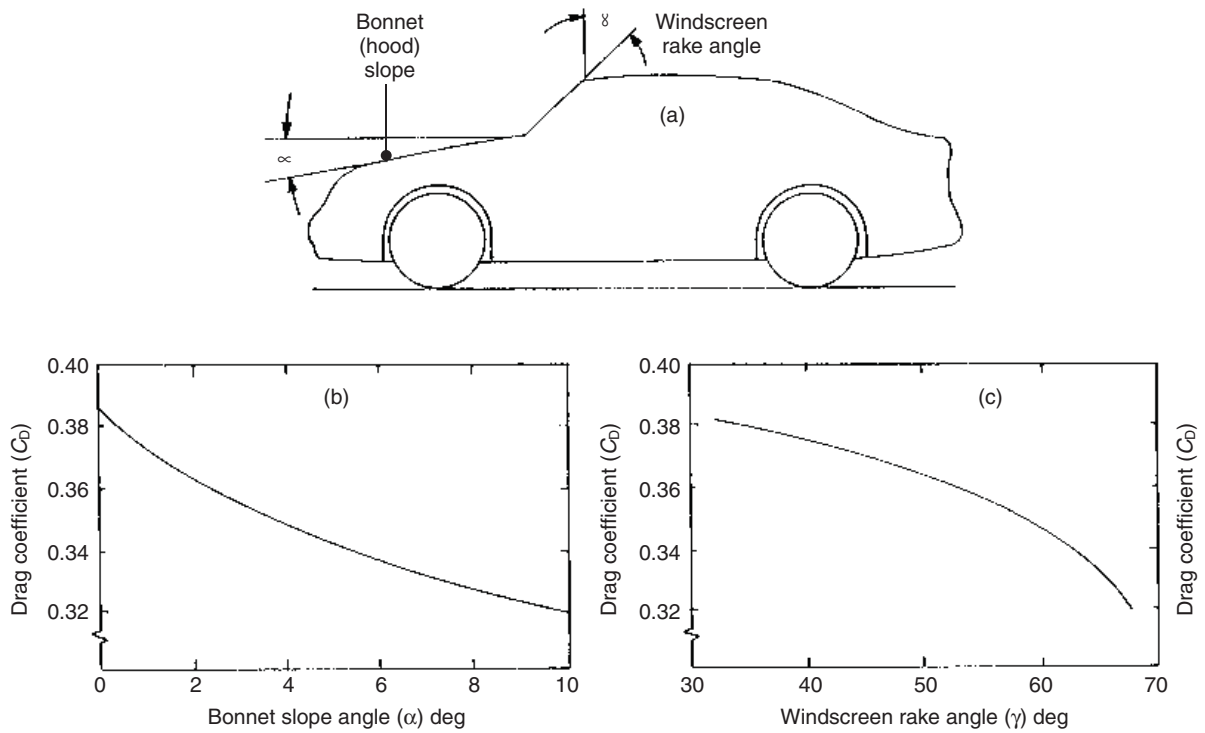
Fig. 14.28 (a–c)   Bonnet slope and windscreen rake angle versus drag coefficient
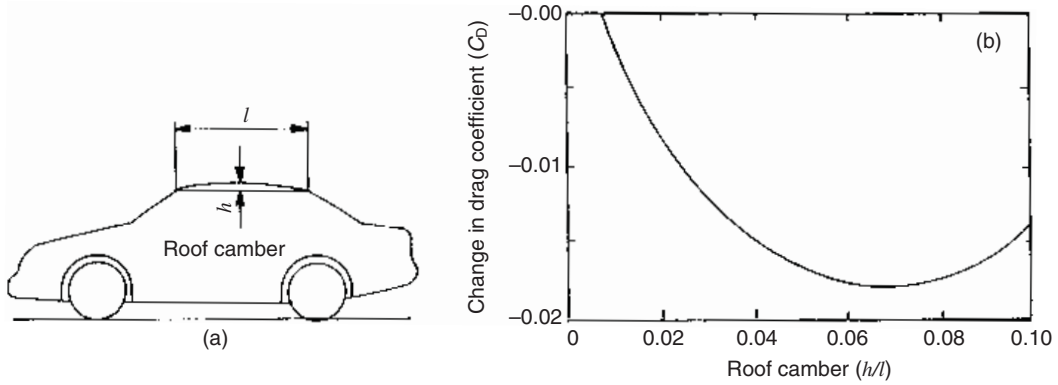
Fig. 14.29 (a and b) Effect of roof camber on drag coefficient
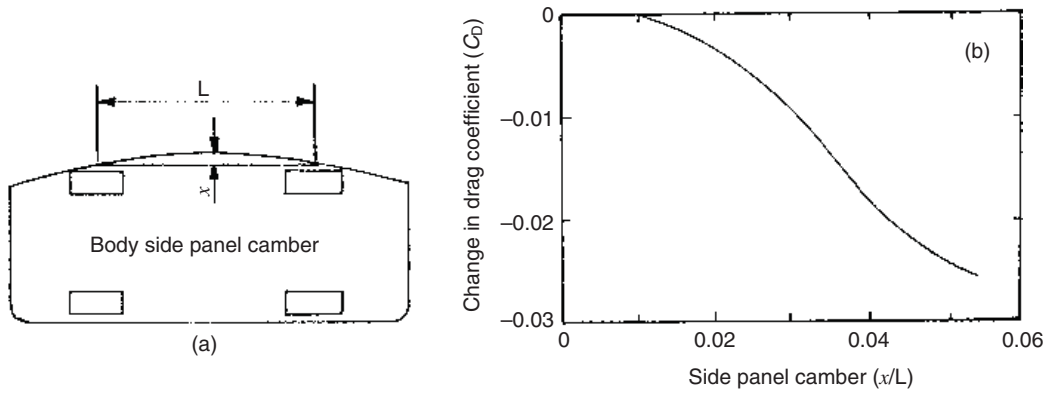
Fig. 14.30 (a and b) Effect of side panel camber on drag coefficient
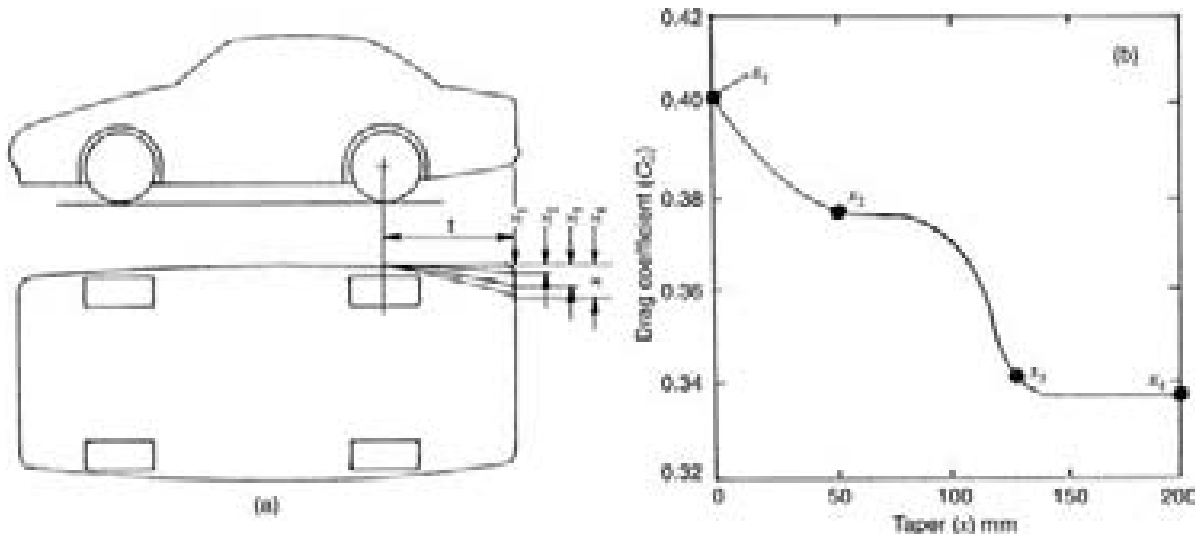
Fig. 14.31 (a and b) Effect of rear side panel taper on drag coefficient

605
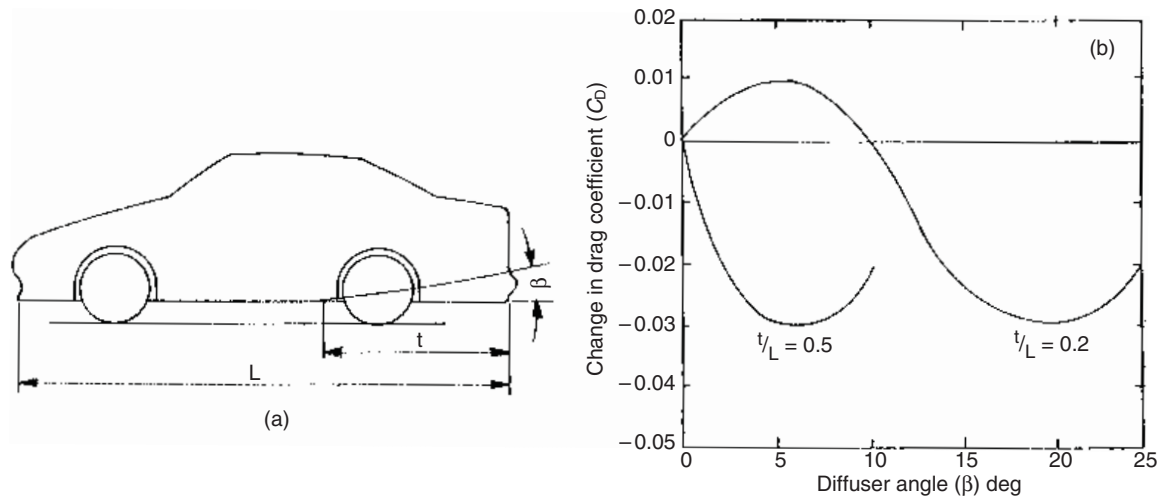
Fig. 14.32 (a and b) Effect of rear end upward taper on drag coefficient

components: front and rear wheel and suspension arch cavities, engine, transmission and steering compartment, side and cross member channelling, floor pan straightening ribs, jacking point straightening channel sections, structural central tunnel and rear wheel drive propeller shaft, exhaust system catalytic converter, silencer and piping, hand brake cable, and spare wheel compartment etc. A rough underbody produces excessive turbulence and friction losses and consequently raises the drag coefficient, whereas trapped air in the underside region slows down the air flow and tends to raise the underfloor pressure and therefore positive lift force. Vehicles with high drag coefficients gain least by smoothing the underside. The underfloor roughness or depth of irregularity defined as the centre line average peak to valley height for an average car is around +150 mm. A predictable relationship between the centre line average roughness and the drag coefficient for a given ground clearance and vehicle length is shown in Fig. 14.34(a and b).

14.5 Aerodynamic lift control

14.5.1 Underbody dams (Fig. 14.35(a–c))
Damming the underbody to ground clearance at the extreme rear blocks the underfloor airstream and causes a partial pressure build-up in this region, see Fig. 14.25(a), whereas locating the underbody dam in the front end of the car joins the rear low pressure wake region with the underfloor space, see Fig. 14.35(b). Thus with a rear end

underfloor air dam the underfloor air flow pressure increase raises the aerodynamic upthrust, that is, it produces positive lift, see Fig. 14.35(a). Conversely a front end air dam reduces the underfloor air flow pressure thereby generating an aerodynamic downthrust, that is, it produces negative lift (see Fig. 14.35(b)). Experimental results show with a front end dam there is a decrease in front lift (negative lift) whereas there is a slight rise in rear end lift (positive lift) as the dam height is increased, and as would be expected, there is also a rise in drag as the frontal area of the dam is enlarged, see Fig. 14.35(c).

14.5.2 Exposed wheel air flow pattern
(Fig. 14.36(a–c))
When a wheel rotates some distance from the ground air due to its viscosity attaches itself to the tread and in turn induces some of the surrounding air to be dragged around with it. Thus this concentric movement of air establishes in effect a weak vortex, see Fig. 14.36(a). If the rotating wheel is in contact with the ground it will roll forwards which makes windtunnel testing under these conditions difficult; this problem is overcome by using a supportive wheel and floor rig. The wheel is slightly submerged in a well opening equal to the tyre width and contact patch length for a normal loaded wheel and a steady flow of air is blown towards the frontal view of the wheel. With the wheel rig simulating a rotating wheel in contact with the ground, the wheel vortex air movement interacts and distorts the parallel main airstream.

606
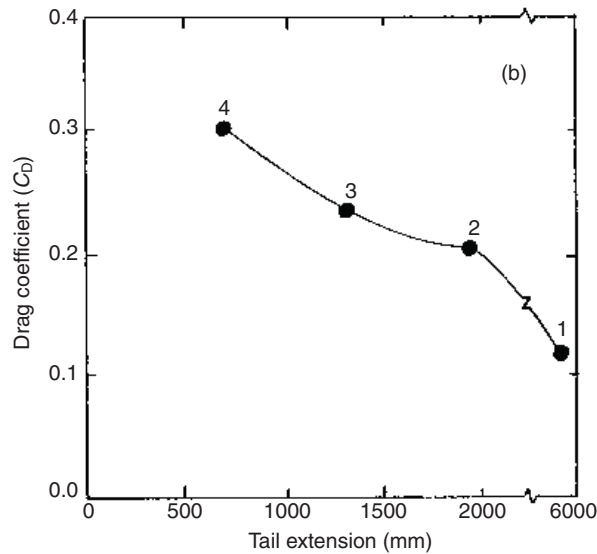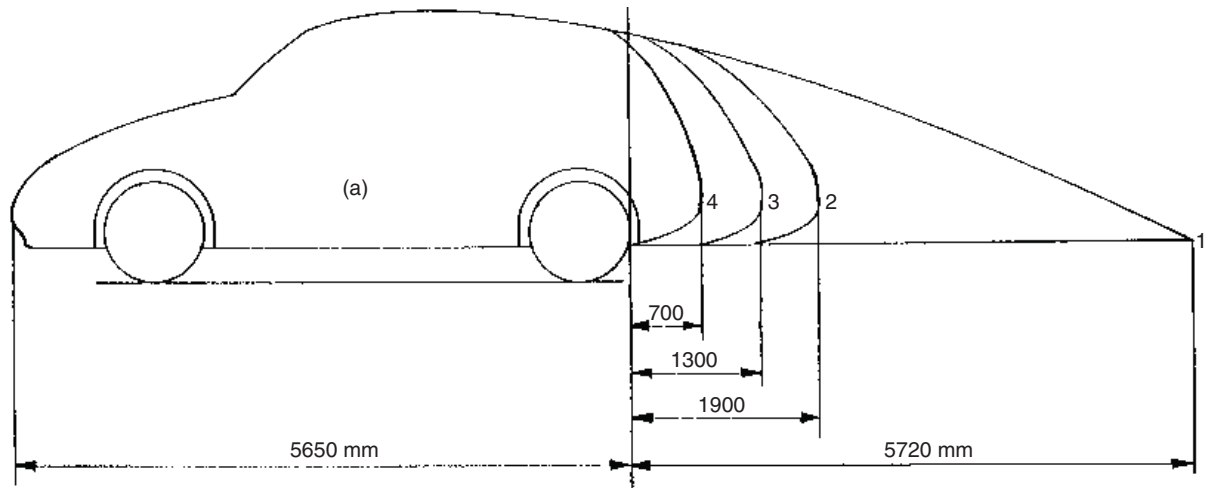
Fig. 14.33 (a and b) Effect of rear end tail extension on drag coefficient

The air flow pattern for an exposed wheel can be visualized and described in the following way. The air flow meeting the lower region of the wheel will be stagnant but the majority of the airstream will flow against the wheel rotation following the contour of the wheel until it reaches the top; it then separates from the vortex rim and continues to flow towards the rear but leaving underneath and in the wake of the wheel a series of turbulent vortices, see Fig. 14.36(b). The actual point of separation will creep forwards with increased rotational wheel speed. Air pressure distribution around the wheel will show a positive pressure build-up in the stagnant air flow front region of the wheel, but this changes rapidly to a high negative pressure where the main air flow breaks away from the wheel rim, see Fig. 14.36(c). It then declines to some extent beyond the highest point of the wheel, and then remains approximately constant around the rear wake region of the wheel. Under these described conditions, the exposed rotating wheel produces a resultant positive upward lift force which tends to reduce the adhesion between the tyre tread and ground.

607
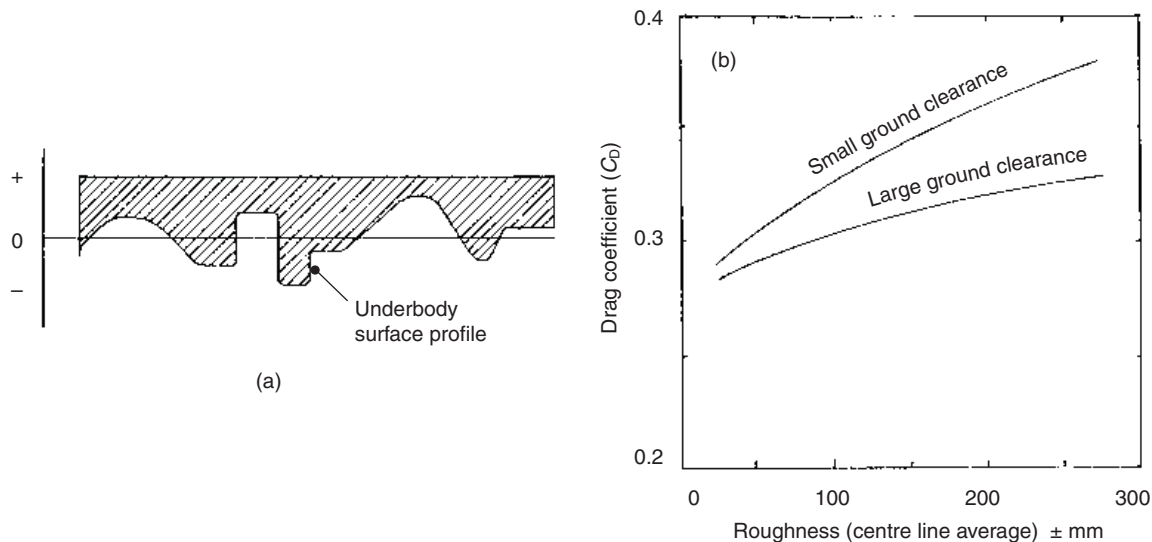
Fig. 14.34 (a and b)   Effect of underbody roughness on drag coefficient

(a) Rear end underbody air dam

(b) Front end underbody air dam
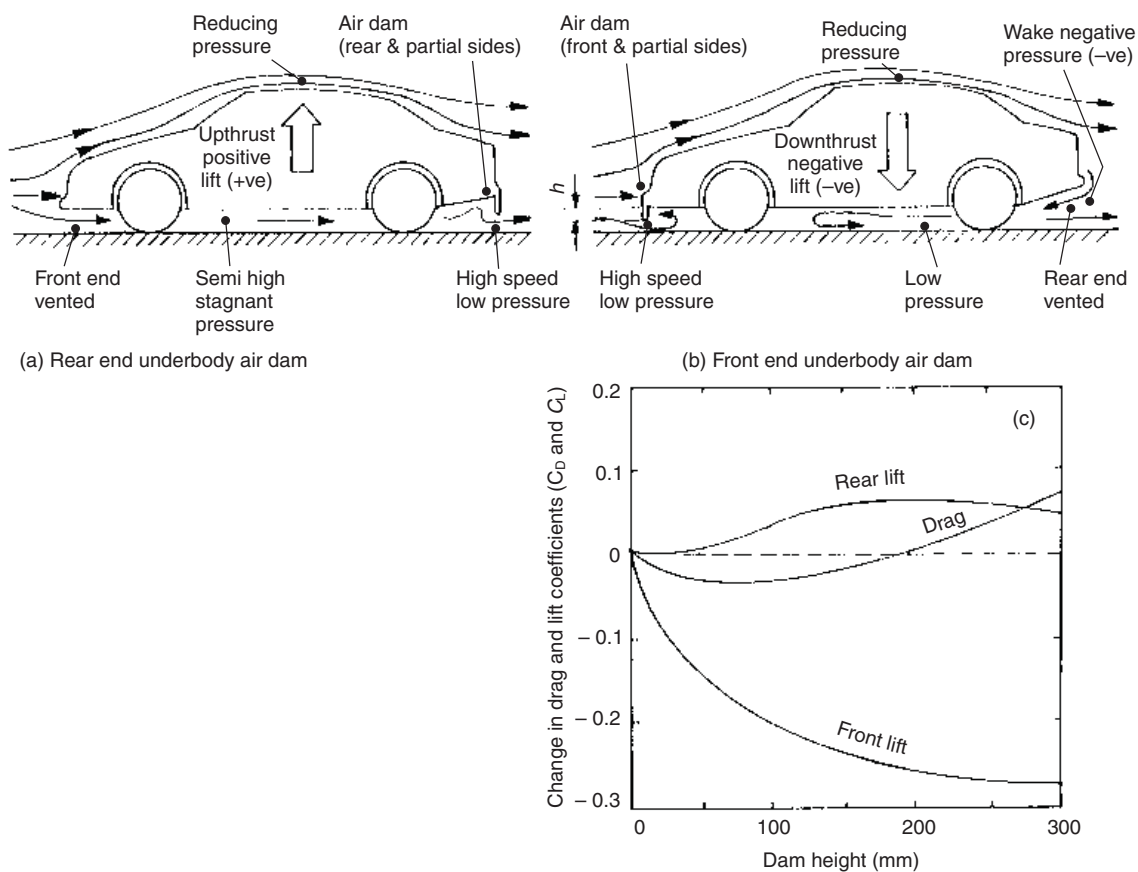
Fig. 14.35 (a–c)   Effects of underbody front and rear end air dams relative to the lift and drag coefficient

Concentric
streamlines

Weak
vortex

ω

(a) Wheel rotation in still air away from the ground

Velocity
gradient

Separation
vortices

Positive
lift (+ve)

Point of
separation

ω θ

Airstream

Direction of
wheel movement

Low pressure
region

High pressure
region

(b) Air flow pattern with wheel rolling on the ground

Resultant
upward
lift (+ve)

Negative
pressure (–ve)

Pressure
distribution

Direction
of motion

ω

Positive
pressure (+ve)

(c) Air pressure distribution with wheel rolling on the ground

Fig. 14.36 (a–c)   Exposed wheel air flow pattern and pressure distribution

14.5.3 *Partial enclosed wheel air flow pattern*
(Figs 14.37(a and b) and 14.38(a–c))
The air flow passing beneath the front of the
car initially moves faster than the main airstream,
this therefore causes a reduction in the local air
pressure. At the rear of the rotating wheel due to
viscous drag air will be scooped into the upper
space formed between the wheel tyre and the
wheel mudguard arch (see Fig. 14.37(a and b)). The

air entrapped in the wheel arch cavity circulates
towards the upper front of the wheel due to a
slight pressure build-up and is then expelled
through the front end wheel to the mudguard gap
which is at a lower pressure in both a downward
and sideward direction. Decreasing the clearance
between the underside and the ground and shield-
ing more of the wheel with the mudguard tends to
produce a loss of momentum to the air so that both

609
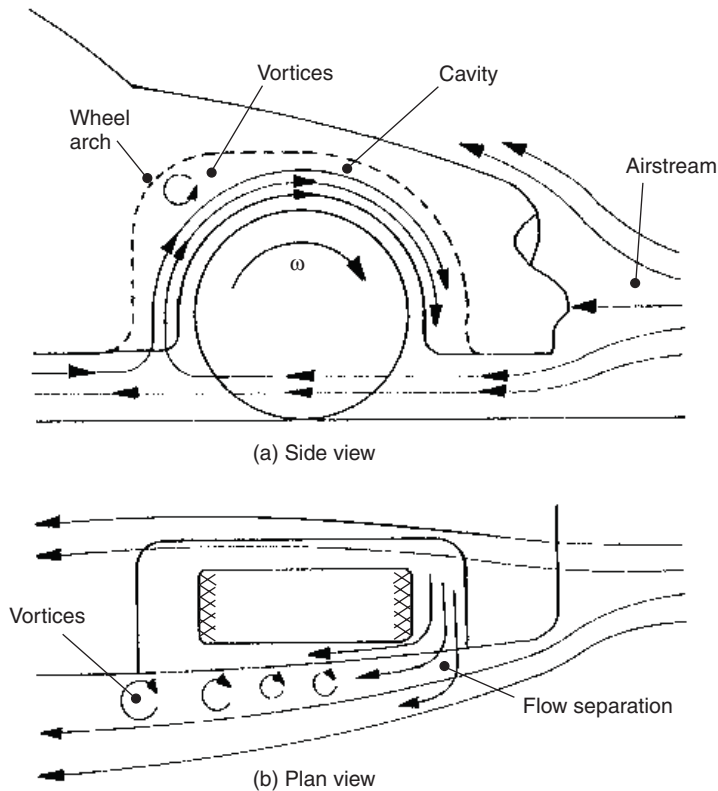
Fig. 14.37 (a and b) Wheel arch air flow pattern

aerodynamic lift C_{LW} and drag C_{DW} coefficients, and therefore forces, are considerably reduced Fig. 14.38(a–c).

14.5.4 Rear end spoiler (Fig. 14.39(a–c))
Generally when there is a gentle rear end body profile curvature change, it will be accompanied with a relatively fast but smooth streamline air flow over this region which does not separate from the upper surface. However, this results in lower local pressures which tend to exert a lift force (upward suction) at the rear end of the car. A lip, see Fig. 14.39(a), or small aerofoil spoiler, see Fig. 14.39(b), attached to the rear end of the car boot (trunk) interrupts the smooth streamline air flow thereby slowing down the air flow and correspondingly raising the upper surface local air pressure which effectively increase the downward force known as negative lift. A typical relationship between rear lift, front lift and drag coefficients relative to the spoiler lip height is shown in Fig. 14.39(c). The graph shows a general increase in negative lift (downward force) by increasing the spoiler lip height. However, this is at the expense of a slight rise in the front end lift coefficient, whereas the drag coefficient initially decreases and then marginally rises again with increased spoiler lip height. It should be appreciated that the break-up of the smooth streamline air flow and the increase in rear downward pressure should if possible be achieved without incurring too much, if any, increase in front end positive lift and aerodynamic drag.

14.5.5 Negative lift aerofoil wings
(Fig. 14.40(a–c))
A negative lift wing is designed when attached to the rear end of the car to produce a downward thrust thereby enabling the traction generated by the rear driving wheels to be increased, or if a forward negative lift wing is fitted to improve the grip of both front steering wheels.

With the negative lift wing the aerofoil profile is tilted downward towards the front end with the negative and positive aerofoil section camber at the top and bottom respectively, see Fig. 14.40(a). The airstream therefore moving underneath the

610

(a) Side view   (b) Front view
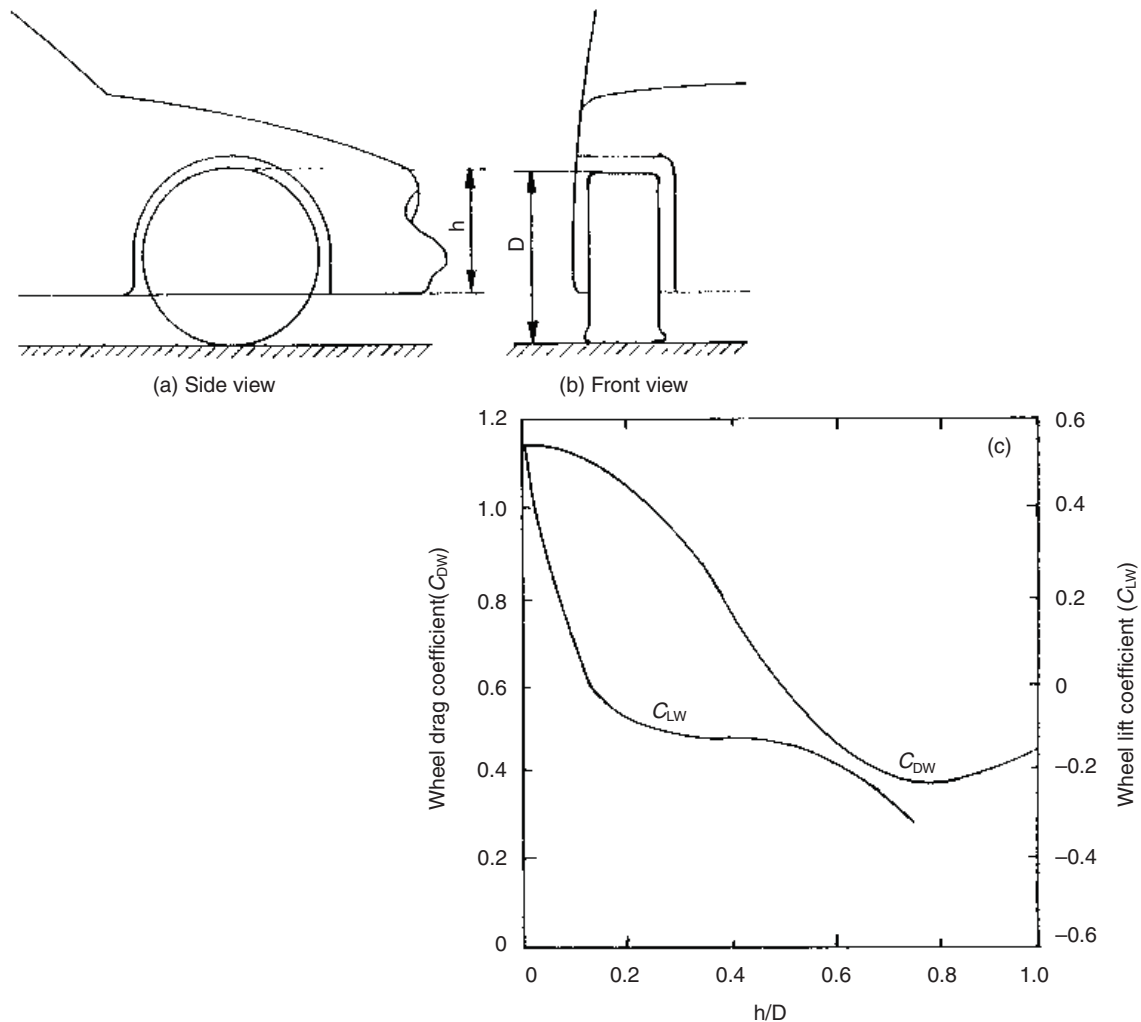
Fig. 14.38 (a–c) Effect of underside ground clearance on both lift and drag coefficients

aerofoil wing has to move further and faster than the airstream flowing over the upper surface; the pressure produced below the aerofoil wing is therefore lower than above. Consequently there will be a resultant downthrust perpendicular to the cord of the aerofoil (see Fig. 14.40(b)) which can be resolved into both a vertical downforce (negative lift) and a horizontal drag force. Enlarging the tilt angle of the wing promotes more negative lift (downthrust) but this is at the expense of increasing the drag force opposing the forward movement of the wing, thus a compromise must always be made between improving the downward wheel grip and the extra drag force opposing the motion of the car. Racing cars have the aerofoil wing over the rear wheel axles or just in front or behind them, see Fig.

14.40(c). However, the drag force produces a clockwise tilt which tends to lift the front wheels of the ground, therefore the front aerofoil wings (see Fig. 14.40(c)) are sometimes attached low down and slightly ahead of the front wheels to counteract the front end lift tendency.

14.6 Afterbody drag

14.6.1 Squareback drag (Figs 14.41 and 14.42)
Any car with a rear end (base) slope surface angle ranging from 90° to 50° is generally described as a squareback style (see Fig. 14.42). Between this angular surface inclination range for a squareback car there is very little change in the air flow pattern

611
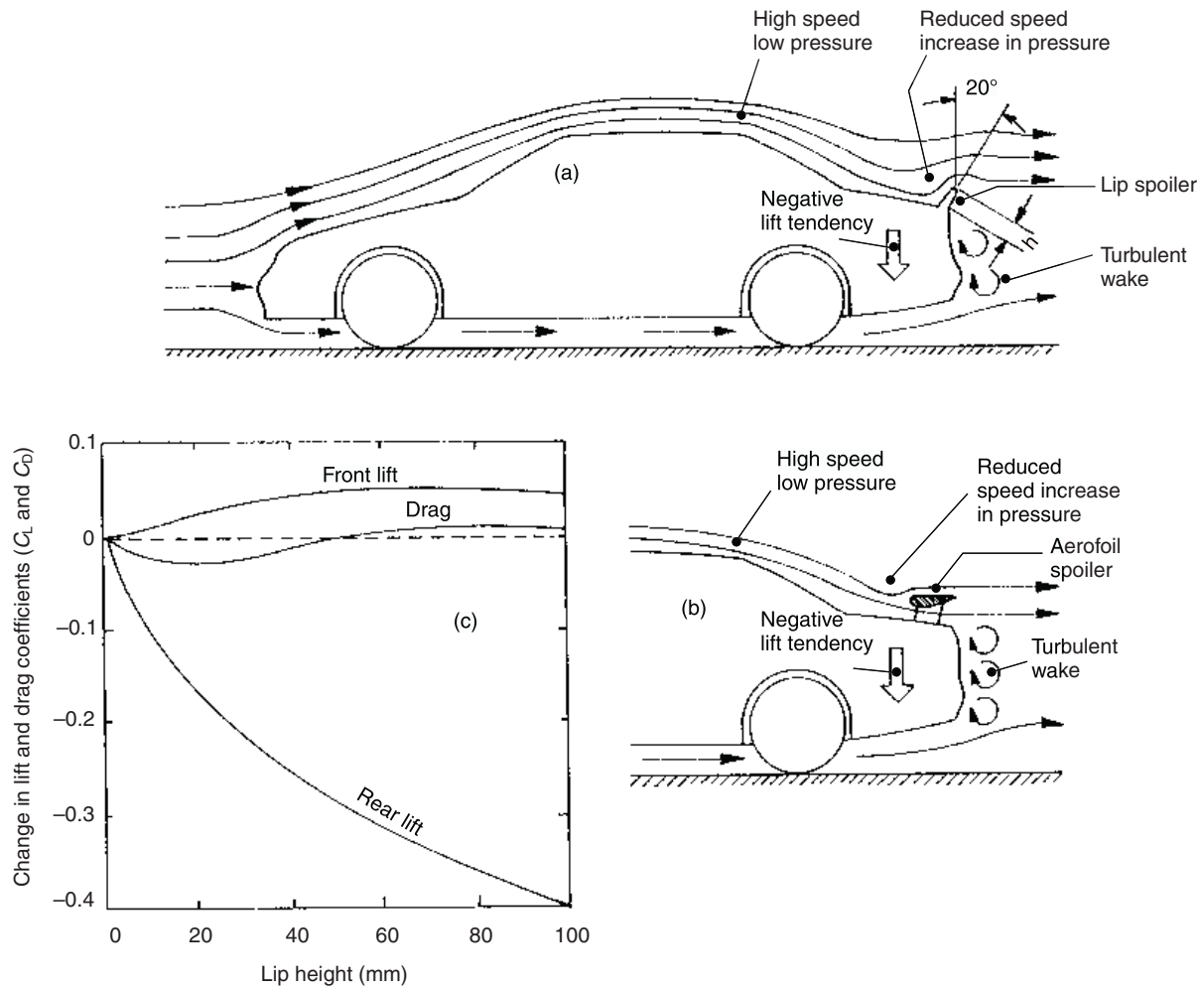
Fig. 14.39 (a–c) Effect of rear end spoiler on both lift and drag coefficients

and therefore there is virtually no variation in the afterbody drag (see Fig. 14.41). With a parallel sided squareback rear end configuration, the whole rear surface area (base area) becomes an almost constant low negative pressure wake region. Tapering the rear quarter side and roof of the body and rounding the rear end tends to lower the base pressure. In addition to the base drag, the afterbody drag will also include the negative drag due to the surrounding inclined surfaces.

14.6.2 Fastback drag (Figs 14.41 and 14.43)
When the rear slope angle is reduced to 25° or less the body profile style is known as a fastback, see Fig. 14.43. Within this much reduced rear end inclination the airstream flows over the roof and rear downward sloping surface, the airstream remain-

ing attached to the body from the rear of the roof to the rear vertical light-plate and at the same time the condition which helps to generate attached and trailing vortices with the large sloping rear end is no longer there. Consequently the only rearward suction comes from the vertical rear end projected base area wake, thus as the rear end inclined angle diminishes, the drag coefficient decreases, see Fig. 14.41. However, as the angle approaches zero there is a slight rise in the drag coefficient again as the rear body profile virtually reverts to a squareback style car.

14.6.3 Hatchback drag (Figs 14.41, 14.44 and 14.45)
Cars with a rear sloping surface angle ranging from 50° to 25° are normally referred to as hatchback

612
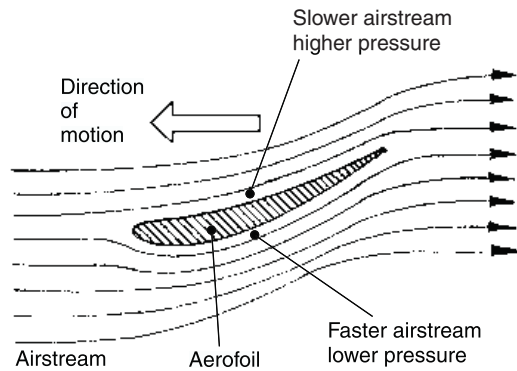
(a) Air streamlining for an inclined
negative lift aerofoil wing
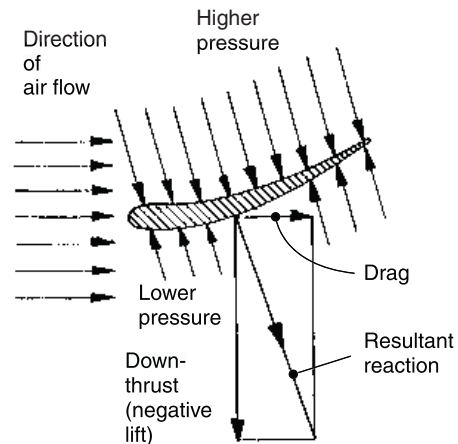
(b) Lift and drag components on
an inclined negative lift wing

$$W = \frac{F_D\,h}{L}$$
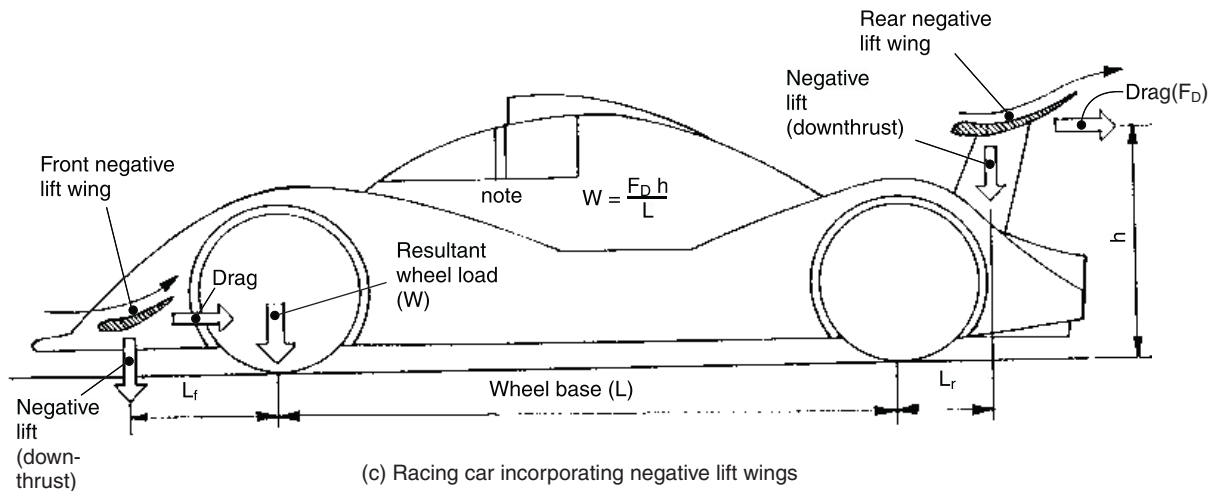
(c) Racing car incorporating negative lift wings

Fig. 14.40 (a–c)   Negative lift aerofoil wing considerations

style, see Fig. 14.44. Within this rear end inclination range air flows over the rear edge of the roof and commences to follow the contour of the rear inclined surface; however, due to the steepness of the slope the air flow breaks away from the surface. At the same time some of the air flows from the higher pressure underfloor region to the lower pressure roof and rear sloping surface, then moves slightly inboard and rearward along the upper downward sloping surface. The intensity and direction of this air movement along both sides of the rear upper body edging causes the air to spiral into a pair of trailing vortices which are then pushed downward by the downwash of the airstream

flowing over the rear edge of the roof, see Fig. 14.45. Subsequently these vortices re-attach themselves on each side of the body, and due to the air's momentum these vortices extend and trail well beyond the rear of the car. Hence not only does the rear negative wake base area include the vertical area and part of the rearward projected slope area where the airstream separates from the body profile, but it also includes the trailing conical vortices which also apply a strong suction pull against the forward motion of the car. As can be seen in Fig. 14.41 there is a critical slope angle range (20 to 35°) in which the drag coefficient rises steeply and should be avoided.

613
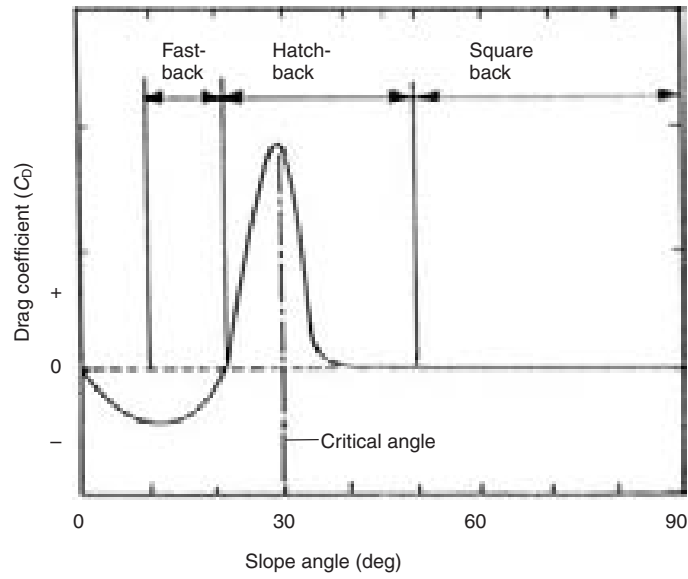
Fig. 14.41 Effect of rear panel slope angle on the afterbody drag
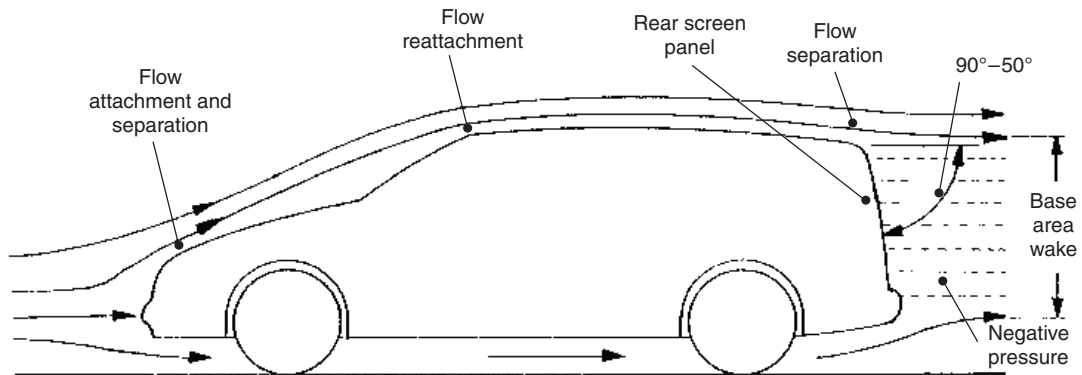
Fig. 14.42 Squareback configuration
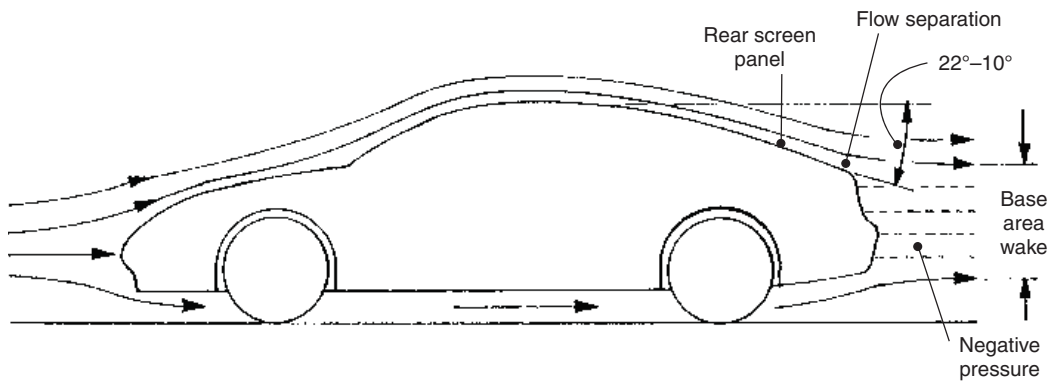
Fig. 14.43 Fastback configuration

614
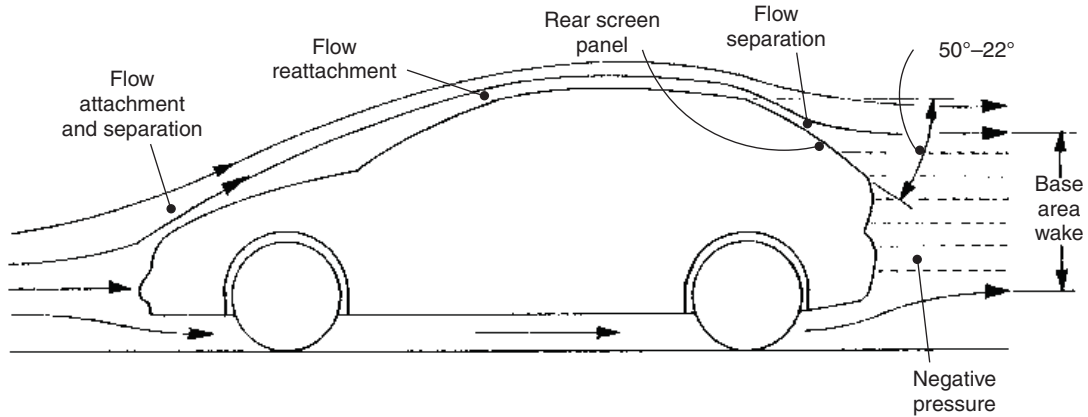
Fig. 14.44 Hatchback configuration
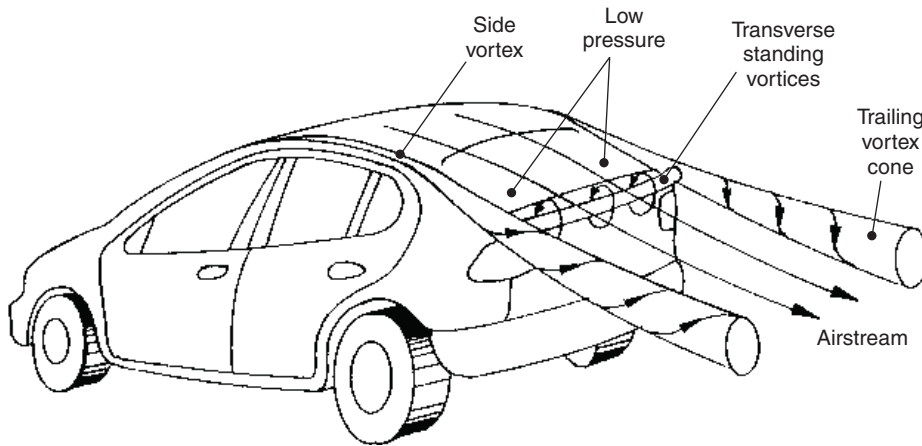
Fig. 14.45 Hatchback transverse and trailing vortices

14.6.4 Notchback drag (Figs 14.46, 14.47(a and b) and 14.48(a and b))

A notchback car style has a stepped rear end body profile in which the passenger compartment rear window is inclined downward to meet the horizontal rearward extending boot (trunk) lid (see Fig. 14.46). With this design, the air flows over the rear roof edge and follows the contour of the downward sloping rear screen for a short distance before separating from it; however, the downwash of the airstream causes it to re-attach itself to the body near the rear end extended boot lid. Thus the base-wake area will virtually be the vertical rear boot and light panel; however, standing vortices will be generated on each side of the body just inboard on the top surface of the rear window screen and boot lid, and will then be projected in the form of trailing conical vortices well beyond the rear end of the

boot, see Fig. 14.19(b). Vortices will also be created along transverse rear screen to boot lid junction and across the rear of the panel light.

Experiments have shown (see Fig. 14.47(a)) that the angle made between the horizontal and the inclined line touching both the rear edges of the roof and the boot is an important factor in determining the afterbody drag. Fig. 14.47(b) illustrates the effect of the roof to boot line inclination; when this angle is increased from the horizontal the drag coefficient commences to rise until reaching a peak at an inclination of roughly 25°, after which the drag coefficient begins to decrease. From this it can be seen that raising the boot height or extending the boot length decreases the effective inclination angle Φ_e and therefore tends to reduce the drag coefficient. Conversely a very large effective inclination angle Φ_e will also cause a reduction in the

615
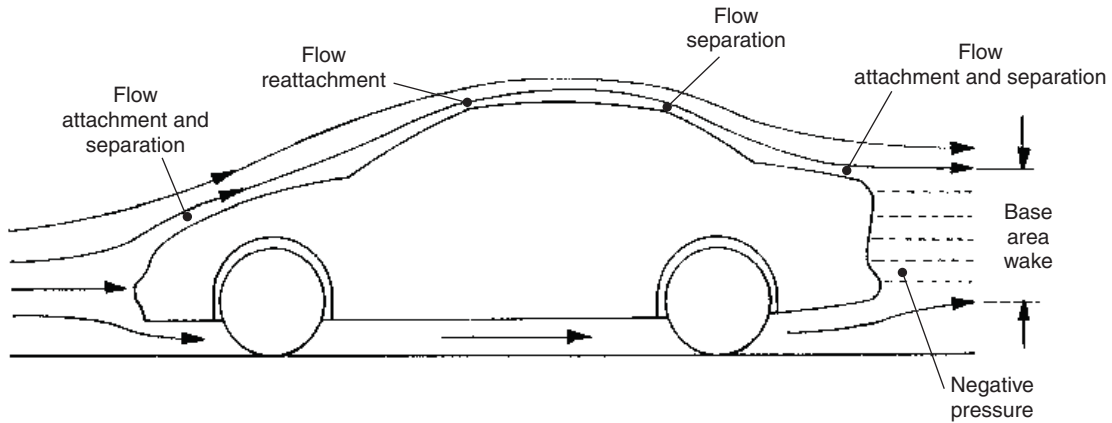
Fig. 14.46 Notchback configuration
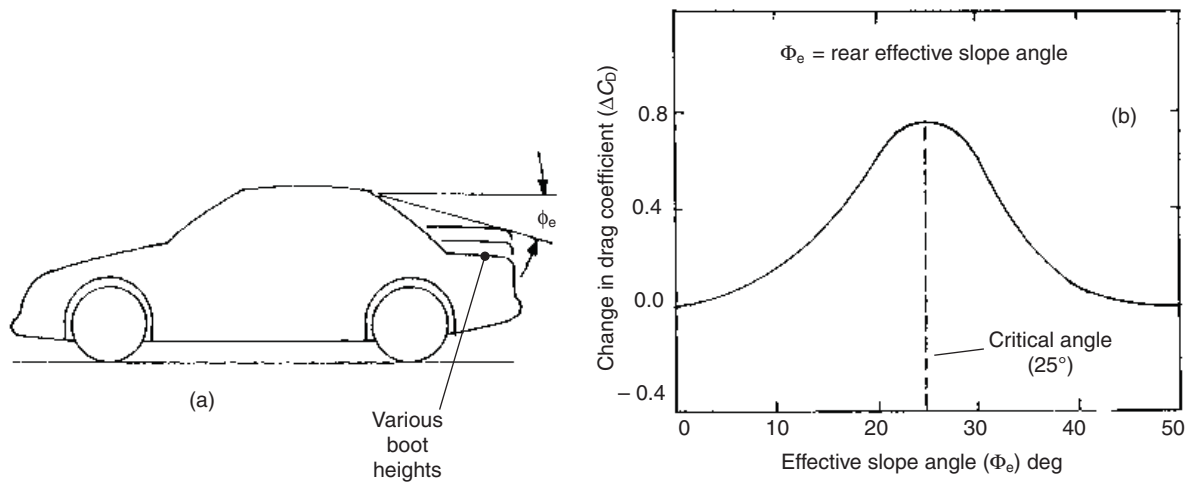
Fig. 14.47 (a and b) Influence of the effective slope angle on the drag coefficient

drag coefficient but at the expense of reducing the volume capacity of the boot. The drag coefficient relative to the rear boot profile can be clearly illustrated in a slightly different way, see Fig. 14.48(a). Here windtunnel tests show how the drag coefficient can be varied by altering the rear end profile from a downward sloping boot to a horizontal boot and then to a squareback estate shape. It will be observed (see Fig 14.48(b)) that there is a critical increase in boot height in this case from 50 to 150 mm when the drag coefficient rapidly decreases from 0.42 to 0.37.

14.6.5 Cabriolet cars (Fig. 14.49)
A cabriolet is a French noun and originally referred to a light two wheeled carriage drawn by one horse. Cabriolet these days describes a car with a folding roof such as a sports (two or four seater) or roadster (two seater) car. These cars may be driven with the folding roof enclosing the cockpit or with the soft roof lowered and the side screen windows up or down. Streamlining is such that the air flow follows closely to the contour of the nose and bonnet (hood), then moves up the windscreen before overshooting the screen's upper horizontal edge (see Fig. 14.49). If the rake angle of the windscreen is small (such as with a high mounted off road four wheel drive vehicle) the airstream will be deflected upward and rearward, but with a large rake angle windscreen the airstream will not rise much above the windscreen upper leading edge as the air flows over the open driver/passenger
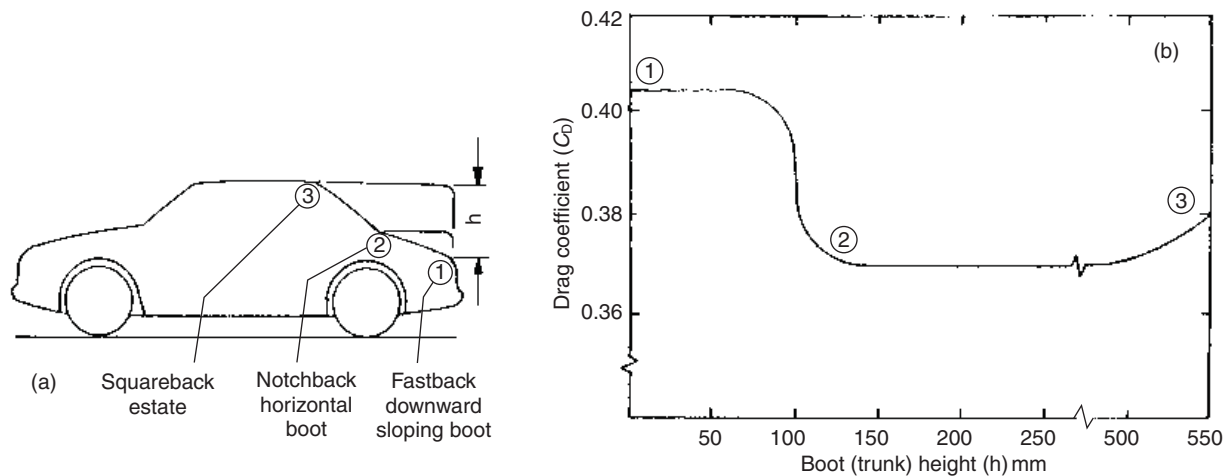
Fig. 14.48 (a and b) Effect of elevating the boot (trunk) height on the drag coefficient

compartment towards the rear of the car. A separation bubble forms between the airstream and the exposed and open seating compartment, the downstream air flow then re-attaches itself to the upper face of the boot (trunk). However, this bubble is unstable and tends to expand and burst in a cyclic fashion by the repetition of separation and re-attachment of the airstream on top of the boot (trunk), see Fig. 14.49. Thus the turbulent energy causes the bubble to expand and collapse and the fluctuating wake area (see Fig. 14.49), changing between h_1 and h_2 produces a relatively large drag resistance. With the side windscreens open air is drawn into the low pressure bubble region and in the process strong vortices are generated at the side entry to the seating compartment; this also therefore contributes to the car drag resistance. Typical drag coefficients for an open cabriolet car are given

as follows: folding roof raised and side screens up C_D 0.35, folding roof down and side screens up C_D 0.38, and folding roof and side screens down C_D 0.41. Reductions in the drag coefficient can be made by attaching a header rail deflector, streamlining the roll over bar and by neatly storing or covering the folding roof, the most effective device to reduce drag being the header rail deflector.

14.7 Commercial vehicle aerodynamic fundamentals

14.7.1 The effects of rounding sharp front cab body edges (Fig. 14.50(a–d))
A reduction in the drag coefficient of large vehicles such as buses, coaches and trucks can be made by rounding the front leading edges of the vehicle.

Fig. 14.49 Open cabriolet

617

Over flow

Flow separation

Side flow

$C_D = 0.88$
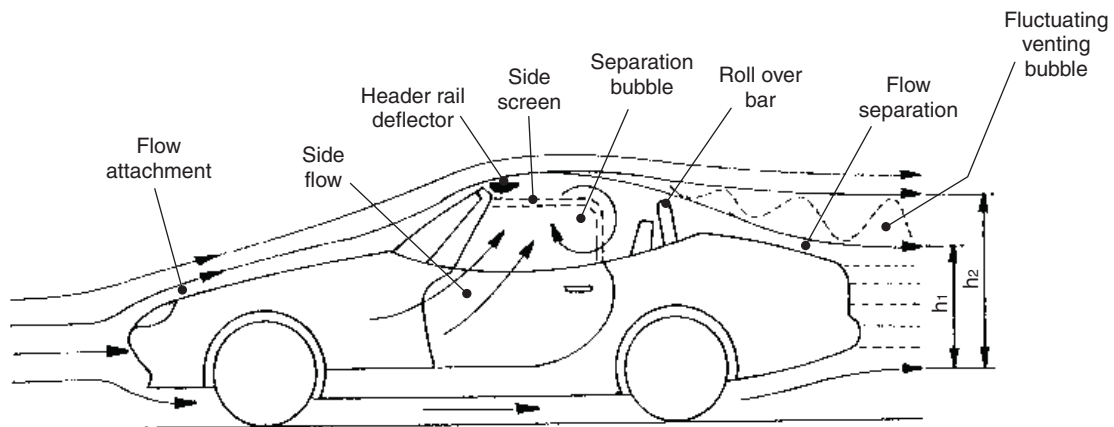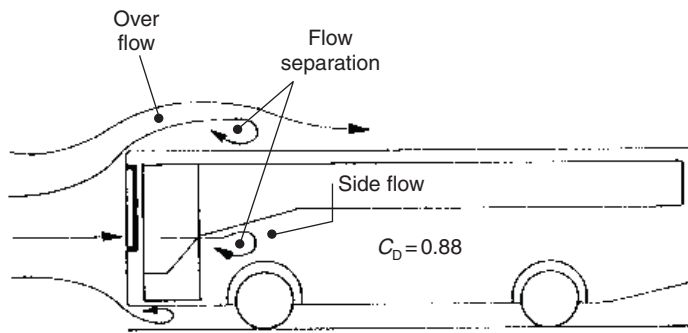
(a) Coach with sharp leading edges

Flow almost remains attached

$C_D = 0.36$
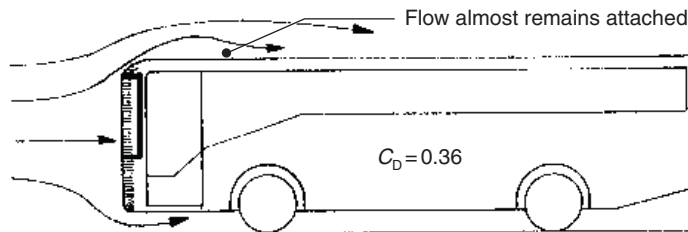
(b) Coach with rounded leading edges

Flow remains attached

$C_D = 0.34$
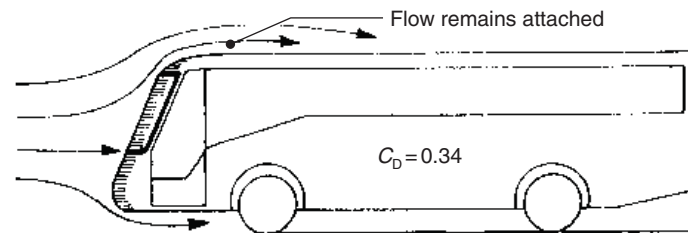
(c) Coach with rounded edges and backsloping front

(d) Effect of rounding vehicle leading edges upon the aerodynamic drag

Change in drag coefficient (ΔC_D)

R

Leading edge radius (R) mm
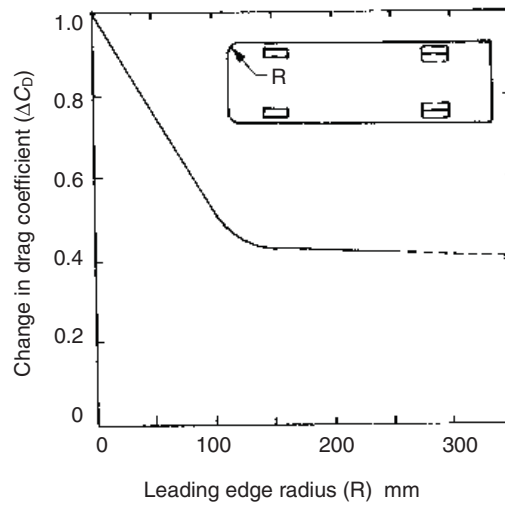
Fig. 14.50 (a–d) Forebody coach streamlining

Simulated investigations have shown a marked decrease in the drag coefficient from having sharp forebody edges (see Fig. 14.50(a)) to relatively large round leading edge radii, see Fig. 14.50(b). It can be seen from Fig. 14.50(d) that the drag coefficient progressively decreased as the round edge radius was increased to about 120 mm, but there was only a very small reduction in the drag coefficient with further increase in radii. Thus there is an optimum radius for the leading front edges, beyond this there is no advantage in increasing the rounding radius. The reduction in the drag coefficient due to rounding the edges is caused mainly by the change from flow separation to attached streamline flow for both cab roof and side panels, see Fig. 14.50(a and b). However, sloping back the front profile of the coach to provide further streamlining only made a marginal reduction in the drag coefficient, see Fig. 14.50(c).

14.7.2 The effects of different cab to trailer body heights with both sharp and rounded upper windscreen leading edges (Fig. 14.51(a–c))

A generalized understanding of the air flow over the upper surface of an articulated cab and trailer can be obtained by studying Fig. 14.51(a and b). Three different trailer heights are shown relative to one cab height for both a sharp upper windscreen leading edge (Fig. 14.51(a)) and for a rounded upper windscreen edge (Fig. 14.51(b)). It can be seen in the case of the sharp upper windscreen leading edge cab examples (Fig. 14.51(a)) that with the low trailer body the air flow cannot follow the contour of the cab and therefore overshoots both the cab roof and the front region of the trailer body roof thereby producing a relatively high coefficient of drag, see Fig. 14.51(c). With the medium height trailer body the air flow still overshoots (separates) the cab but tends to align and attach itself early to the trailer body roof thereby producing a relatively low coefficient of drag, see Fig. 14.51(c). However, with the high body the air flow again overshoots the cab roof; some of the air then hits the front of the trailer body, but the vast majority deflects off the trailer body leading edge before re-attaching itself further along the trailer body roof. Consequently the disrupted air flow produces a rise in the drag coefficient, see Fig. 14.51(c).

In the case of the rounded upper windscreen leading edge cab (see Fig. 14.51(b)), with a low trailer body the air flowing over the front windscreen remains attached to the cab roof, a small proportion will hit the front end of the trailer body

then flow between the cab and trailer body, but the majority flows over the trailer roof leading edge and attaches itself only a short distance from the front edge of the trailer roof thereby producing a relatively low drag coefficient, see Fig. 14.51(c). With the medium height trailer body the air flow remains attached to the cab roof; some air flow again impinges on the front of the trailer body and is deflected between the cab and trailer body, but most of the air flow hits the trailer body leading edge and is deflected slightly upwards and only re-attaches itself to the upper surface some distance along the trailer roof. This combination therefore produces a moderate rise in the drag coefficient, see Fig. 14.51(c). In the extreme case of having a very high trailer body the air flow over the cab still remains attached and air still flows downwards into the gap made between the cab and trailer; however, more air impinges onto the vertical front face of the trailer body and the deflection of the air flow over the leading edge of the trailer body is even steeper than in the case of the medium height trailer body. Thus re-attachment of the air flow over the roof of the trailer body takes place much further along its length so that a much larger roof area is exposed to air turbulence; consequently there is a relatively high drag coefficient, see Fig. 14.51(c).

### 14.7.3 Forebody pressure distribution (Fig. 14.52(a and b))

With both the conventional cab behind the engine and the cab over or in front of the engine tractor unit arrangements there will be a cab to trailer gap to enable the trailer to be articulated when the vehicle is being manoeuvred. The cab roof to trailer body step, if large, will compel some of the air flow to impinge on the exposed front face of the trailer thereby producing a high pressure stagnation region while the majority of air flow will be deflected upwards. As it brushes against the upper leading edge of the trailer the air flow then separates from the forward region of the trailer roof before re-attaching itself further along the flat roof surface, see Fig. 14.52(a). As can be seen the pressure distribution shows a positive pressure (above atmospheric pressure) region air spread over the exposed front face of the trailer body with its maximum intensity (stagnant region) just above the level of the roof; this contrasts the negative pressure (below atmospheric pressure) generated air flow in the forward region of the trailer roof caused by the air flow separation turbulence. Note the negative pressure drops off towards the rear of the roof due to air flow re-attachment.
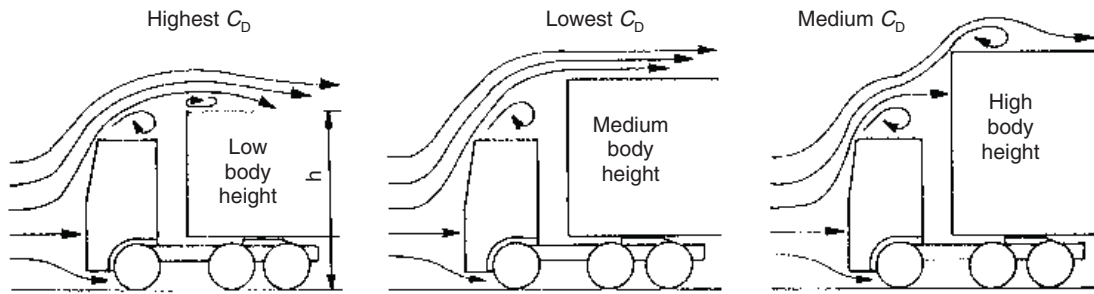
(a) Tractor cab with sharp windscreen/roof leading edge (flow separation over cab roof)
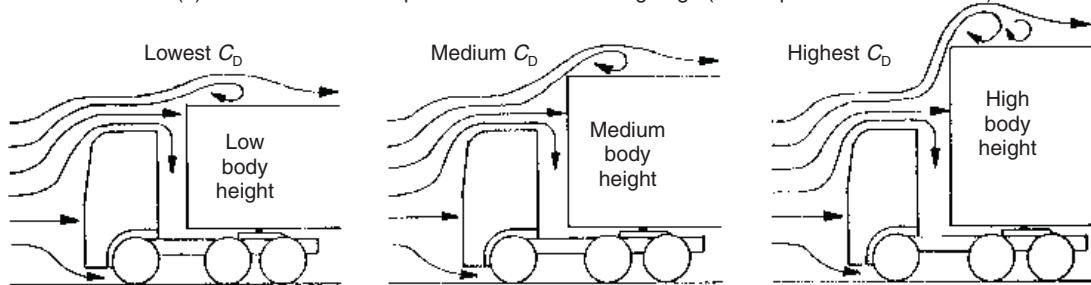
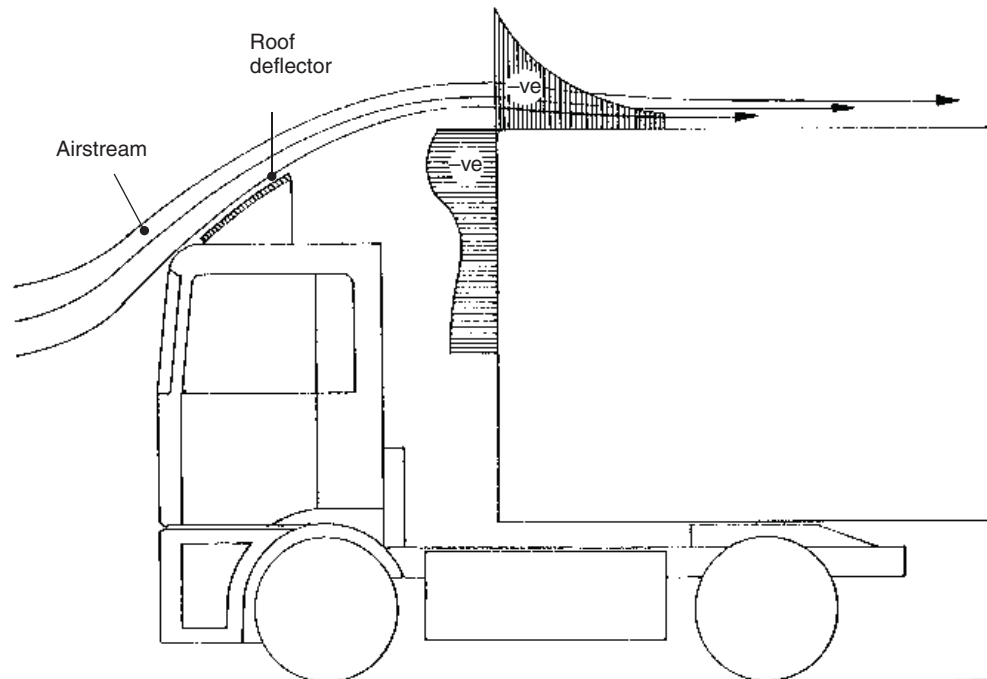(b) Tractor cab with rounded windscreen/roof leading edge (attached air flow over cab roof)
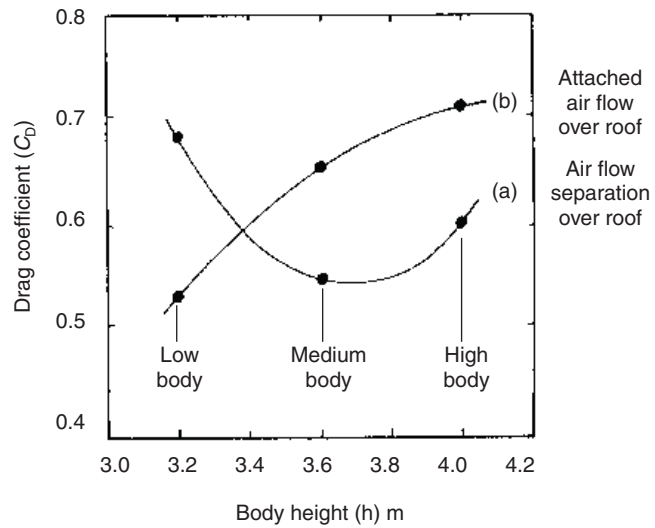
(c) Influence of cab to body height and cab shape
upon the drag coefficient

Fig. 14.51 (a–c) Comparison of air flow conditions with both sharp and rounded roof leading edge cab with various trailer body heights

620

Airstream

Trailer roof
pressure distribution

+ve

−ve

Trailer front
panel pressure
distribution
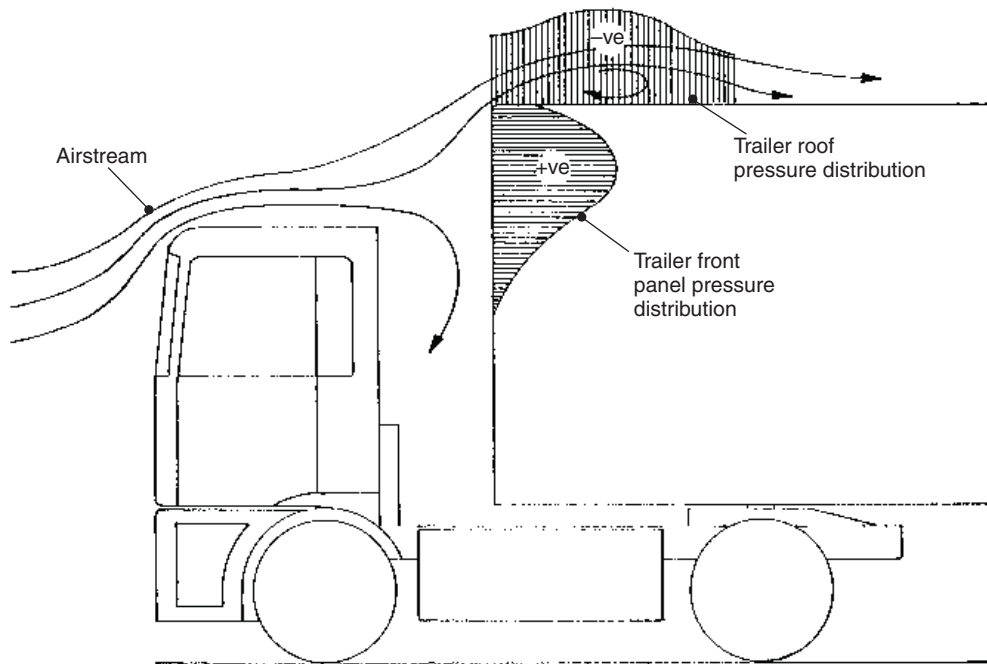
(a) Cab without roof deflector

Roof
deflector

Airstream

−ve

−ve

(b) Cab with roof deflector

Fig. 14.52 (a and b)　Trailer flow body pressure distribution with and without cab roof deflector

By fitting a cab roof deflector the pattern of air flow is diverted upwards and over the roof of the trailer body, there being only a slight degree of flow separation at the front end of the trailer body roof, see Fig. 14.52(b). Consequently the air flow moves directly between the cab roof deflector and the roof of the trailer body; it thus causes the air pressure in the cab to trailer gap to decrease, this negative pressure being more pronounced on the exposed upper vertical face of the trailer, hence the front face upper region of the trailer will actually reduce that portion of drag produced by the exposed frontal area of the trailer. Conversely the negative pressure created by the air flow over the leading edge of the roof falls rapidly, indicating early air flow re-attachment.

14.7.4 The effects of a cab to trailer body roof height step (Fig. 14.53(a and b))

Possibly the most important factor which contributes to a vehicle's drag resistance is the exposed area of the trailer body above the cab roof relative to the cab's frontal area (Fig. 14.53(a)). Investigation into the forebody drag of a truck in a windtunnel has been made where the trailer height is varied relative to a fixed cab height. The drag coefficient for different trailer body to cab height ratios (t/c) were then plotted as shown in Fig. 14.53(b). For this particular cab to trailer combination dimensions there was no noticeable change in the drag coefficient C of 0.63 with an increase in trailer body to cab height ratio until about 1.2, after which the drag coefficient commenced to rise in proportion to

the increase in the trailer body to cab height ratio up to a t/c ratio of 1.5, which is equivalent to the maximum body height of 4.2 m; this corresponded to a maximum drag coefficient of 0.86. Hence increasing the trailer body step height ratio from 1.2 to 1.5 increases the step height from 0.56 m to 1.4 m and in turn raises the drag coefficient from 0.63 to 0.86. The rise in drag coefficient of 0.23 is considerable and therefore streamlining the air flow between the cab and trailer body roof is of great importance.

14.8 Commercial vehicle drag reducing devices

14.8.1 Cab roof deflectors (Figs 14.54(a and b), 14.55(a and b) and 14.56(a–c))

To partially overcome the large amount of extra drag experienced with a cab to trailer height mismatch a cab roof deflector is commonly used. This device prevents the air movement over the cab roof impinging on the upper front of the trailer body and then flowing between the cab and trailer gap, see Fig. 14.54(a). Instead the air flow is diverted by the uptilted deflector surface to pass directly between the cab to trailer gap and then to flow relatively smoothly over the surface of the trailer roof, see Fig. 14.54(b). These cab roof deflectors are beneficial in reducing the head on air flow but they do not perform so well when subjected to side winds. Slight improvements can be made to prevent air flowing underneath and across the deflector plate by enclosing the sides; this is usually achieved
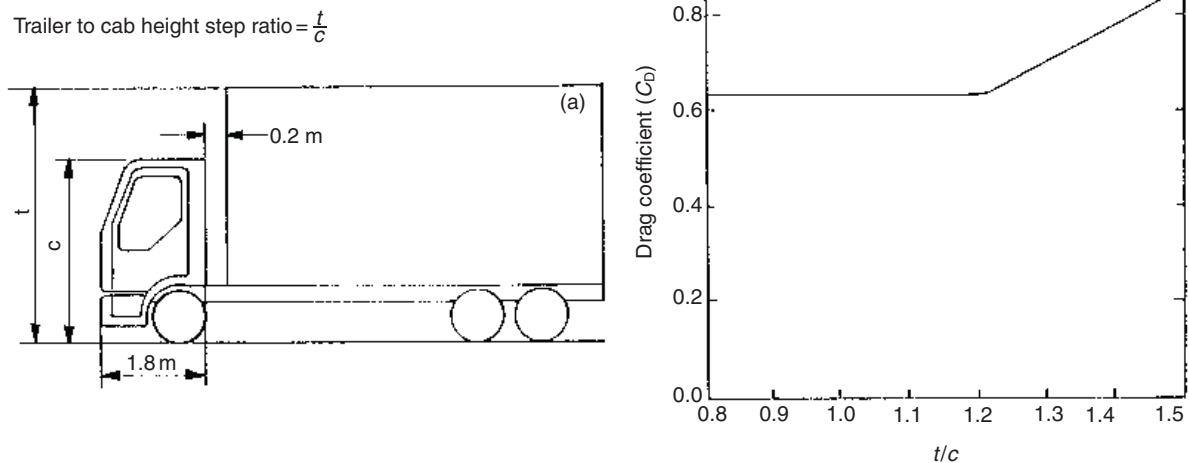

Fig. 14.53 Influence of cab to trailer body height upon the drag coefficient

(a) Cab to trailer height mismatch

(b) Cab to trailer height mismatch bridged with roof deflector
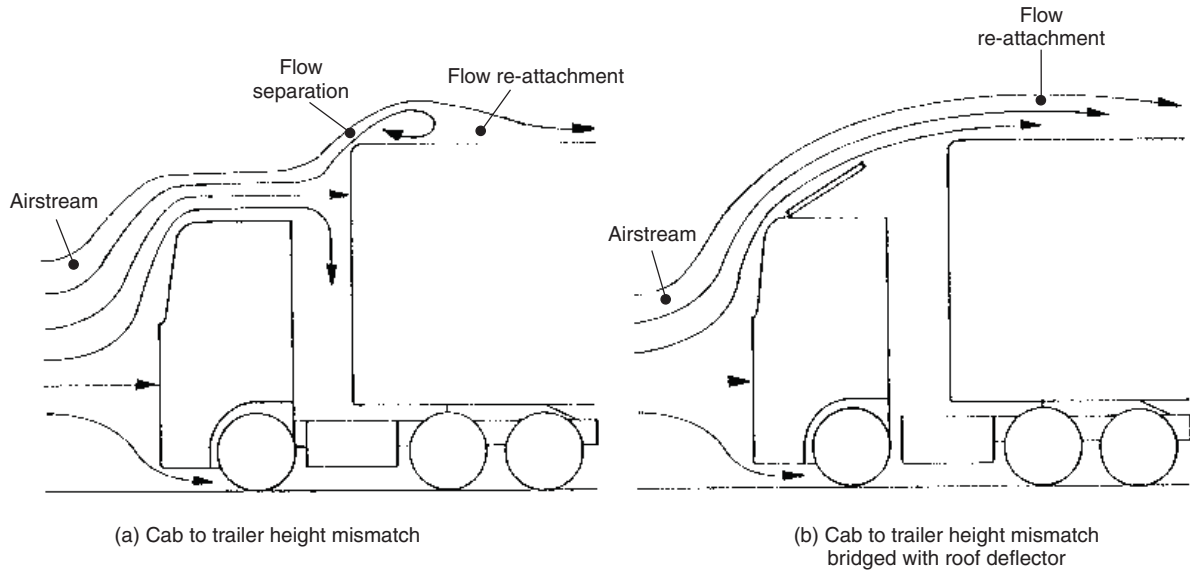
Fig. 14.54 (a and b) Air flow between cab and trailer body with and without cab roof deflector

by using a fibre glass or plastic moulded deflector, see Fig. 14.55(b).

If trailers with different heights are to be coupled to the tractor unit while in service, then a mismatch of the deflector inclination may result which will again raise the aerodynamic drag. There are some cab deflector designs which can adjust the tilt of the cab deflector to optimize the cab to trailer air flow transition (see Fig. 14.55(a)), but in general altering the angle setting would be impractical. How the

cab roof deflector effectiveness varies with deflector plate inclination is shown in Fig. 14.56(c) for both a narrow and a wide cab to trailer gap, representing a rigid truck and an articulated vehicle respectively (see Fig. 14.56(a and b)). These graphs illustrate the general trend and do not take into account the different cab to trailer heights, cab to trailer air gap width and the width to height ratio of the deflector plate. It can be seen that with a rigid truck having a small cab to trailer gap the

(a) Section view

(b) Pictorial view
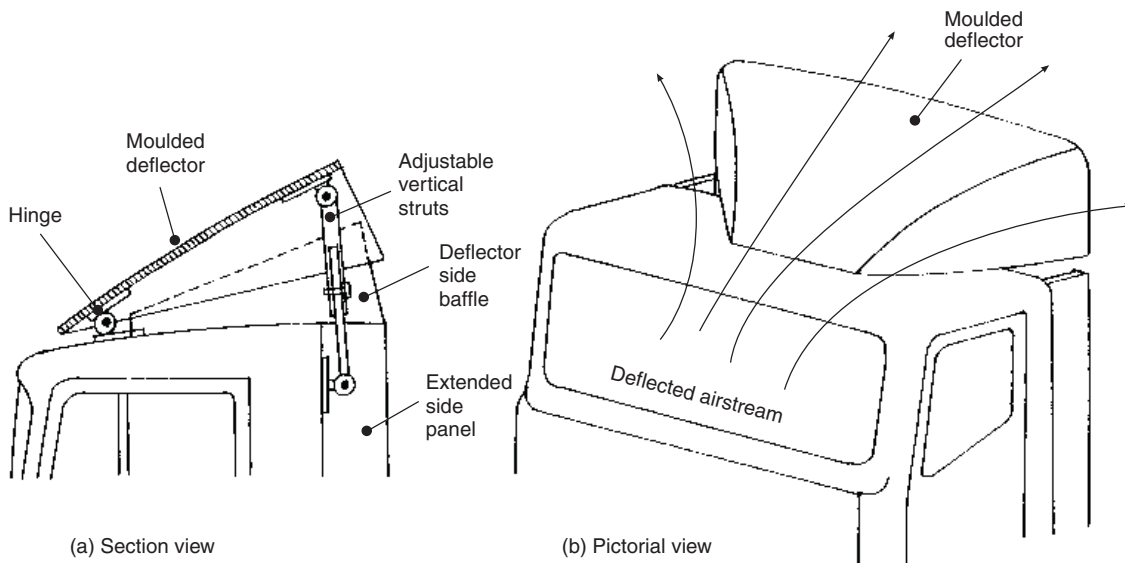
Fig. 14.55 (a and b) Moulded adjustable cab roof deflector

623
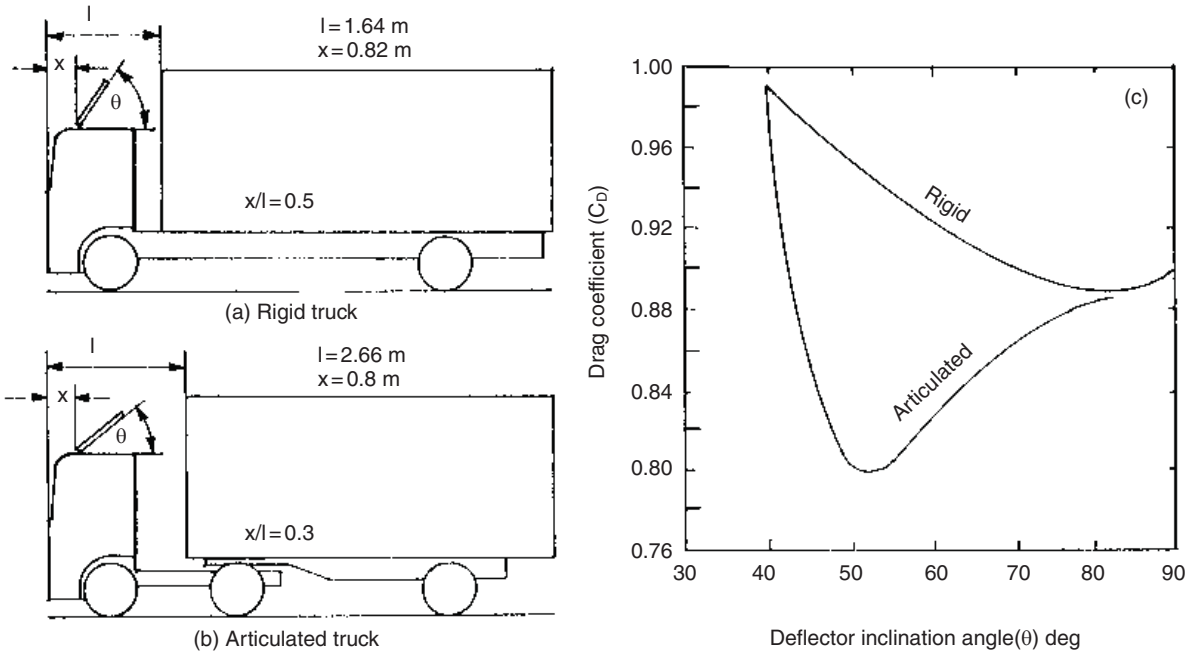
Fig. 14.56 (a–c) Optimizing roof deflector effectiveness for both rigid and articulated trucks

reduction in the drag coefficient with increased deflector plate inclination is gradual, reaching an optimum minimum at an inclination angle of 80° and then commencing to rise again, see Fig. 14.56(c). With the articulated vehicle having a large cab to trailer gap, the deflector plate effectiveness increases rapidly with an increase in the deflector inclination angle until the optimum angle of 50° is reached, see Fig. 14.56(c). Beyond this angle the drag coefficient begins to rise steadily again with further increase in the deflector plate angle; this indicates with the large gap of the articulated vehicle the change in drag coefficient is much more sensitive to deflector plate inclination.

14.8.2 Yaw angle (Figs 14.57 and 14.58)

With cars the influence of crosswinds on the drag coefficient is relatively small; however, with much larger vehicles a crosswind considerably raises the drag coefficient therefore not only does the direct air flow from the front but also the air movement from the side has to be considered. It is therefore necessary to study the effects crosswinds have on the vehicle's drag resistance, taking into account the velocity and angle of attack of the crosswind relative to the direction of motion of the vehicle and its road speed. This is achieved by drawing to scale a velocity vector triangle, see Fig. 14.57. The vehicle velocity vector line is drawn, then the crosswind
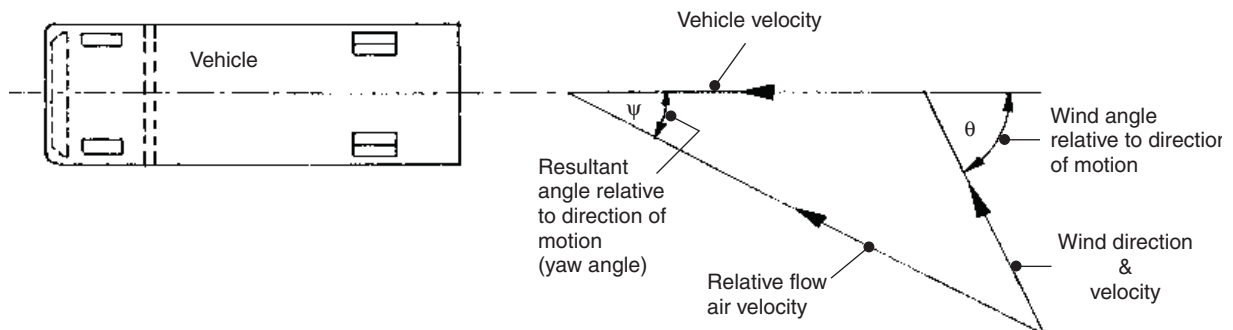
Fig. 14.57 The yaw angle

624

velocity vector at the crosswind angle to the direction of motion; a third line representing the relative air velocity then closes the triangle. The resultant angle made between the direction the vehicle is travelling and the resultant relative velocity is known as the yaw angle, and it is this angle which is used when investigating the effect of a crosswind on the drag coefficient.

In addition to head and tail winds vehicles are also subjected to crosswinds; crosswinds nearly always raise the drag coefficient, this being far more pronounced as the vehicle size becomes larger and the yaw angle (relative wind angle) is increased. The effect crosswinds have on the drag coefficient for various classes of vehicles expressed in terms of the yaw angle (relative wind angle) is shown in Fig. 14.58. Each class of vehicle with its own head on (zero yaw angle) air flow drag coefficient is given a drag coefficient of unity. It can be seen using a drag coefficient of 1.0 with zero yaw angle (no wind) that the drag coefficient for a car reaches a peak of 1.08 at a yaw angle of 20°, whereas for the van, coach, articulated vehicle and rigid truck and trailer the drag coefficient rose to 1.18, 1.35, 1.5 and 1.7 respectively for a similar yaw angle of 20°.

14.8.3 Cab roof deflector effectiveness versus yaw angle (Fig. 14.59(a and b))
The benefits of reducing the drag coefficient with a cab roof deflector are to some extent cancelled out when the vehicle is subjected to crosswinds. This can be demonstrated by studying data taken from
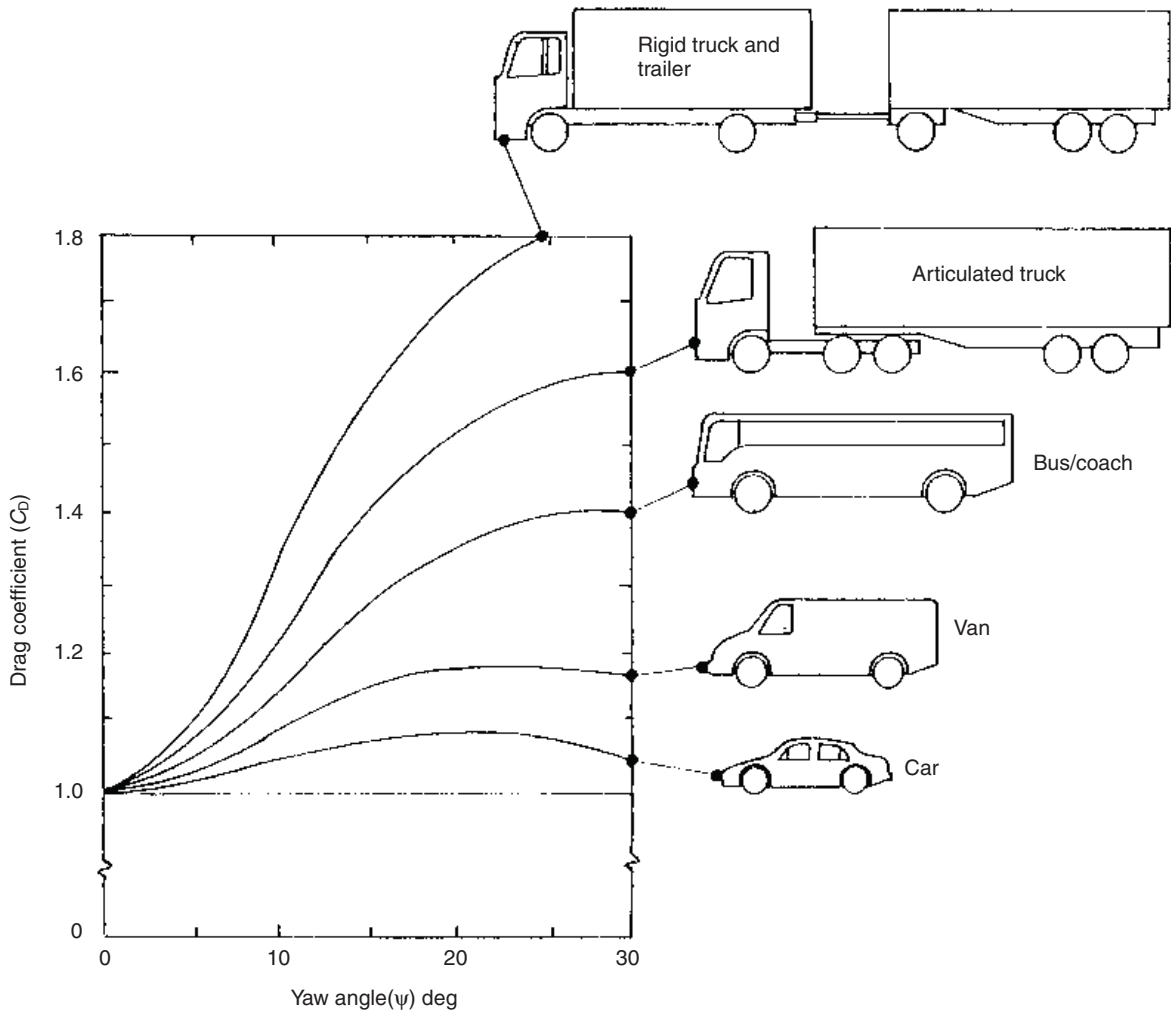
Fig. 14.58 Influence of yaw angle upon aerodynamic drag

625

one particular vehicle, see Fig. 14.59(a and b), which utilizes a cab roof deflector; here with zero yaw angle the drag coefficient reduces from 0.7 to 0.6 as the deflector inclination changes from 90° (vertical) to 50° respectively. With a 5° yaw angle (relative wind angle) the general trend of drag coefficient rises considerably to around 0.9 whereas the tilting of the deflector from the vertical over an angle of 40° only shows a marginal decrease in the drag coefficient of about 0.02; with a further 10° inclination decrease the drag coefficient then commenced to rise steeply to about 0.94. As the yaw angle is increased from 5 to 10° the drag coefficient rises even more to 1.03 with the deflector in the vertical position, however this increase in drag coefficient is not so much as from 0 to 5°. Hence the reduction in the drag coefficient from 1.03 to 0.98 as the deflector is tilted from the vertical to 40° is relatively small compared to the overall rise in drag coefficient due to crosswind effects. However, raising the yaw angle still further from 10 to 15° indicates on the graph that the yaw angle influence on the drag coefficient has peaked and is now beginning to decline: both the 10 and 15° yaw angle curves are similar in shape but the 15° yaw angle

curve is now below that of the 10° yaw angle curve. Note the minimum drag coefficient deflection inclination angle is only relevant for the dimensions of this particular cab to trailer combination.

14.8.4 Comparison of drag resistance with various commercial vehicle cab arrangements relative to trailer body height (Fig. 14.60(a–e))

The drag coefficient of a tractor–trailer combination is influenced by the trailer body height and by different cab configurations such as a conventional low cab, low cab with roof deflector and high sleeper cab, see Fig. 14.60(a–c). Thus a high cab arrangement (see Fig. 14.60(c, d and e)) is shown to be more effective in reducing the drag coefficient than a low cab (see Fig. 14.60(a, d and e)) and therefore for long distance haulage the sleeper compartment above the driver cab has an advantage in having the sleeper area behind the driver's seat. Conversely with a low cab and a roof deflector which has an adjustable plate angle (see Fig. 14.60(b, d and e)), the drag coefficient can be kept almost constant for different trailer body heights. However, it is not always practical to adjust the deflector angle, but fortunately a great many commercial vehicle
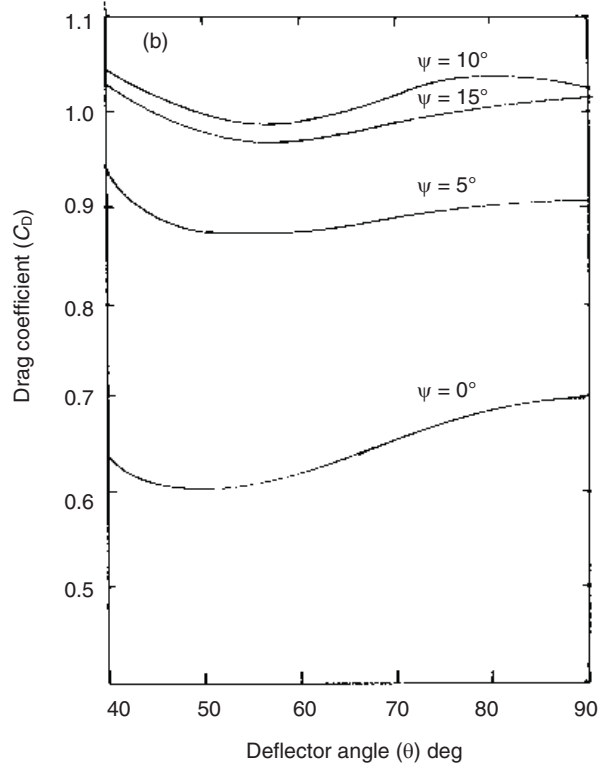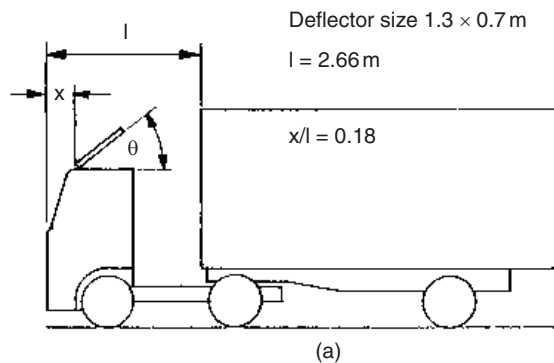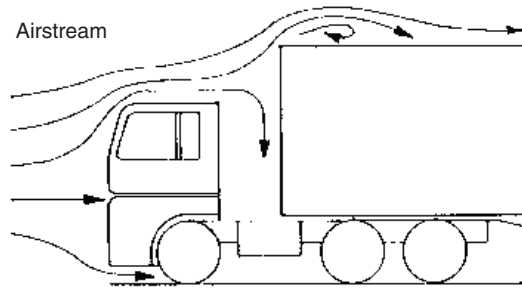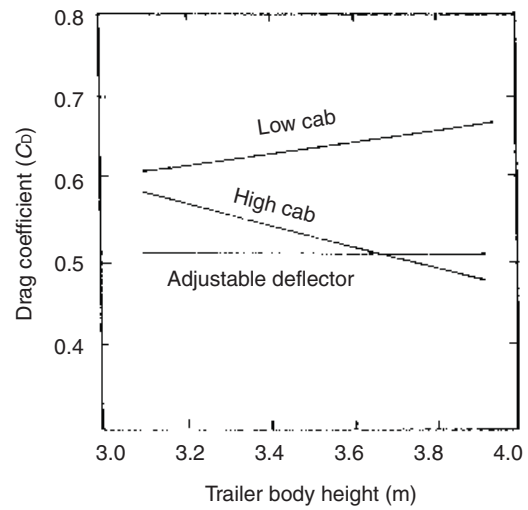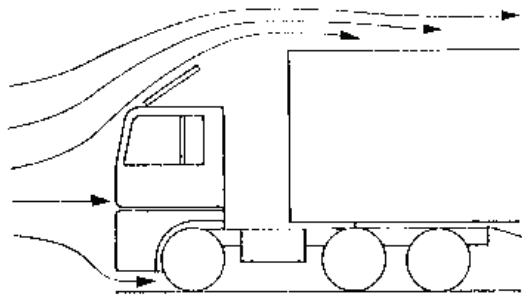

Fig. 14.59 (a and b) Effect of yaw angle upon drag reducing effectiveness of a cab roof deflector
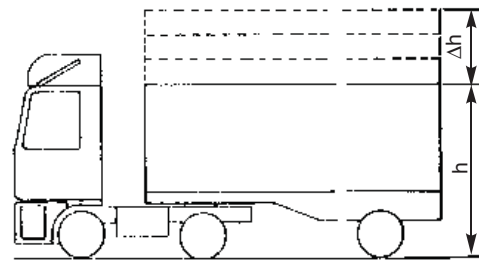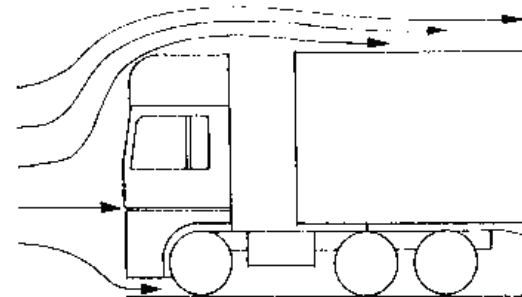
(a) Low cab and high trailer body

(d) Effects of different cab roof configurations relative to trailer body height

(b) Low cab with deflector and high trailer body

(e) Articulated truck with different trailer body heights

(c) High cab and trailer body

Fig. 14.60 (a–e) Methods of optimizing air flow conditions with different trailer body heights

627

cab–trailer combinations use the same size trailer bodies for one particular application so that the roof deflector angle can be pre-set to the minimum of drag resistance. With a low cab the drag coefficient tends to increase as the cab roof to body roof step height becomes larger whereas with a high cab the drag coefficient tends to decrease as trailer body height rises, see Fig. 14.60(d and e).

14.8.5 Corner vanes (Fig. 14.61(a–c))

The cab of a commercial vehicle resembles a cube with relatively flat upright front and side panels, thus with well rounded roof and side leading edges the cab still has a blunt front profile. When the vehicle moves forward the cab penetrates the surrounding air; however, the air flow passing over the top, underneath and around the sides will be far from being streamlined. Thus in particular the air flowing around the side leading edges of the cab may initially separate from the side panels, causing turbulence and a high resistance to air flow, see Fig. 14.61(a).

One method of reducing the forebody drag is to attach corner vanes on each side of the cab (Fig. 14.61(c)). The corner vane is set away from the rounded vertical edges and has several evenly spaced internal baffles which bridge the gap between the cab and corner vane walls. Air meeting the front face of the cab moves upwards and over the roof, while the rest flows to the left and right hand side leading edges. Some of this air also flows around the leading edge through the space formed between the cab and corner vanes (see Fig. 14.61 (b and c)); this then encourages the airstream to remain attached to the cab side panel surface. Air drag around the cab front and side panels is therefore kept to a minimum.

14.8.6 Cab to trailer body gap (Fig. 14.62)

Air passing between the cab and trailer body gap with an articulated vehicle due to crosswinds significantly increases the drag resistance. As the crosswind angle of attach is increased, the flow through the cab-trailer gap produces regions of flow separated on the sheltered side of the trailer body, see Fig. 14.62. This flow separation then tends to spread rearwards, eventually interacting with and enlarging the trailer wake, the net result being a rise in the rearward pull due to the enlarged negative pressure zone.
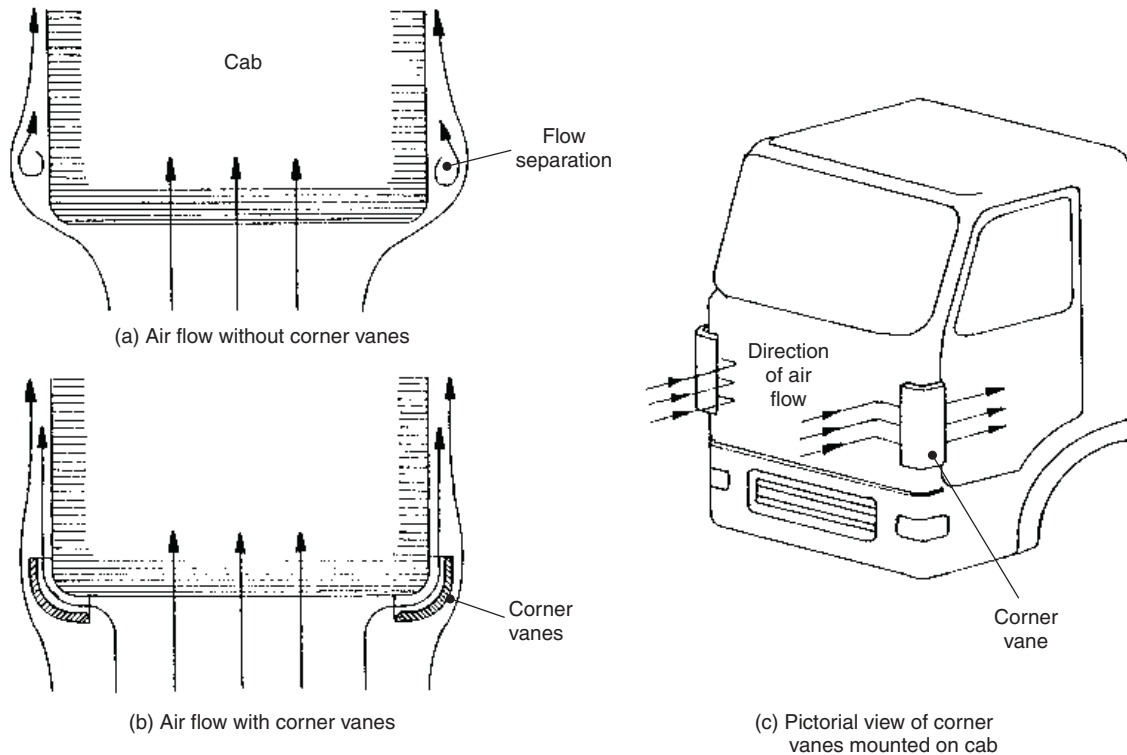

Fig. 14.61 (a–c)   Influence of corner vanes in reducing cab side panel flow separation
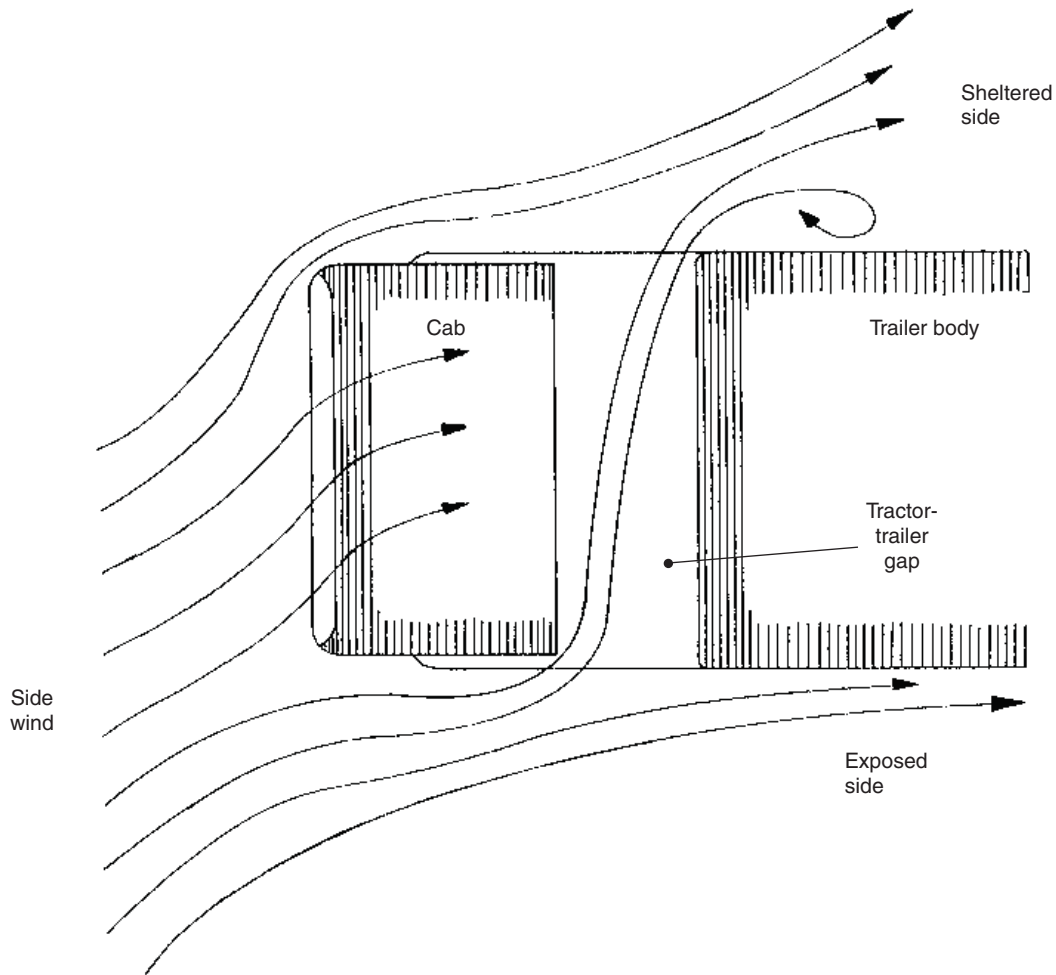
Fig. 14.62 Air flow through tractor-trailer gap with crosswind

14.8.7 Cab to trailer body gap seals (Fig. 14.63
(a and b), 14.64, 14.65(a and b) and 14.66(a–d))
Simple tilt plate cab roof deflectors when subjected
to side winds tend to counteract the gain in head on
airstream drag resistance unless the deflector sides
are enclosed. With enclosed and streamlined cab
roof deflector sides, see Fig. 14.55(a and b), improve-
ments in the drag coefficient can be made with
yaw angles up to about 20°, see Fig. 14.63(a and b).
Further reductions in the drag coefficient are
produced when the cab to trailer gap is sealed by
some sort of partition which prevents air flowing
through the cab to body gap, see Fig. 14.63
(a and b). The difficulty with using a cab to trailer
air gap partition is designing some sort of curtain
or plate which allows the trailer to articulate when
manoeuvring the vehicle-trailer combination.

Cab to trailer gap seals can be divided into three
basic designs:

1 Cab extended side panels
2 Centre line gap seals (splitter plate seal)
3 Windcheater roller edge device (forebody edge
 fairing).

Cab extended side panels (Fig. 14.64) These
devices are basically rearward extended vertical
panels attached to the rear edges of the cab which
are angled towards the leading edges of the trailer
body. This type of gap fairing (side streamlining) is
effective in reducing the drag coefficient with
increasing crosswind yaw angle. With zero and
10° yaw angles a drag coefficient reduction of
roughly 0.05 and 0.22 respectively have been
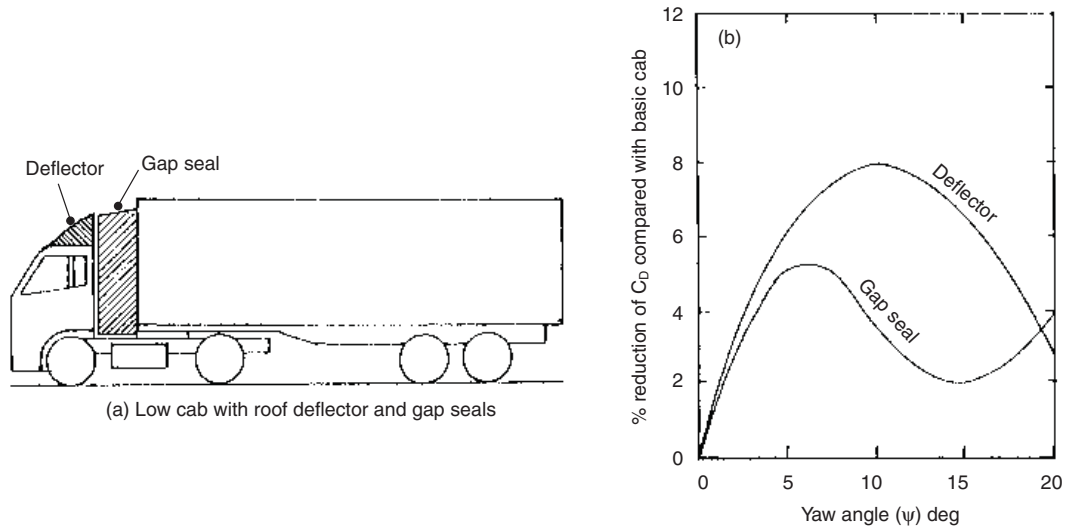made possible.

629

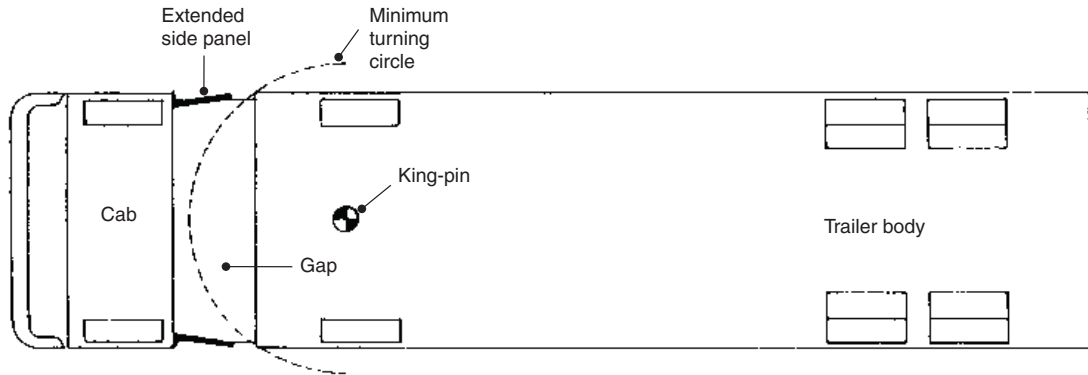Fig. 14.63 (a and b)   Drag reductions with crosswinds when incorporating a roof deflector and gap seal
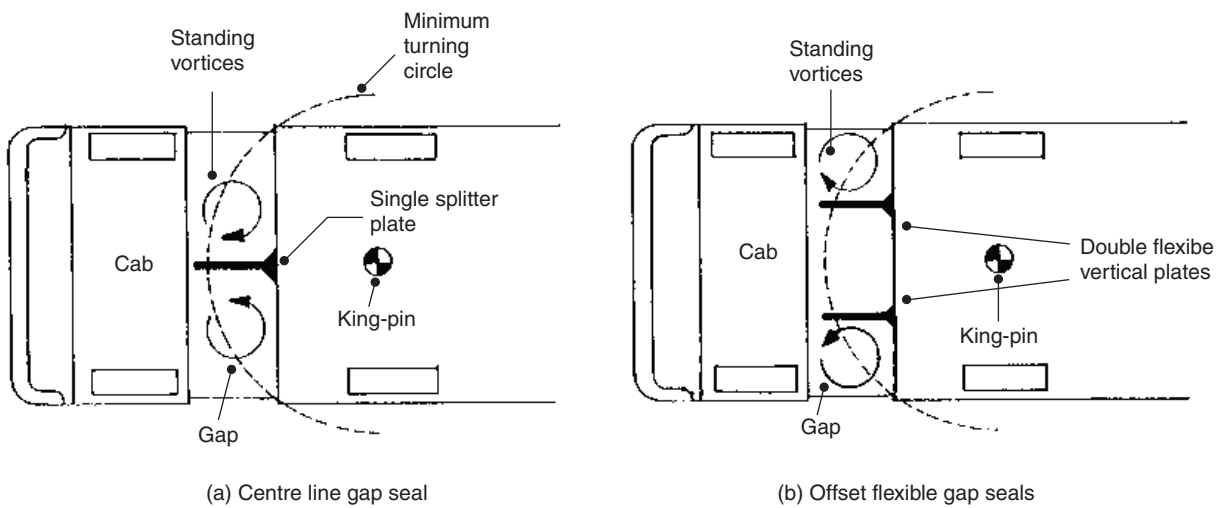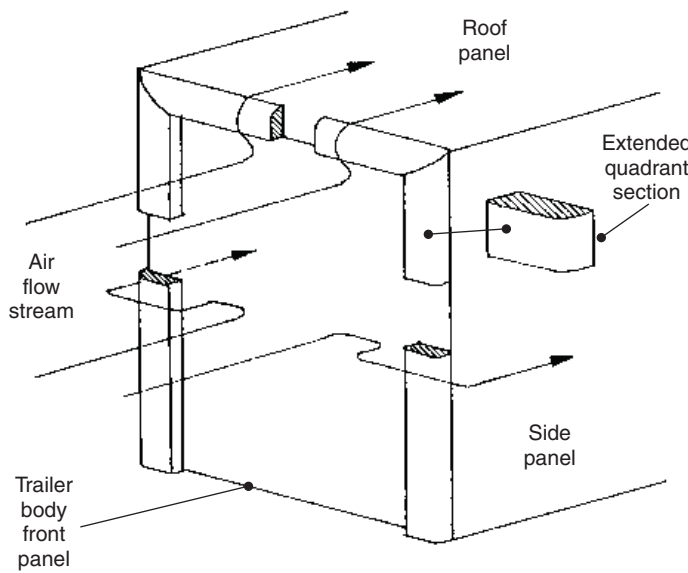

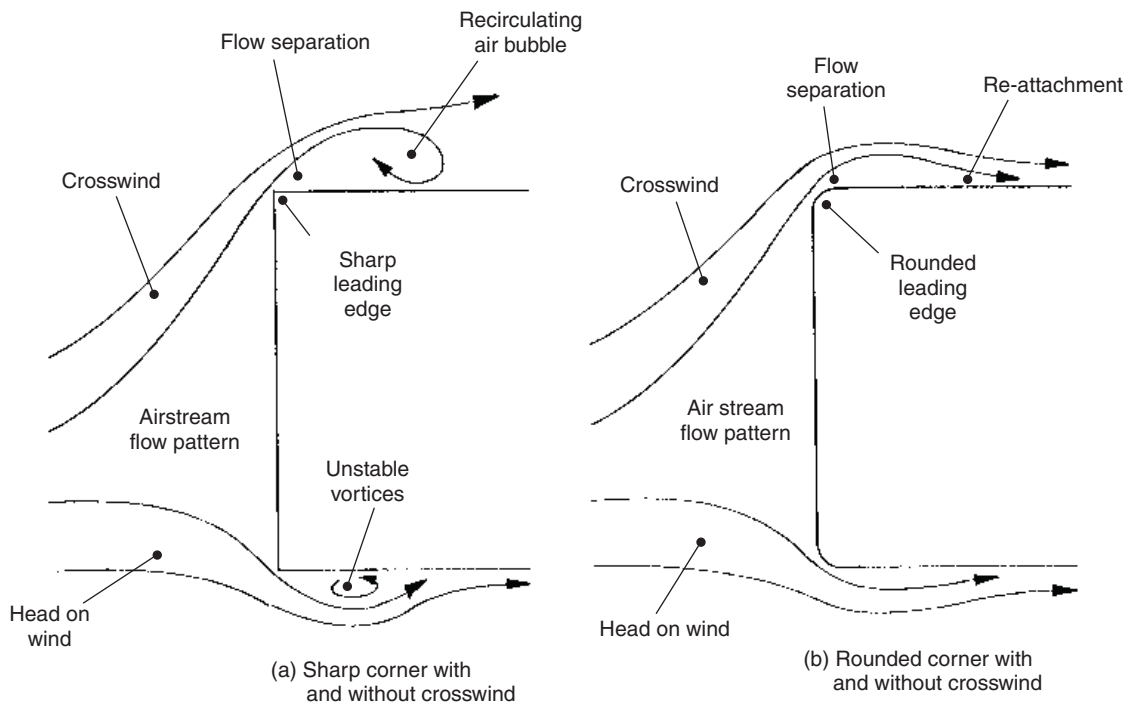Fig. 14.64   Cab extended side panels

(a) Centre line gap seal

(b) Offset flexible gap seals

Fig. 14.65 (a and b)   Cab to trailer body gap seals

(a) Sharp corner with and without crosswind

(b) Rounded corner with and without crosswind

(c) Extended quadrant corner (windcheater)
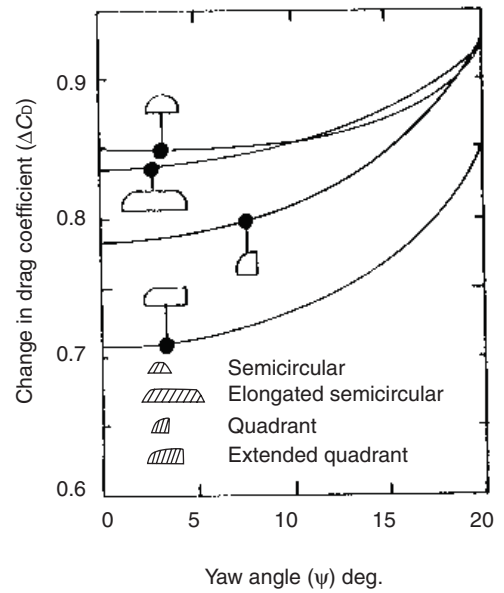
(d) Effectiveness of various forebody edge sections upon drag coefficient

Fig. 14.66 (a–d) Trailer forebody edging fairings

631

Centre line and offset double gap seal (vortex stabilizer) (Fig. 14.65(a and b)) A central vertical partition or alternatively a pair of offset flexible vertical plates attached to the trailer body (see Fig. 14.65(a and b)) is effective in not only preventing crosswinds passing through the cab to trailer gap but also stabilizes the air flow entering the gap by generating a relatively stable vortex on either side of the plate or plates. The vortex stabilizer is slightly less effective than the extended side panel method in reducing the drag coefficient when side winds prevail.

Rolled edge windcheater (Fig. 14.66(a–d)) This device consists of an extended quadrant section moulding attached to the roof and both sides of the leading edges of the forebody trailer panel. When there are sharp leading edges around the trailer body air flowing through the cab to trailer space tends to overshoot and hence initially separate from the side panels of the trailer body (see Fig. 14.66(a)) and even with rounded edging there is still some overshoot and flow separation (Fig. 14.66(b)). The effectiveness of different sectioned forebody edge fairings are compared corresponding to a yaw angle (relative wide angle) variation from 0 to 20°, see Fig. 14.66(d). Here it can be seen that there is very little difference between the semi circular and elongated semi circular moulding but there is a moderate improvement in the drag coefficient at low yaw angles from 0 to 10° for the quadrant section; however, with the extended-quadrant moulding there is a considerable improvement as the yaw angle is increased from 0 to 20°. With the extended quadrant moulding (see Fig. 14.66(c)) the air flow tends to move tangentially between the cab to trailer air gap; some of the air then scrubs along the flat frontage of the trailer body until it reaches the extended-quadrant step, is then deflected slightly rearwards and then again forwards before closely following the contour of the curved corner. This makes it possible for the air flow to remain attached to the side panel surface of the trailer body, therefore keeping the drag resistance on the sheltered trailer panel side to the minimum.

14.8.8 *Tractor and trailer skirting*
(Fig. 14.67(a and b))
Crosswinds sweeping tangentially underneath the tractor and trailer chassis and between and around the road wheels and axles produce a rise in the drag coefficient. To partially counteract the increase in vehicle drag with increased yaw angle, side skirts can be attached either to the trailer or the tractor or both units. The effectiveness of both tractor and trailer skirting for one particular commercial vehicle is shown in Fig. 14.67(a and b); here it can be seen that with increased yaw angle (relative wind angle) the effectiveness of the trailer skirt peaked at a drag coefficient of 0.07 for a yaw
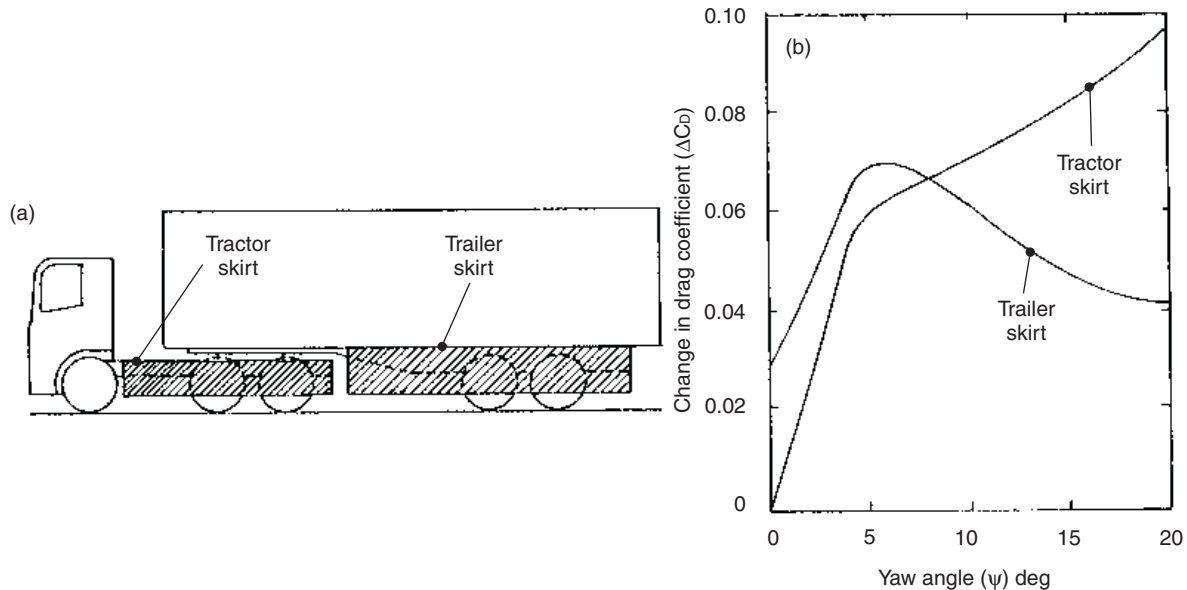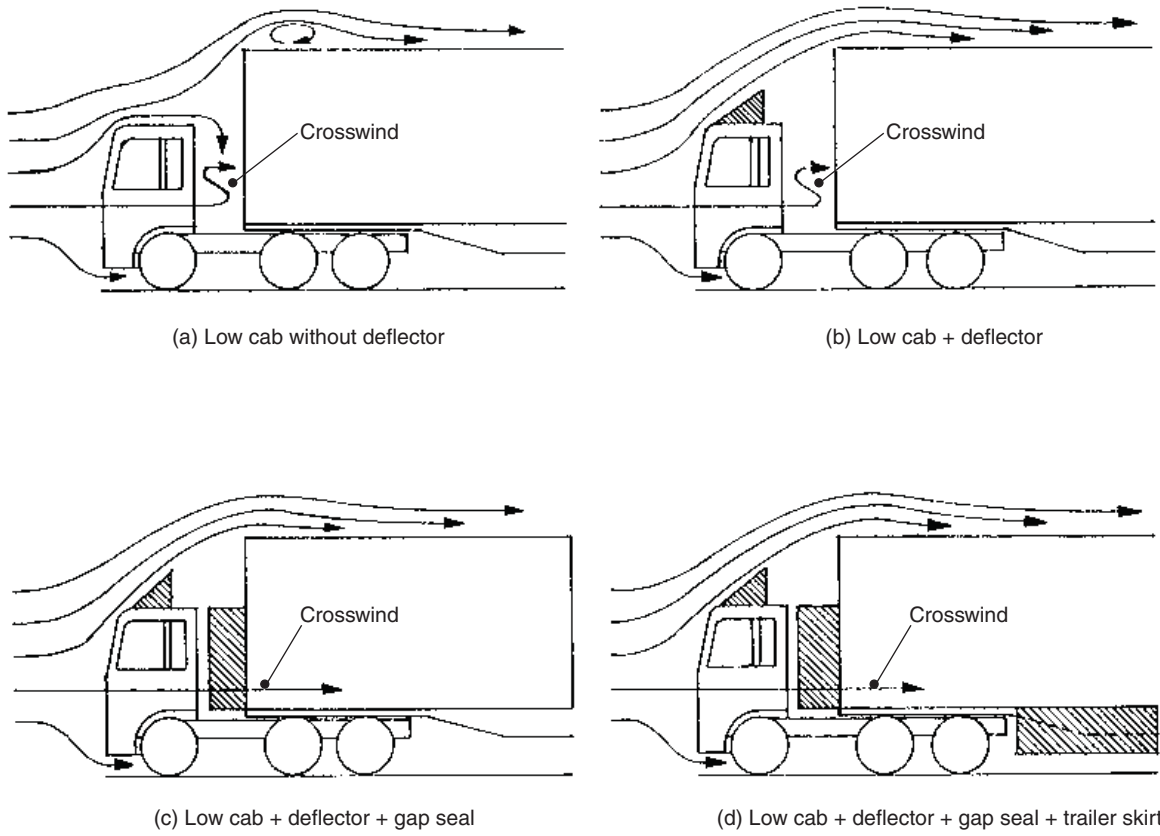

Fig. 14.67 (a and b) Effectiveness of tractor and trailer skirting upon drag coefficient

(a) Low cab without deflector

(b) Low cab + deflector

(c) Low cab + deflector + gap seal

(d) Low cab + deflector + gap seal + trailer skirt

(e) Comparison of the effectiveness of various drag reducing devices
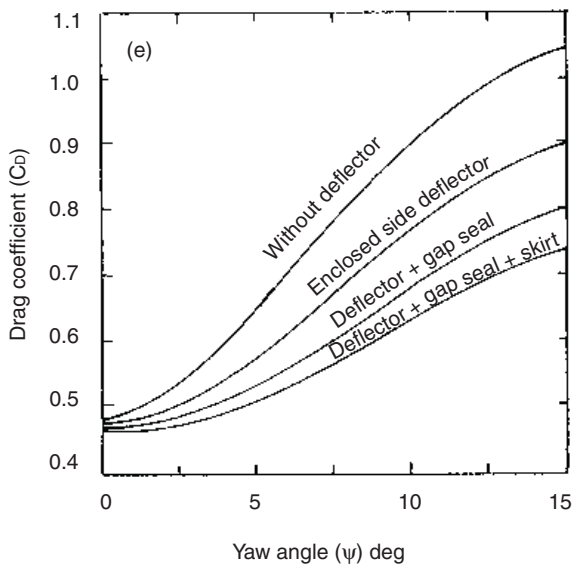
Fig. 14.68 (a–e) Influence of various devices used to reduce drag when vehicle is subjected to crosswinds
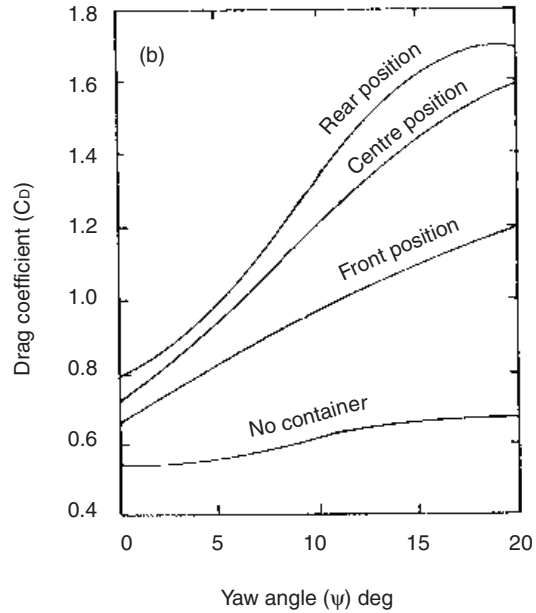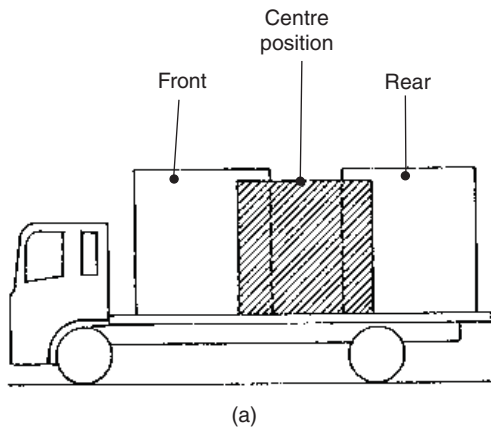
633

Fig. 14.69 (a and b)   Effects of trailer load position upon drag coefficient

angle of 5°. The skirt effectiveness in reducing the drag coefficient then decreases steadily over an increasing yaw angle until it reaches its minimum of 0.04 at a yaw angle of 20°. Conversely the trailer skirt's effectiveness with respect to the yaw angle rose rapidly to 0.06 over a yaw angle range of 5°; the drag coefficient then continued to rise at a slower rate so that for a yaw angle of 20° the drag coefficient effectiveness reached a maximum of just over 0.09. However when considering attaching skirts to a vehicle there can be a problem with the accessibility for routine inspection and for maintenance of the steering, suspension, transmission and brakes; they can also restrict the cooling of the brake drums/discs.

14.8.9 *Comparison of various devices used to reduce vehicle drag* (Fig. 14.68(a–e))

A comparison of various devices used to reduce vehicle drag particularly when there are crosswinds can be seen in Fig. 14.68(a–e). The graph shows for a low cab and no roof deflector that the drag is at its highest due to the cab to trailer step and that the drag

coefficient rises with increasing crosswind yaw angle (relative wide angle) from 0.48 to about 1.05 over a 15° increase in yaw angle (see Fig. 14.68(a and e)); a reduction in the drag coefficient occurs when a cab roof deflector is matched to the trailer body (Fig. 13.68(b and e)). When a cab to trailer gap seal is attached to the trailer there is a further reduction in the drag coefficient particularly with increasing yaw angles (Fig. 14.68(c and e)), and finally there is even a greater drag coefficient reduction obtained when fitting a trailer skirt (Fig. 14.68(d and e)).

14.8.10 *Effects of trailer load position on a vehicle's drag resistance* (Fig. 14.69(a and b))

The effects of positioning a container load on a platform container truck considerably influences the drag resistance. This becomes more noticeable with crosswinds (see Fig. 14.69(a and b)), and that with a yaw angle of 20° the drag coefficient without a container was only 0.7 whereas with the container mounted in front, centre and towards the rear, the drag coefficients reached 1.2, 1.6 and 1.7 respectively.